Degradation Effects on Combined Cycle Power Plant Performance—Part II: Steam Turbine Cycle Component Degradation Effects

A. Zwebek
e-mail: gotross@hotmail.com

P. Pilidis
e-mail: p.pilidis@cranfield.ac.uk

Department of Power Engineering and Propulsion,
School of Engineering,
Cranfield University,
Bedford MK43 0AL, UK

This is the second paper exploring the effects of the degradation of different components on combined cycle gas turbine (CCGT) plant performance. This paper investigates the effects of degraded steam path components of steam turbine (bottoming) cycle have on CCGT power plant performance. Areas looked at were, steam turbine fouling, steam turbine erosion, heat recovery steam generator degradation (scaling and/or ashes deposition), and condenser degradation. The effect of gas turbine back-pressure on plant performance due to HRSG degradation is also discussed. A general simulation FORTRAN code was developed for the purpose of this study. This program can calculate the CCGT plant design point performance, off-design plant performance, and plant deterioration performance. The results obtained are presented in a graphical form and discussed.
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Introduction

It is the nature of power plants that they do not work at all times at their design point conditions. From the experience built up over the past years of operation of such plants it was observed that two main sources are available which offsets the plant from its design point conditions, these are

1. off-design due to normal conditions (changes of ambient conditions and part load) and
2. off-design due to abnormal conditions (changes in fluid path components configuration due to degradation).

While the first problem can be dealt with up to some extent, the second one is really hard to control and it is a function of many interrelated parameters.

According to [1] and others, fouling of heat exchanger surfaces in power plants results in huge economical losses. Von Nostrand [2] estimated the total cost of fouling of heat transfer surfaces for petroleum refining in the non-Communist countries as high as $4.41 billion per year.

Although it has been recognized since, a long time ago the effect of degradation of heat exchanges have on heat transfer, there seems little field data or experimental research work found on this subject in the open literature, [1] and [3].

It is well known that the efficiency of the steam turbine (bottoming) cycle as a part of CCGT plant is primarily a function of GT efficiency, [4], and HRSG efficiency. Therefore, it is very important to see how degraded HRSG affects the steam turbine power plant performance, and hence CCGT plant performance. For the effects of gas turbine component degradation see [4].

Therefore it is becoming of great importance to predict in advance the behavior of thermal plants as they get older and older so as plans can then be made in advance to avoid long shutdowns and hence a lot of economical losses. Also this prediction of plant behavior helps in pre-arranging maintenance plans.

This paper describes how common faults affect CCGT plant performance. In this study, it is assumed that the gas turbine is working at its original design point conditions, regardless of its lifetime. Then by applying a certain amount, as described below, of degradation to different components of the steam (bottoming) cycle the performance behavior of the plant accordingly could have been simulated. In this way the plant's behavior can be monitored.

This study is performed by way of simulation. The gas turbine degradation effects on a CCGT plant were investigated by using Turbomatch (a FORTRAN code available at Cranfield University), [4]. To simulate the degradation effects of steam turbine (bottoming) cycle on the CCGT plant; a new FORTRAN code was developed. The obtained results are then discussed and compared with published data wherever possible.

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Gas Turbine Performance Deterioration

Contrary to the first paper, [4], the strategy thoughout this paper is to keep the gas turbine at its design point conditions while applying different amounts of degradation to the steam turbine cycle components. The only one exception case to this is the simulation of increased back-pressure of the gas turbine as a result of HRSG heat transfer surfaces fouling (see the discussion below).

To have comparable simulation results of CCGT plant components degradation as a whole, it was meant to use the same gas turbine (topping) and steam turbine (bottoming) cycle plants (Fig. 1) that were used in the previous paper, [4]. This helps in having a global overview of the plant's behavior in accordance with different component degradation. It also helps in investigating the effects of degradation of both plants as separate units. And hence identify the sensitive parts of both plants to degradation. Also by comparing the results obtained in both cases a wider understanding of the response of CCGT plants to different component degradation can be observed.

Therefore, main gas path components of the gas turbine, namely compressor, combustion chamber, and turbine, were assumed that they are working at their design point conditions at all times.

In this study a typical gas turbine having the following design point conditions, [4], was used:

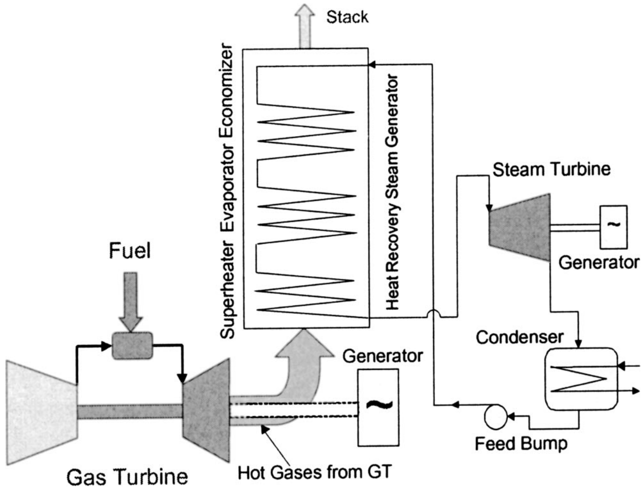

Fig. 1 Schematic diagram of a single pressure CCGT power plant (Zwebek and Pilidis [4])

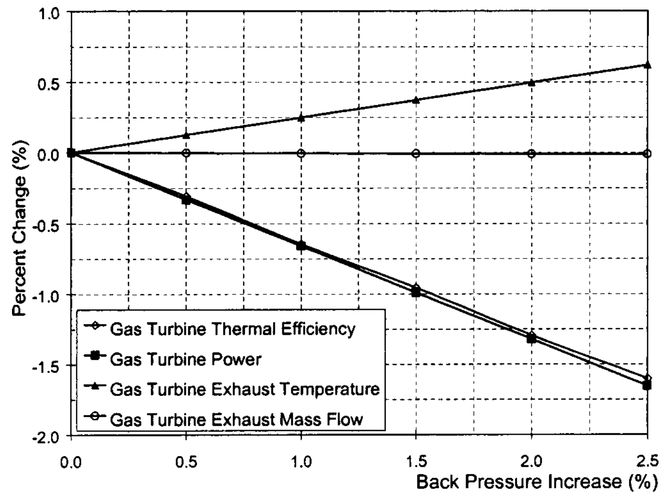

Fig. 2 Back pressure effects on GT performance

inlet mass flow	=	408.66 kg/sec
compressor pressure ratio1	=	15.2
turbine entry temperature	=	1697.80 K
exhaust mass flow	=	419.4 kg/sec
exhaust temperature	=	871.24 K
net power output	=	165.93 MW
thermal efficiency	=	35.57%

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, as in [4], 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 to be implanted on each component has to be established.

Unfortunately, not much literature was found on the subject of CCGT plant degradation, or on modeling of this problem, including the effect of back-pressure. Therefore to simulate the effect of back-pressure on gas turbine performance, due to HRSG degradation some assumptions has been made. According to [5], an increase in back-pressure by ≈ 0.0025 atm results in a reduction in gas turbine power by $\approx 0.3\%$. Because of the inherent problems which accompanies the increase of back-pressure, e.g., high torque on the shaft, coupling forces on thrust bearing, and vibration, it was assumed that maximum it can go up to 0.025 atm over the DP value. Reference [5] stated that typical back-pressure ranges from 0.025 to 0.037 atm above the design value.

Gas Turbine Degradation Simulation Results

As already mentioned above, in this paper the gas turbine of the current CCGT plant was kept at its DP conditions, but because of the increased back-pressure due to HRSG degradation, the gas turbine will not work at its DP condition any more.

As Fig. 2 shows, a back-pressure increase of only 2.5% resulted in a reduction of gas turbine thermal efficiency and power of approximately 1.7%, while the exhaust mass flow was almost constant and the exhaust temperature increased by about 0.65%.

Steam Cycle Performance Deterioration

As with the case of gas turbine gas path components, [4], the steam turbine cycle steam path components are also subjected to degradation due to fouling, erosion, and/or corrosion.

While in the case of the gas turbine almost only one surface (outer) is subjected to these degradation effects, which simplifies, up to some extent, the process of investigating this problem. In the case of a steam turbine plant the degradation is affecting two surfaces of some components of the plant at the same time. In the case of the HRSG and the condenser (heat exchangers) two types of degradation are available, one is the outer tubes surface fouling and corrosion, and another is the innertubes surface scaling or erosion. This is, of course, in addition to the degradation of steam turbine unit.

Even with the latest fuel treatment techniques, the exhaust gases from the gas turbine will contain some chemicals in a form of flying ash and soot which deposits on the outer heat transfer surfaces of the HRSG. Also, the impurities, however, water treatment techniques are used in circulating water deposits on the inner walls of the heat exchanger pipes. These then lead to a reduction in the heat exchanger performance (effectiveness). The condenser will also behave in a similar way. The third cause that leads to steam cycle plant performance deterioration comes from the steam turbine unit degradation.

In order to cover most types of degradation that might attack the bottoming cycle of the CCGT plant, it was assumed that each component might degrade separately. Then all components were assumed to degrade together. This helps to establish the nature of the faults and to assess if they are additive or not. The faults investigated were the following:

1. economizer degradation,
2. evaporator degradation,
3. superheater degradation,
4. steam turbine fouling,
5. steam turbine erosion,
6. ST isentropic efficiency degradation,
7. condenser degradation,
8. combination of all faults mentioned above, and
9. gas turbine back-pressure increase due to heat exchanger (HRSG) surfaces fouling.

Fault Representation

In order to investigate the effects of faults mentioned in the previous section on the steam turbine plant performance as a stand alone unit, and hence on the CCGT plant as a whole, these faults were fed into the program as a percent reduction of the original design point value. This is done as follows:

(i) Heat Exchanger Degradation. The degradation of either of the heat exchangers (economizer, evaporator, superheater, and condenser) was simulated by assuming a percent reduction in the original DP value of the overall heat transfer coefficient of the heat exchanger in concern.

Table 1 Representation of component degradation

Fault	Represented by	Range
Gas turbine back-pressure	GT Back-pressure rise	0.0–(−2.5%)
Economizer degradation	Drop in U	0.0–(−5.0%)
Evaporator degradation	Drop in U	0.0–(−5.0%)
Superheater degradation	Drop in U	0.0–(−5.0%)
Condenser degradation	Drop in U	0.0–(−5.0%)
Steam turbine fouling	Drop in Γ	0.0–(−5.0%)
	Drop in η_T	0.0–(−2.5%)
Steam turbine erosion	Rise in Γ	0.0–(+5.0%)
	Drop in η_T	0.0–(−2.5%)
FOD	Drop in η_T	0.0–(−5.0%)

(ii) Steam Turbine Fouling. The steam turbine fouling is represented by a reduced flow capacity at the inlet of the turbine plus a reduction in turbine isentropic efficiency. By doing so, it is assumed that there is a blockage in the inlet area of the turbine due to particles deposition accompanied by a decrease in its isentropic efficiency due to surface roughness, for example.

(iii) Steam Turbine Erosion. The steam turbine erosion is represented by increasing mass flow capacity at the turbine inlet and at the same a reduction in turbine isentropic efficiency.

The two previously mentioned phenomenon (fouling and erosion) are represented by changing the so-called nondimensional mass flow (Eq. (1)) of the component map.

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

(iv) Component Efficiency Degradation. This fault affects the steam turbine unit. It is modeled by reducing the unit's isentropic efficiency of the appropriate map and keeping all other map parameters at their normal condition. 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.

(v) Gas Turbine Back Pressure. The increased back-pressure at the gas turbine exhaust is represented as an increase in the GT exhaust outlet pressure.

The above-mentioned faults are applied to different components of the plant in different values. Table 1 summarizes these faults and their ranges at which they were applied to each component.

Steam Cycle Degradation Simulation

Before starting any degradation simulation it was necessary to establish a datum working line of the plant. Therefore, by using the developed code, a steam cycle DP was arrived at. This cycle was having the following DP conditions, [4]:

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
economizer surface area	=	38004.1 m^2
condenser surface area	=	3942.9 m^2
HRSG efficiency	=	=81.11%
steam turbine plant power output	=	76454.1 kW
steam turbine plant efficiency	=	33.97%

Having established these DP conditions, the steam cycle was then analyzed in a degraded mode. The amount of degradation applied to each component was really a matter of assumption as there was no such documented work of the similar type in the open literature. Therefore, in the present work, when modeling steam turbine unit, it was assumed that every 1.0% deterioration in mass flow capacity (fouling or erosion) would result in a deterioration of (0.50%) in steam turbine isentropic efficiency. In the case of HRSG degradation simulation, it was assumed that every 1.0% deterioration in all components (economizer, evaporator, and superheater) would result in an increase in the gas turbine back-pressure by 0.5% (see Fig. 2).

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

Steam Cycle Degradation Simulation Results

The most important steam turbine cycle deterioration simulation results are represented graphically in Figs. 3 through 7. It is worth reminding the reader here that the point (0.0) on (Y) axes of all graphs represents the design point value.

Figure 3 below shows the steam turbine power variation with different components degradation. As it can be seen from this figure, the steam turbine unit isentropic efficiency (as an individual fault) was the supreme affecting parameter on the plant's power. When the steam turbine unit isentropic efficiency degraded by 5.0% the reduction in ST power output was about 5.6%. The power deterioration due to steam turbine fouling or erosion, as this figure shows, is less in value than in the case of steam turbine unit isentropic efficiency degradation alone. The reason for this is that the ratio of isentropic efficiency degradation to the ST fouling and/or erosion applied in each case when simulating ST fouling or erosion is 0.5:1.0. This means that the value of ST isentropic efficiency degradation along with 5% of fouling or erosion is 2.5%. Whereas in the case of simulating the ST isentropic efficiency degradation individually with 5.0% degradation, the value used was 5.0%.

It is well known that the ST power is a function of steam mass flow. Now by looking at Fig. 5 it will be observed that evaporator degradation, compared to economizer and superheater degradations, resulted in the highest level of steam mass flow reduction, which then resulted in reducing the plant's power output.

Also as this figure shows, the erosion effects on ST cycle power output was predominant over the effect of steam turbine fouling. In the case of steam turbine fouling by 5.0% the deterioration in ST power was about 2.6%, where as in the case of steam turbine erosion by the same amount (5.0%) the deterioration in ST power was in the region of 3.3%. The reason for this is that in the case of steam turbine fouling the increase in the inlet pressure due to inlet area blockage did compensated for some of the power loss and hence resulted in a lower reduction in ST power output compared to ST erosion.

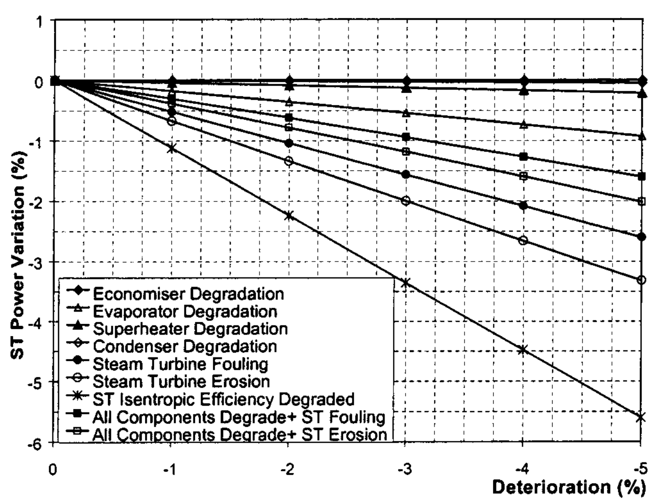

Fig. 3 Steam turbine power variation with component deterioration

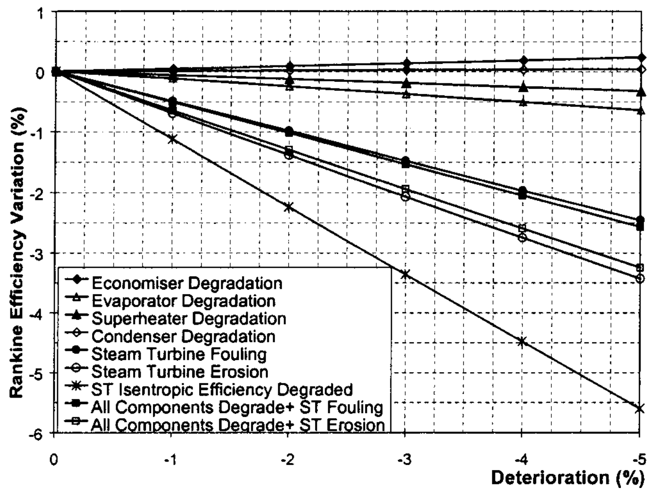

Fig. 4 Rankine efficiency variation with component deterioration

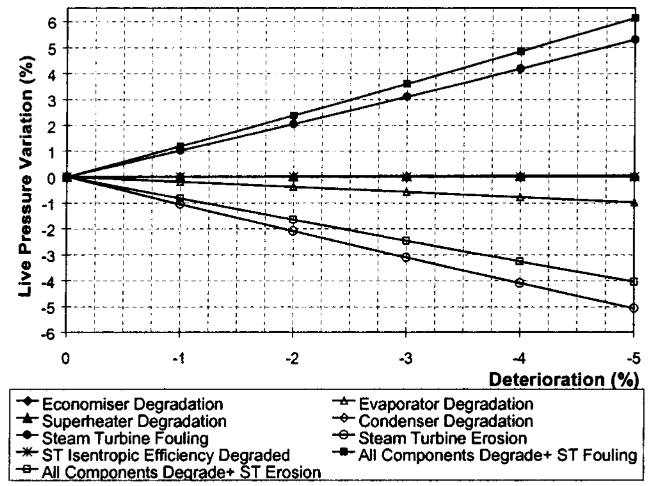

Fig. 6 Live steam pressure variation with steam turbine component deterioration

The GT efficiency has a predominant effect on CCGT power plant efficiency over steam cycle and HRSG efficiencies.

Figure 4 shows the variation of steam cycle (Rankine) efficiency with different types of steam cycle component degradations. In general by comparing Figs. 3 and 4 it will be noticed that the ST cycle thermal efficiency is more or less following the plant's power, since efficiency is a function of power output.

As this figure also shows, while Rankine efficiency is almost constant with condenser and economizer degradations, it showed a little decrease (about -0.32%) with degraded superheater by 5.0%. The largest deterioration in Rankine efficiency due to degradation of any of the three components of the HRSG was encountered with evaporator degradation. This was about 0.64% efficiency deterioration with 5.0% degradation.

As an individual fault, the steam turbine fouling by 5.0% gave about 2.5% reduction in steam cycle efficiency. When added to this the degradation of other components, the increase in Rankine efficiency deterioration was about 0.11%. A similar result was obtained in the case of steam turbine erosion simulation. Among all the faults investigated, the steam turbine isentropic efficiency degradation gave the highest level in Rankine efficiency deterioration. Sanders [6] stated that the degradation of state line efficiency of 1.0% in each section of the steam turbine unit would result in approximately 1.0% deterioration in cycle heat rate. Now, as it can be seen from Fig. 4, the steam cycle efficiency thermal deterioration was approximately 5.6% with 5.0% degradation of steam turbine unit isentropic efficiency degradation. This leads us to two conclusions:

1. The steam turbine isentropic efficiency as a performance parameter has the highest effect on Rankine efficiency.
2. The simulation results obtained from the developed code are in agreement with what the authors put their hands on in the open literature. This gives a sensible confidant in the code developed at Cranfield University, which is the source of simulation result of this paper.

By looking at both Figs. 3 and 4 at the same time, it will be observed that the evaporator degradation was the most affecting fault on ST cycle deterioration. This is due to the fact that the evaporator is producing the largest duty, and hence the reflection of its degradation on the cycle was the highest.

Figures 6 and 7 shows how live steam pressure and temperature varies with different component degradation. As Fig. 6 shows the blockage of the steam turbine inlet due to fouling by 5.0% resulted in about 5.3% increase in live steam pressure at the ST turbine inlet. The combination of all other types of degradation

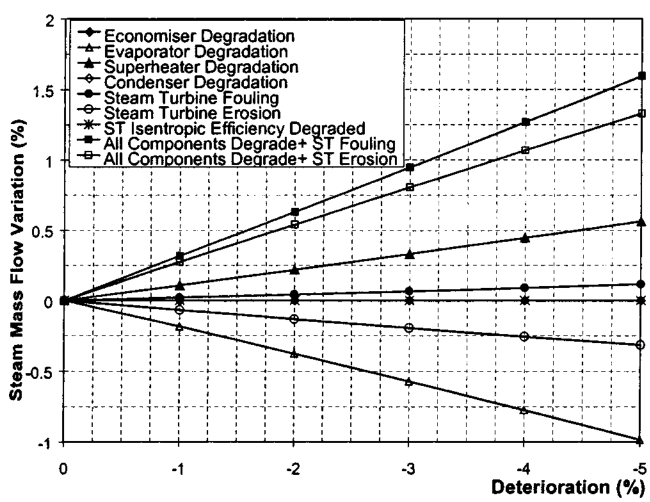

Fig. 5 ST steam mass flow variation with steam turbine component deterioration

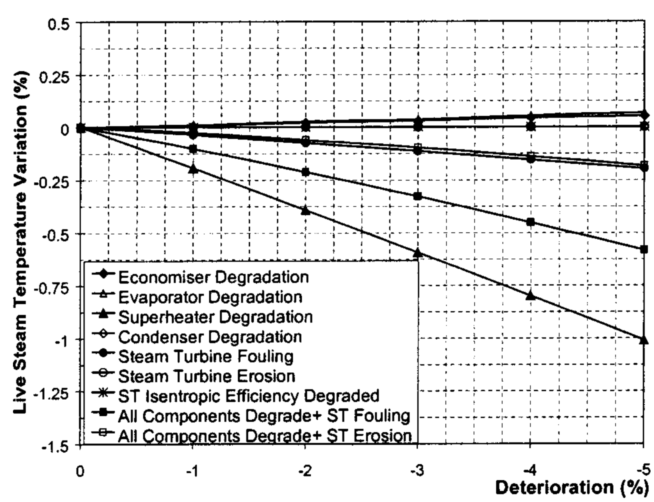

Fig. 7 Live steam temperature variation with steam turbine component deterioration

with steam turbine fouling boosted the inlet pressure to about 6.4%. On the other hand, the steam turbine erosion by 5.0% resulted in about 5.1% reduction in live steam pressure.

As far as live steam temperature concern, the control method during simulation was adjusted in such away that to keep the temperature at the steam turbine inlet not higher than the DP temperature. This is done due to the limitations of mechanical properties of the steam turbine blades. As Fig. 7 shows, the change in live steam temperature was not as high as the change in live steam pressure. The highest change in this case was with superheater degradation, which was -1.1% approximately with 5.0% degradation.

Combined Cycle Degradation Simulation Results

The most important performance deterioration simulation results of combined cycle gas turbine are shown in Figs. 8 through 10. It is an obvious result to find out that the CCGT power output would follow the steam turbine power behavior as the gas turbine power was kept constant during this study, except in the case of increased GT back pressure due to HRSG degradation.

Although the deterioration was high with some steam turbine cycle components degradation, e.g., -5.6% with steam turbine erosion, the GT power output which measures approximately for 2/3's of the CCGT power output did compensate for this reduction in power output.

As Fig. 8 shows, the steam turbine isentropic efficiency degradation and ST erosion accompanied by all components degradation resulted in the higher amount of degradation. As already mentioned, although steam turbine isentropic efficiency degradation by 5.0% resulted in about 5.6% reduction in ST cycle power output, the same amount of degradation resulted in only 1.8% deterioration in CCGT power output. The same comment is applicable to other components degradation.

As in the case of ST power deterioration (see Fig. 3), the effect of steam turbine fouling on CCGT power output (for the same reason mentioned above) was less than the effect of steam turbine erosion.

As Eq. (2) below shows, the combined cycle efficiency is a function of gas turbine cycle efficiency, HRSG efficiency, and steam cycle efficiency, [4].

$$\eta_{CC} = \eta_{GT} + (1 - \eta_{GT}) \cdot \eta_{\mathrm{HRSG}} \cdot \eta_{SC} \tag{2}$$

As per this equation, given that the GT efficiency is constant as mentioned above, any decrease in steam turbine efficiency will give its effect directly on CCGT plant efficiency. Even though, as Fig. 10 shows, there was an increase in HRSG efficiency with

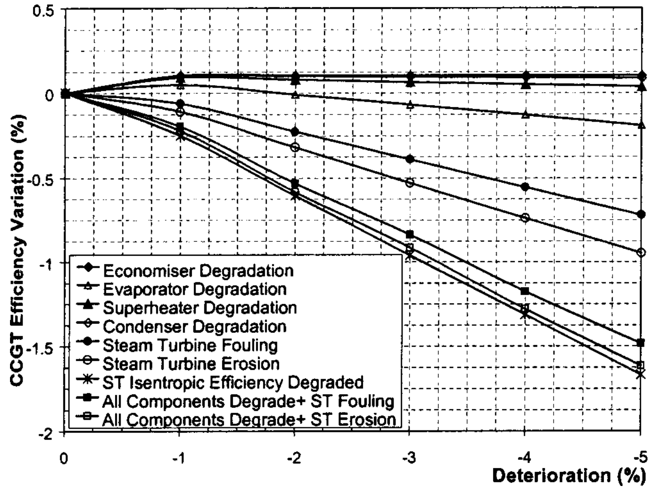

Fig. 9 CCGT efficiency variation with steam turbine component deterioration

some cases of ST component degradation (e.g., all components degrade plus ST erosion), still the effect of decreased ST efficiency for the same fault (see Fig. 4) has a higher effect on CCGT efficiency.

As Fig. 9 shows, the combined cycle efficiency has fallen with all types of ST unit component degradations. The effect of any of the heat exchangers of the HRSG and condenser degradations on CCGT plant efficiency, as can be seen from this figure, is almost negligible. One small exception can be made here where the evaporator degradation by 5.0% led to decrease the CCGT efficiency by about 0.2%.

The stack temperature by it self as a performance parameter is only a measure for the amount of heat extracted from the stream of the GT exhaust gas. It also helps in finding out how far is the HRSG efficient in converting the energy available in the GT exhaust gases to a useful energy. As Fig. 11 shows, the degradation of the combination of all components with ST fouling resulted in the highest increase in stack temperature.

Conclusion

The obtained back-pressure simulation results are in agreement with was found in the open literature.

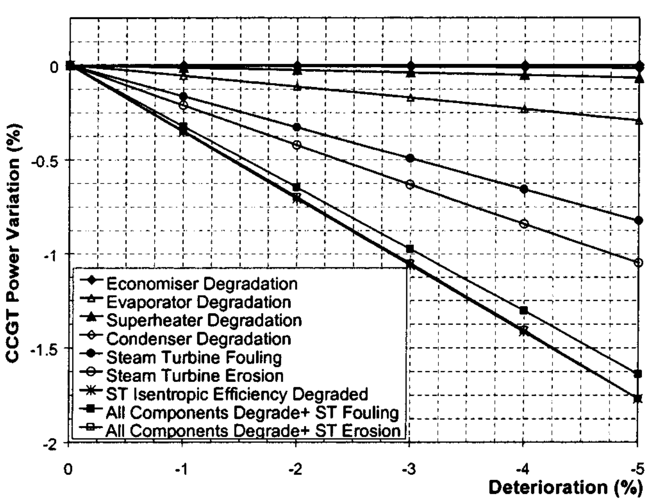

Fig. 8 CCGT power variation with steam turbine component deterioration

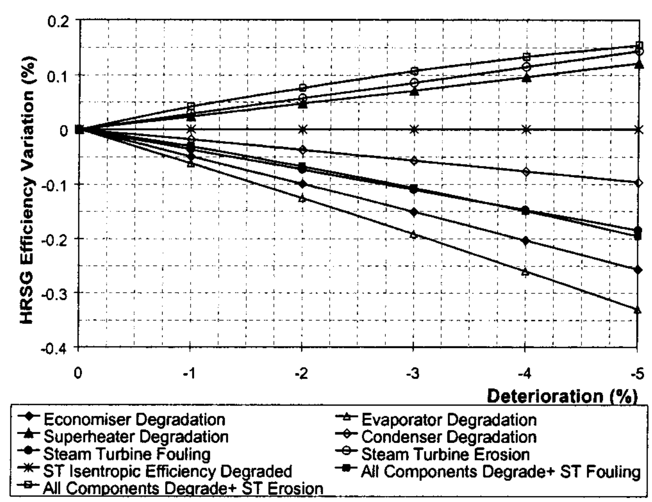

Fig. 10 HRSG efficiency variation with steam turbine component deterioration

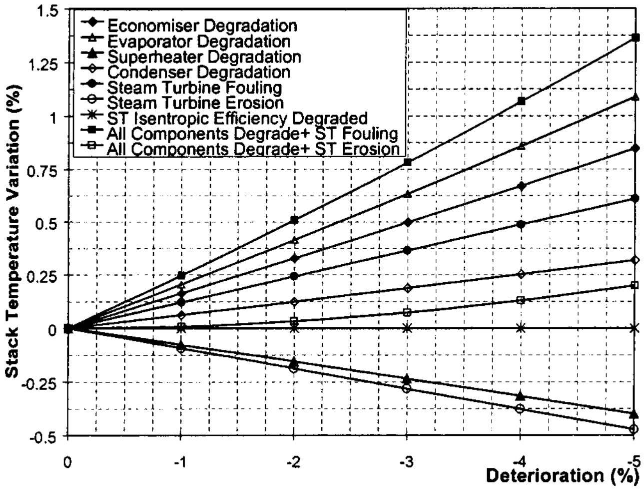

Fig. 11 Stack temperature variation with steam turbine component deterioration

The simulation results obtained from the developed code are in agreement with what the authors put their hands on in the open literature. This gives a sensible confidant in the code developed at Cranfield University, which was the main source of the results in this paper.

Among the three components of HRSG, evaporator degradation is the utmost effecting fault on steam turbine cycle.

The steam turbine isentropic efficiency as a performance parameter has the uppermost effect on steam turbine cycle power and efficiency.

The effect of HRSG and condenser degradations on steam cycle and hence on CCGT plants performance is very low compared to the steam turbine unit components degradation.

Nomenclature

atm = atmospheric pressure
CCGT = combined cycle gas turbine plant
DP = design point
GT = gas turbine
ST = steam turbine
Γ = nondimensional mass flow
η = efficiency
U = heat transfer coefficient

Subscripts

CC = combined cycle
GT = gas turbine
ST = steam turbine
HRSG = heat recovery steam generator
i = Inlet
SC = steam cycle

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