PERFORMANCE AND DEGRADATION EVALUATION OF A COMBINED CYCLE POWER PLANT

By

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Submitted to the Graduate School of Engineering and Natural Sciences
in partial fulfillment of
the requirements for the degree of
Master of Science

SABANCI UNIVERSITY
August 2010

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ME, M.Sc. Thesis, 2010

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Keywords: Combined Cycle, Performance, Degradation, Rehabilitation, Upgrade, Power, Efficiency.

ABSTRACT

The performance of the power plants became an important consideration for energy industry in recent years. Many factors such as the deregulation of the energy market, latest strict environmental rules, depletion of the fossil fuels, continuously increasing high fuel prices and growing energy demand increase pressure on authorities to further consider the power plant performance. Although there are many studies concerning thermodynamic cycles and theoretical performance of various combined cycle power plant configurations, publications about actual combined cycle power plant hardware performance and their observed degradation data remain rather limited in the open literature. This study presents a performance and degradation evaluation study of an actual combined cycle power plant. A case study has been carried out to analyze the performance degradation of the 1350 MW Ambarli Combined Cycle Power Plant which is in service more than 20 years. In order to conduct performance analyses a computer based model of the power plant has been developed complete with all subsystems currently in service. First, the plant performance has been modeled per design specs. Then the calculated design performance has been validated by commissioning test values of the power plant. After this model calibration and validation step, current operating

performance of the plant has been calculated. Degradation analysis of the power plant has been carried out by comparing the design and current performance results. A sensitivity analysis has been performed to illustrate the components' and subsystems' contributions to the overall power plant performance. Possible rehabilitation scenarios for the power plant have been analyzed. Through sample analysis it has been demonstrated how the possible performance improvement alternatives can raise the overall performance of the power plant. Finally, cost-benefit analyses of various rehabilitation options have been performed to find out the most effective rehabilitation combinations.

KOMBİNE ÇEVRİM BİR GÜÇ SANTRALİNİN PERFORMANS VE YAŞLANMASININ DEĞERLENDİRİLMESİ

Tuğrul BAŞARAN

ME, Yüksek Lisans Tezi, 2010

Tez Danışmanı: Doç. Dr. Mahmut F. AKŞİT

Anahtar Kelimeler: Kombine Çevrim, Performans, Yaşlanma, Rehabilitasyon, Modifikasyon, Güç, Verimlilik.

ÖZET

Elektrik Santrallerinin performansı enerji sektörü için son yıllarda önemli bir konu haline gelmiştir. Elektrik piyasasının serbestleşmesi, en son çıkan ağır çevre yasaları, fosil yakıtların tükenmesi, devamlı artan pahalı yakıt fiyatları ve sürekli büyüyen enerji talebi bunun ana nedenleridir. Bu nedenler otoriteler üstündeki baskıyı artırarak santral performansları üzerindeki dikkatleri daha da artırmıştır. Güç santrallerinin değişik kurulum kombinasyonları üzerine bir çok termodinamik ve teorik çalışmalar olmasına rağmen gerçek bir santral üzerine yapılmış performans ve yaşlılığa bağlı kayıpların modellenmesi ve ölçülmesine ait çalışma ve yayınlar çok sınırlı sayıdadır. Bu çalışmada gerçek bir kombine çevrim santrali komple modellenmiştir. Çalışmada örnek olarak 20 seneyi aşan bir süre boyunca çalışan ve 1350 MW kurulu gücü olan Ambarlı Kombine Çevrim Santrali performansı ve ekipman yaşlanmasına bağlı performans düşümü analiz edilmiştir. Analizleri yapabilmek için bilgisayar ortamında santralin bütün alt sistemleri ile beraber bütün bir simulasyon modeli oluşturulmuştur. Öncelikle santralin tasarım performans değerleri hesaplanmıştır. Bu model, kabul testi değerleri kullanılarak doğrulanmış ve kalibrasyonu yapılmıştır. Daha sonra mevcut

durumdaki yaşlanmış santral performansı da hesaplanarak bulunan tasarım performansı ile karşılaştırılmıştır. Böylece karşılaştırmalı olarak performans düşümü analizi yapılmıştır. Her bir ekipmanın santral performans düşümüne katkısını bulmak için duyarlılık analizi de yapılmıştır. Bu analizler yanında muhtemel rehabilitasyon seçenekleri de hesaplanarak değerlendirilmiştir. Olası performans iyileştirme çalışmalarında değişik seçeneklerin performansı ne ölçüde artırabileceği hesaplanmıştır. Son olarak da en etkili rehabilitasyon seçeneklerini bulmak için bu değişik seçeneklerin fayda-maliyet analizleri yapılmıştır.

To my country, Türkiye…

ACKNOWLEDGEMENTS

I wish to express my gratitude to a number of people who have supported me during this thesis work. First of all, I thank my wife and my daughter for their moral support. I want to express my sincere thanks to my parents, who have supported and spent the greatest effort on me. I also thank my father as he encouraged me to study power plants. My thanks to Mahmut F. Akşit, as my thesis advisor. He has given me the opportunity to study power plants, which I really like. I also would like to thank him for being a wise adviser that forced me to observe and learn many things about the subject.

I would like to thank Mr. Serhat Yeşilyurt, Mr. Mehmet Yıldız, Mr. Kürşat Şendur and Mr. Ali Koşar, thank you for kindly accepting to be in being my thesis jury and spending their time for reading my thesis report and attending my thesis presentation.

I also would like to thank Ali Müslim Çeliktepe, the chief engineer of the Ambarlı Power Plant Turbine Maintenance Department for his tremendous help. Also I thank Recep Çıtlak, the technical manager of the Yeniköy Power Plant for his valuable suggestions and opinions. As a special thank, I would like to send my appreciation to all of the Trakya Combined Cycle Power Plant staff for sharing their knowledge and experience. Thank you for your patience, during explaining the power plant operation and answering my numerous questions for days.

Finally, thanks to all my instructors and friends in Sabancı University mechatronics program for being supportive, friendly and helpful.

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TABLE OF SYMBOLS

m	Mass
in	Input
out	Output
E	Energy
P	Power
Q	Heat
W	Work Output
η	Efficiency
η_{carnot}	Ideal Carnot Engine Efficiency
T_L	Low Temperature Reservoir Temperature
T_H	High Temperature Reservoir Temperature
η_{II}	Second Law Efficiency
η_{actual}	Actual Efficiency
η_{ise}	Isentropic efficiency
ε	Effectiveness
Σ	Sum
h	Enthalpy
comp	Compressor
exp	Expander
exh	Exhaust
e	Electricity
imp	Improved
C_e	Electricity Price
C_{NG}	Natural Gas Price
add	Additional
aft	After
bef	Before
Rhb	Rehabilitation

TABLE OF ABBREVIATIONS

CCPP	Combined Cycle Power Plant
CCGT	Combined Cycle Gas Turbine
GT	Gas Turbine
ST	Steam Turbine
GTG	Gas Turbine Generator
STG	Steam Turbine Generator
HRSG	Heat Recovery Steam Generator
Eva	Evaporator
EPRI	Electric Power Research Institute
BTU	British Thermal Unit
HP	High Pressure
LP	Low Pressure
IGV	Inlet Guide Vane
FOD	Foreign Object Damage
NG	Natural Gas
HB	Hourly Benefit
LHV	Lower Heating Value
HGP	Hot Gas Path

1. INTRODUCTION AND LITERATURE SURVEY

1.1. Introduction

Energy is what drives our lives. There is an ongoing global energy challenge caused by increasing energy demand, heavy dependence on oil and other fossil fuels which leads to air, water and land pollution. Large carbon emissions lead to global warming, climate instability and raises health concerns over pollution. Depletion of the fossil resources that are not uniformly distributed globally force the humanity to use the available precious energy resources as efficiently as possible [1].

Electricity power generation industry being the most important energy sector in many countries, faces real problems; the continuous increase in fuel prices, exploding growth in energy demand, the recent strict environmental regulations and the severe competition after the liberalization of the energy market. As a result, power generation authorities seek performance improvements of the power plants. Operating more efficiently is important for the power plants to be able to compete in the deregulated energy market [2].

The rapid improvement of gas turbine technology in the 1990s drove combined cycle thermal power plant efficiency to nearly 60 % with natural gas as the fuel. This efficiency is very high compared to the conventional nuclear and coal-fired power plants. As a result combined cycle power plants, CCPP, (Figure 1) have become the most favorite electric generation facilities as they yield both very good economic and thermodynamic performance compared to the other conventional power plants [3]. Combined Cycle Power Plants are also preferred due to their competitive capital costs, low operation & maintenance costs, good availability, low emissions, short repair time, flexibility, and small number of staff required for production [4]. As a result, most of the recently built power plants all over the world are natural gas fired combined cycle power plants.

The state-of-the-art of the combined cycle power plant components have matured. The gas & steam turbine manufacturers, such as General Electric, ABB-Alstom, Siemens and Mitsubishi claim that they can construct combined cycle plants with very high efficiency up to 60 % [5]. The “Ideal Carnot Machine” efficiency that runs between 1250 °C (typical turbine inlet temperature of the modern gas turbines), and room temperature is 80 %. Therefore, it can be said that the achieved combined cycle efficiency is very high compared to the conventional coal fired power plants as they can only reach around 40 % efficiencies [6]. With new advanced cooling and coating techniques, the manufacturers expect to reach to 1700 °C turbine inlet temperature in the near future, which will boost both the power and the efficiency of the combined cycle plants [7] (Figure 2). Although the natural gas prices have become very expensive, this high cycle efficiency allows combined cycle power plants to remain competitive in the energy market [8].

. Figure 1: A Combined Cycle Power Plant

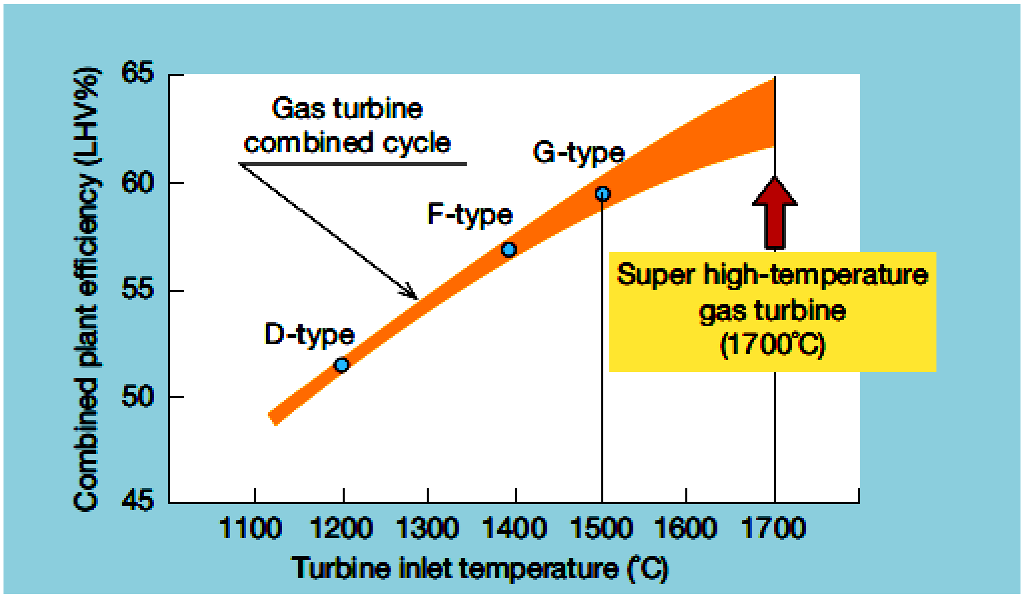

Figure 2: Turbine Inlet Temperature vs. CCPP Efficiency [7]

Today, fossil fuels are the main resource for the electricity generation industry. However, these fossil resources are expected to deplete in the near future. In some reports, the expected depletion period is around 60 years for natural gas, 40 years for oil and 150-200 years for coal [9]. Therefore, cycle efficiency becomes a main concern for these fossil fuel fired power plants. On the other hand, the electricity demand continuously increases [9]. Therefore, the capacity of the power plants using fossil fuels should be increased to meet some of this demand. At the moment, highly efficient combined cycle power plants are available on market [5]. However, constructing new plants requires large amount of capital. As a result, alternative solutions such as upgrading or life extension of the existing plants become ever more attractive. The operators can get performance increases and great benefits with these rehabilitations [10].

Components of combined cycle power plants degrade even with good air filtration and burning clean fuel. The flow path of the engine will face fouling, erosion, corrosion and other degradation mechanisms such as rust scale or oxidation. These degradation mechanisms lead to performance deterioration of the engine [11]. Since the steam cycle in a combined cycle operation depend on the gas turbine exhaust gas flow, the degradation of the gas turbine also affects the steam cycle performance [12].

All the facts that are presented above force the energy authorities to focus more on the power plant performance. As a result, manufacturers, universities and operators have focused on performance issues of the power plants in recent years.

This study aims to observe the performance characteristics of a combined cycle power plant, and how the degradation of various subsystems affects its overall performance. It is important to identify how individual plant component degradation attributes to overall power plant performance loss. This information is very important to decide which components should be overhauled or replaced. This information can also help operators if they decide to make retrofits on the existing equipments like gas turbines and heat exchangers. As a result comprehensive thermodynamic analyses should be performed to identify the effects of such component degradations on overall plant performance. After such a thermodynamic study, an economic analysis must also be performed to determine whether the retrofits justify their costs. In order to find the most cost effective upgrades or life extensions, detailed cost benefit analyses should be carried out before any performance improvement activity. These efforts will reduce operating costs significantly, and save many other expenses to the power plant operators.

1.1.1. Combined Cycle Power Plants

Combined Cycle Power Plants utilize both gas turbines and steam turbines to generate electricity. In a combined cycle power plant, the gas turbine is coupled to a generator to allow it to produce electricity even it runs solo without a steam turbine. As the exhaust stream of the gas turbine has high energy, the gas turbine exhaust is connected to a heat recovery steam generator (HRSG) where steam is produced from the waste heat in the exhaust gas. The generated steam is used to turn the steam turbine that produces more electricity in addition to the simple gas turbine cycle. The steam turbine is connected to a condenser where the excess heat is rejected to the environment. These equipments are the main components of a combined cycle power plant. These are also the components which determine and dominate the performance of a power plant. Apart from these main components, there are many auxiliary systems such as cooling systems, lubrication systems, pumps etc. in a combined cycle power plant [13].

The steam generated at HRSG can also be used in different industrial processes or heating of the buildings. Such a configuration is called a cogeneration plant [14].

Table 1 shows the General Electric's Combined Cycle Power Plant fleet performance [5].

Table 1: GE Combined Cycle Fleet Technical Data [5]

Combined Cycle Designation	Net Plant Power (MW)	Thermal Efficiency (%, LHV)
S106B	64.3	49
S206B	130.7	49.8
S406B	261.3	49.8
S106FA	107.4	53.2
S206FA	218.7	54.1
S109E	189.2	52
S209E	383.7	52.7
S109EC	259.3	54
S209EC	522.6	54.4
S109FA	390.8	56.7
S209FA	786.9	57.1
S109H	480	60

1.2. Literature Survey and Motivation

In the open literature there are many studies about combined cycle power plant performance issues. Although there are many studies concerning thermodynamic cycles and theoretical performance of various combined cycle power plant configurations, publications about actual combined cycle power plant hardware performance and their observed degradation data remain rather limited.

The researchers aim is to find the optimum design parameters for combined cycle power plants to increase their performance. Koroneas et al. [15] studied optimum gas turbine and combined cycle operating design parameters that maximize the thermodynamic efficiency. They considered both simple and combined cycle operation. Bassily A.M. [16] also studied optimum design parameters for dual-pressure reheat combined cycle. Karthikeyan et al. [17] optimized HRSG design parameters for a particular gas turbine design.

Many authors used exergy concept and second law of thermodynamics to find the optimum cycle that maximizes the performance of the combined cycle. However, the recent studies revealed that considering only the thermodynamic parameters is not sufficient to evaluate the performance. It is realized that the economic factors should be also considered in energy facility optimization studies. This is an expected consequence, because improving the power plant components generally increase the costs. For example, the heat surface of the heat exchangers should be increased for better performance. However, increasing the heat surface means additional tubes and thus additional costs. As a result, the researchers concluded that the analyses that only consider thermodynamic parameters do not meet today's necessities. Economic parameters should be also considered in energy facility optimization studies. Therefore, a new discipline called Thermoeconomics is developed. In this new discipline, the thermodynamic and economic concepts are combined together. Nowadays, most of the researchers in this field use this concept to evaluate the performance issues of the energy systems.

Several thermoeconomic optimization methods are developed in the literature [18-20]. These methods are focused on the design phase of the power plants. Von Spakovsky et al. [21] also considered environmental parameters. They developed a multi-objective optimization method where they implemented environmental parameters to the thermoeconomic analysis. The thermoeconomic literature is reviewed in detail by Kanoglu et al. [22].

The thermoeconomic optimization methods are used extensively in recent design optimization studies. The researchers used thermoeconomic principles to optimize the design parameters of combined cycle operation. Some of the studies considered both the gas turbine and HRSG design parameters for the thermoeconomic optimization [23,24]. Li et al. [25] also included environmental parameters to the thermoeconomic optimization.

The gas turbines have matured design. Generally, it is not possible to customize the gas turbine design parameters. The power plant designers have the only option to select a model from the manufacturer gas turbine fleets considering the desired gas turbine power output. The designers do not have a chance to choose a steam turbine that fits their optimal design either. Actually in a CCPP, the only equipment that can be customized by the designers is the HRSG. As a result, the researchers generally focused on HRSG optimization rather than the whole plant [26-28].

Many researchers studied design operating parameters of HRSG system. Mohagheghi et al. [29] performed thermodynamic optimization using genetic algorithms to find the optimum layout for a HRSG. They presented a method to find the optimum HRSG layout. However, they did not include the economic parameters.

Most of the operating power plants in the world are quite old. The actual restrictions of the energy market force the authorities to optimize, recover and improve the performance of these old power plants. As a result, many other researchers have focused on this demanding area.

The exergy analysis is widely used as a tool to find possible improvement opportunities in combined cycle power plants. The researchers analyzed equipments that have the greatest exergy losses in existing power plants [30-32]. Researchers also performed thermoeconomic optimization of existing power plant operating parameters in their studies [33,34]. In their studies, they found optimum set points of the operating parameters that maximize the profits.

Some of the researchers observed the off-design performance of combined cycle power plants. Arrieta et al. [35] observed the influence of the ambient temperature on CCPP performance. Unver et al. [36] included part load variation effects in addition to ambient temperature impacts. Wu [37] performed a sensitivity analysis to find the effects of various operating parameters on the performance. Chuang et al. [38] investigated the effect of condenser pressure variations on the overall performance.

There are additional auxiliary systems that improve CCPP performance. Inlet cooling of the gas turbine is the most popular option. Many published studies consider improving the CCPP performance with this technique. Bhargava et al. [39] and Boonnasa et al. [40] also analyzed this issue.

The power plant components degrade with time. These equipment degradations decrease the overall cycle performance. It is very important to find the impact of the equipment degradations on the overall performance. However, there are few published papers about the issue. Zwebek et al. [12,41,42] observed equipment degradation impacts on the overall combined cycle performance. Kurz et al. [43] observed the degradation effects on industrial gas turbines.

Combined cycle power plants are very complex facilities. The equipments are connected to each other, and any deviation of an equipment highly affects operation of the other equipments. As a result, it is very difficult to determine the magnitude of the equipment degradation. However, it is very important for the operators to know which equipment has degraded. According to this valuable information, they can carry out corrective actions. In recent years, several researchers are focused on degradation analysis of combined cycle power plants. They found the term "degradation diagnosis" for these studies. They used thermoeconomic principles in their studies. Valero et al. [44-47] developed degradation diagnosis methods. Other researchers also performed studies [48,49] where they presented degradation diagnosis methods. The researchers applied these methods on existing power plant configurations [50-53].

It is stated that degradation is unavoidable for power plants. In addition to that it is a cumulative process. At some point, the operators should perform corrective actions. Some of the degradations can be recovered by just cleaning the machine. However, most of the time repair or replacement of the parts is needed to recover degradation. If a part is changed with a new one, it is called life extension. On the other hand, the technology is evolving continuously and the manufacturers research and develop advanced materials and components to increase the equipment performance. They offer upgrade packages that increase the engine performance. Life extension and upgrades are generally named as rehabilitations. In the literature there are few studies [10,54-61] about the issue.

It should be noted that most operating power plants have unique equipment combinations and custom cycle configurations that are selected to efficiently operate at the site conditions. This work presents a case study for the performance evaluation and degradation of the 1350 MW Ambarlı Combined Cycle Power Plant, which is in service more than 20 years.

The studies in the literature, generally used second law of thermodynamics to observe and interpret the performance issues. In this study, first law of thermodynamics is also included in performance evaluation methods. It is shown that the first law parameters can be an effective performance indicator with proper use. In this study the effects of the degradation mechanisms on the performance are also presented. A useful and practical degradation diagnosis procedure is developed, and a case study is carried out. The degradations and their impact on the overall combined cycle performance are presented. In this study, rehabilitation options are also studied. Possible performance improvement scenarios are observed. A cost benefit analysis of the possible rehabilitation options is also performed.

Although there are many theoretical studies about the optimum performance of a combined cycle power plant, most do not consider practical restrictions. For example, the exergy studies conclude that the greatest exergy loss occurs at the combustion chamber of the gas turbine. In fact, the big loss at the combustor is well known; however, thermal resistance of the available materials limit the turbine inlet temperature in practice. Another fact, is that the HRSG manufacturers are unlikely to change their matured design. The HRSG design does not involve only the thermodynamic calculations. Two-phase flow phenomenon is also a major consideration for the HRSG design. The two-phase flow is very complex with many aspects not well understood. Therefore, manufacturers rely on their experience and experiments. Similarly, gas and steam turbine manufacturers also rely mostly on their own experience for practical system limitations. This work presents a power plant model with limitations of the existing hardware configuration.

The thesis is arranged in seven chapters as follows. Chapter 1 provides an introduction to the subject, and presents the literature reviewed. In Chapter 2, a review of the thermodynamic principles that are related with the study is provided. Chapter 3 is dedicated to the performance concepts pertaining to the combined cycle power plants. These concepts are utilized throughout the power plant modeling process and performance analyses. Chapter 4 deals with degradation issues. In this chapter degradation mechanisms and their effects on power plant performance are revealed. In Chapter 5, rehabilitation options are discussed. Opportunities to recover or improve the degraded power plants are presented in this chapter. Chapter 6, provides details of the modeling process for Ambarlı combined cycle power plant. The results of the analyses performed with this model, are also presented. Once the model is established for the complete power plant, Sensitivity and Cost Benefit analyses have been performed, and the results are presented in Chapter 7. Finally, a summary of the findings and conclusion statements are presented in Chapter 8.

2. POWER PLANT THERMODYNAMICS

Thermodynamics is a scientific discipline that analyze the energy conversion. Fundamental thermodynamic knowledge is a must to study the performance of a power plant.

In this section brief information on thermodynamics is given. There are 4 main rules in thermodynamics. Here only 1^{st} and 2^{nd} rules are discussed as they are used extensively in performance studies. Similarly, related information about Rankine and Brayton cycles are presented.

2.1. First Law of Thermodynamics

The First Law of Thermodynamics is commonly recognized as the Conservation of Energy principle, that means energy can neither be created nor destroyed. It transforms into various different forms such as chemical energy, mechanical energy, heat, etc. [64].

In simple terms, for any closed or open control volume, energy transfer occurs with mass and energy crossing the control boundaries. These energy types can be work and heat crossing the boundaries,. The mass flow of the fluid with kinetic, potential, internal, and "flow" energy affects the overall energy level of the system. The energy balance is completed with the stored energy in the control volume [64].

A thermodynamic system are commonly categorized in three types: isolated, closed, or open. The open system is the most general one. In these systems mass, heat and external work are allowed to cross the control boundary. The energy and mass balance is such that all energy coming into the system is equal to energy leaving the system plus the change in storage of energy within the system. The components in power plants are generally open systems and in these equipments there is no stored energy. In other words, the total input energy and the total output energy of an open system are equal. This information is very important in calculating the heat and mass balances that reveal the thermodynamic behavior of the equipments like a steam turbine [64].

The heat and mass balance equations for an open control volume at steady state operation are stated as follows [65]:

The mass balance:

$$\sum m_{in} - \sum m_{out} = 0 \qquad [1]$$

where;

$\sum m_{in}$ = total mass that enters the control volume

$\sum m_{out}$ = total mass that exit the control volume

The energy balance:

$$Q - W + \sum E_{in} - \sum E_{out} = 0 \qquad [2]$$

where;

Q = Heat input or output to the control volume. If the control volume rejects heat than the sign of this term will be negative.

W = Work output or work input to the control volume. If the control volume consumes power than this term will get negative sign.

$\sum E_{in}$ = total energy that enters the control volume

$\sum E_{out}$ = total energy that exits the control volume

The efficiency of a power producing system is calculated by dividing the work output by the heat input. Generally, percentage fraction is used to define this efficiency. For example 40 % efficiency means that the engine generates 40 units of work by 100 units of heat input.

The efficiency of a power plant is calculated as follows [65]:

$$\eta = \frac{W_{out}}{Q_{in}} \qquad [3]$$

where;

W_{out} = Net work output of the expanders

Q_{in} = Heat input to the system

2.2. Second Law of Thermodynamics

The first law and efficiency is not sufficient to analyze the performance of an equipment. It does not give any idea about the maximum achievable efficiency that an equipment can perform in certain operating conditions Therefore the Second Law of Thermodynamics is used to find the maximum possible efficiency of any component in a power plant. Once the maximum achievable efficiency is determined, the performance of the equipment can be evaluated by a comparison between the maximum possible efficiency and the actual efficiency of the component [65].

The second law reveals that it is impossible to convert an energy form to another form with 100 % efficiency. For example it is impossible to convert all of the chemical energy of the natural gas to electricity in a combined cycle power plant. Always losses exist in the conversion processes [65].

The second law is very important for power generating industry. It is the main tool to optimize the power plant equipments. It reveals the success of an equipment design. It also shows possible improvement opportunities for the operating components. Besides these, it is also used to monitor the degradation of an aging power plant.

The Ideal Carnot Engine Principle indicates the maximum possible efficiency that a machine can perform which is working between a high temperature reservoir and low temperature reservoir (Figure 3). In fact, no machine can reach the Carnot efficiency. This maximum efficiency is a reference to find out the success of the machine design. The Ideal Carnot Engine efficiency is calculated as follows [65]:

$$\eta_{carnot} = 1 - \frac{T_L}{T_H}; \qquad [4]$$

where;

η_{carnot} : Ideal Carnot Engine Efficiency

T_L : Low temperature reservoir temperature

T_H : High temperature reservoir temperature

The derivation of this equation can be found in reference [65]. The equation indicates that efficiency will increase if either the high temperature reservoir is raised, or the low temperature reservoir is decreased. This is the reason why the gas turbine manufacturers spend great effort to increase the turbine inlet firing temperatures. The condensers are also designed to provide the lowest possible temperature at the low temperature end to optimize the cycle.

The second law efficiency is derived as follows [65]:

$$\eta_{II} = \frac{\eta_{actual}}{\eta_{carnot}} \qquad [5]$$

where;

η_{II} : Second Law Efficiency

η_{actual} : The actual efficiency

η_{carnot} : The Ideal Carnot Engine efficiency

This efficiency reveals the success of the design. As an example; three different machines are considered with 40 % first law efficiency. The low temperature reservoir is the ambient temperature for all of them. The high temperature reservoirs are 1200 K, 1000 K, and 800 K respectively. If these machines are qualified only with the first law efficiency, it can be said that they have the same performance. When the second law is considered, the second law of efficiency levels of the machines will be 53.22 %, 56.98 %, 63.74 % respectively. Therefore, it is clear that the last machine is more successful than the others.

2.3. Rankine and Brayton Cycles

The gas turbine operation is modeled mathematically with Brayton cycle and steam cycle is modeled with Rankine cycle in thermodynamics. The reasons for using these cycles are beyond this study. Such details can be found in reference [65].

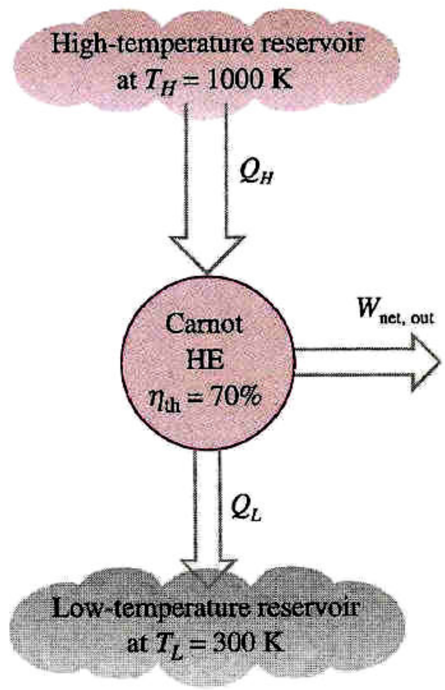

Figure 3: Ideal Carnot Engine [65]

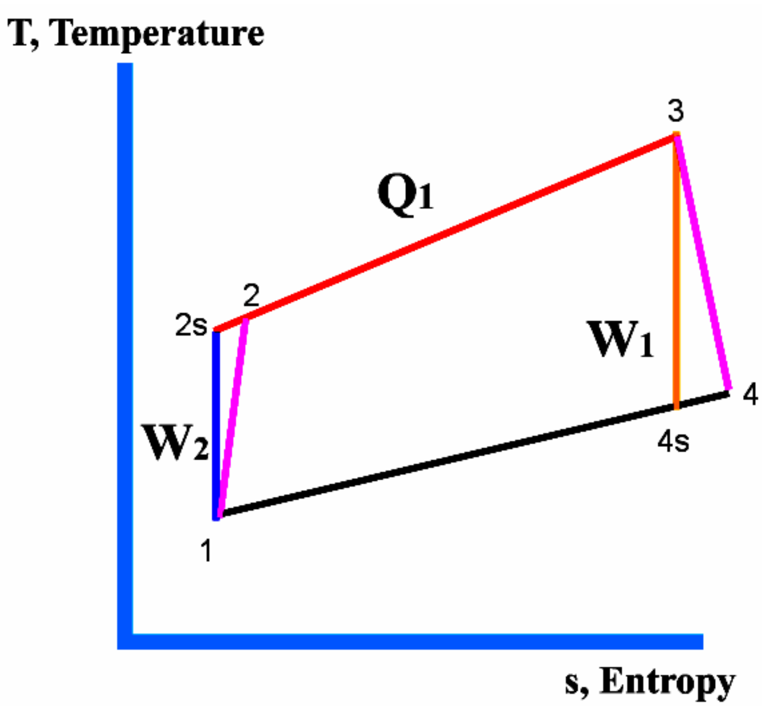

Figure 4: Ideal and Actual Brayton Cycle [65]

The paths of the above graph represents the gas turbine operation. The path between point 1 to 2 and 2s points represent the compression of the air in compressor. Here, 1-2s represents the isentropic compression where 1-2 reveals the actual compression. The compression process consumes power. 2-3 represents the combustion chamber where the heat enters the cycle. 3-4 and 3-4s represents the expansion at the turbine. The expansion of the combustion gases generate work output.

If the compression and expansion lines are connected to 2s and 4s points than it is the ideal Brayton cycle. However, due to various losses like friction, practically such gas turbine operation is impossible. The actual Brayton cycle pass through points 2 and 4. The 2s and 4s points are used to find the isentropic efficiencies of compressor and turbine. These are second law efficiency levels. They are derived as follows [65]:

$$\eta_{ise,comp} = \frac{h_{2s} - h_1}{h_2 - h_1} \qquad [6]$$

where;

$\eta_{ise,comp} =$ isentropic efficiency of compressor

$h_1 =$ enthalpy of air at compressor inlet

$h_2 =$ enthalpy of air at compressor discharge

$h_{2s} =$ enthalpy of air compressor discharge for isentropic compression.

$$\eta_{ise,\exp} = \frac{h_3 - h_4}{h_3 - h_{4s}} \qquad [7]$$

where;

$\eta_{ise,\exp} =$ isentropic efficiency of turbine

$h_3 =$ enthalpy of combustion gas at turbine inlet

$h_4 =$ enthalpy of exhaust gas at turbine outlet

$h_{4s} =$ enthalpy of exhaust gas at turbine outlet for isentropic expansion.

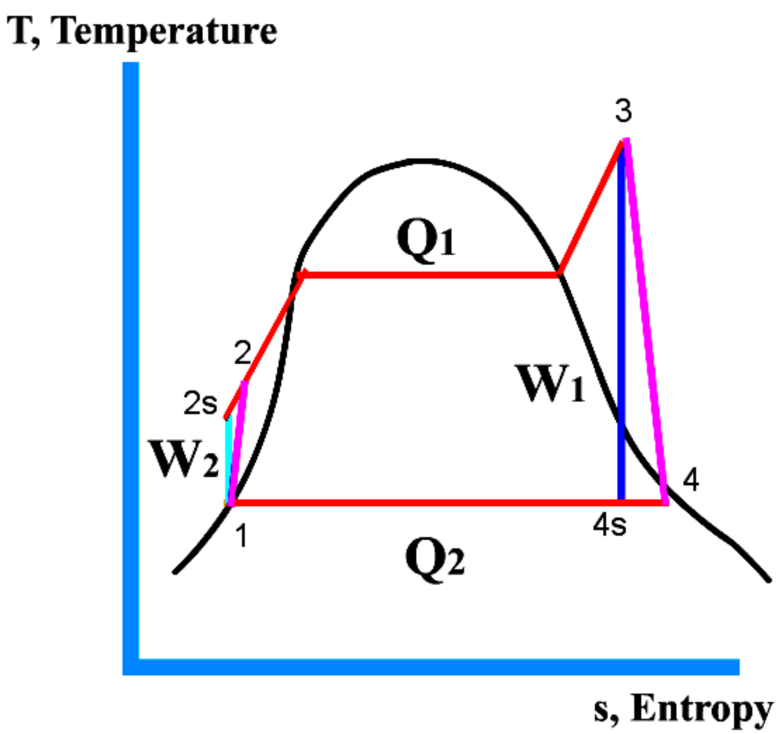

Figure 5: Ideal and Actual Rankine Cycle [65]

As for the Rankine Cycle, the path on the above graph represents the steam cycle operation. Points 1 to 2 and 2s reveals the compression of the condensate water at boiler feed pump. The 1-2s point is the isentropic compression and 1-2 is the actual compression. 2-3 represents the steam generating unit. It could be a boiler, heat recovery steam generator or nuclear reactor according to the type of the power plant. 3-4 and 3-4s represents the expansion of the steam at the steam turbine. 3-4s reveals the isentropic expansion. 4-1 line represent the heat rejection at the condenser.

If the path passes through points 2s and 4s, than the cycle is Ideal Rankine Cycle. However, like Brayton cycle this is an imaginary cycle, and the actual cycle follow the path crossing point 2 and 4. The equations that are used for Brayton cycle are also used for Rankine cycle to calculate the isentropic compression and isentropic expansion efficiencies.

3. PERFORMANCE CONCEPT OF COMBINED CYCLE POWER PLANTS

Performing a combined cycle power plant performance analysis is a very difficult task. Combined cycle power plants are very complex facilities. The operating behavior of the individual components are highly complicated. In order to understand their effects on each other, a mature understanding of applied engineering thermodynamics is needed.

Knowledge on many different engineering disciplines such as thermodynamics, control theory, computer science, heat transfer, etc. are needed to perform performance analyses. In addition, sufficient on-site experience and familiarity with the equipments, and their characteristic behavior are all required. Another difficulty for a performance calculation is that the power plants do not generally have sufficient instrumentation to achieve complete performance analysis. Some parameters like turbine inlet temperature can not be measured directly because of the harsh environment. In addition, some operating parameters like mass flows generally can not be measured accurately. To overcome these problems, a heat and mass balance of the complete power plant should be implemented to acquire the missing measurements. This is also useful to identify and validate the potential incorrect readings [62].

The heat and mass balance calculations can be carried out with classical thermodynamic principles. The first rule of thermodynamics: conservation of the energy and mass principle, chemical balance rules, heat transfer principles and the other necessary equations can help the engineers to calculate the operating parameters. Generally, combined cycle power plants contain adequate instrumentation and measurements to perform a complete heat and mass balance.

For a comprehensive performance analysis, hundreds of heat and mass balance equations should be solved. The most effective and successful way to handle this task is to develop a computer-based model that solves all thermodynamic equations. This will save researchers from repetitive calculations for heat and mass balance. This is a more complete way, and it will save time and prevent human calculation errors. Once the computer based model is created, various performance analyses can be carried out with this master model.

A performance analysis of a CCPP should have the following steps:

First of all, the power plant operation should be adapted properly with a computer-based model using the thermodynamic principles. The actual technical data of the power plant should be used to model the plant. For example, an incorrect modeling of the HRSG layout will lead to incorrect results. After simulating the plant correctly, the model should be tuned with the actual acceptance/commissioning test data. After tuning the overall model of the power plant, various performance studies can be conducted. Such a model can also provide the design performance of the plant.

Once a model with design values is established, another degraded model should be arranged with available measurements under actual running conditions. These steps will provide the missing operating values as well as the current operating performance of the power plant. Once a calibrated degraded plant model is established, design performance and actual performance can be compared. Finally, with the degraded power plant model, rehabilitation options can be analyzed, and economic studies like cost-benefit analyses can be conducted. As a further work an online performance monitoring system can be set by storing the measurements, performance and degradation calculations in a time-logged database.

3.1. Heat and Mass Balance

As mentioned earlier the main purpose of heat and mass balance calculations is to obtain the unmeasured parameters. This additional information is very important for performance monitoring and performance degradation analyses.

Basically the mass and energy balance is stated as follows [64]:

Rate of Storage of Mass = Mass Inflow Rate – Mass Outflow Rate

Rate of Storage of Energy = Energy Inflow Rate – Energy Outflow Rate

Most of the time the power plant equipments operate under steady-state conditions. Therefore, left side of the equations are zero. Generally inflow rates are equal to outflow rates in power plant components.

A control volume must be selected to perform the balance equations. The control volume can be the whole plant or just a pump for a local heat and mass balance. The measured values are used as inputs to find the missing values by applying local heat and mass balances.

Another application of heat and mass balance is validation of the inaccurate measurements. In a power plant there are many instruments and measurement devices. Generally, the temperature devices measure accurately. However, the flow transmitters may not be precise. Heat and mass balance can provide more complete information about the control volume by using the precise measurements like temperature to validate the inaccurate readings. [62]

3.2. Performance Monitoring

Performance Monitoring means continuous tracking of the power plant condition. The main purpose is to follow the performance degradation. Watching the machine health is another intention, which is called “Condition Monitoring”. These monitoring systems are essential tools for the operator to take action before it is too late. It helps the operator to plan the overhauls. It is very important to know in advance that which parts should be replaced. Typically, equipment manufacturers provide some parts one year after the purchase order. Determining the scope of the overhaul is also important to arrange the contractors, and field service personnel. The condition monitoring systems can also prevent the forced outages. These shutdowns cost heavily to the power plant companies because of the unplanned production loss. If a catastrophic failure occurs at the prime movers, the cost can be enormous.

There are traditional techniques for Condition Monitoring that are used in the industry for many years. These are vibration monitoring, (Figure 6) oil quality tests, boroscope, endoscope and visual inspections [66].

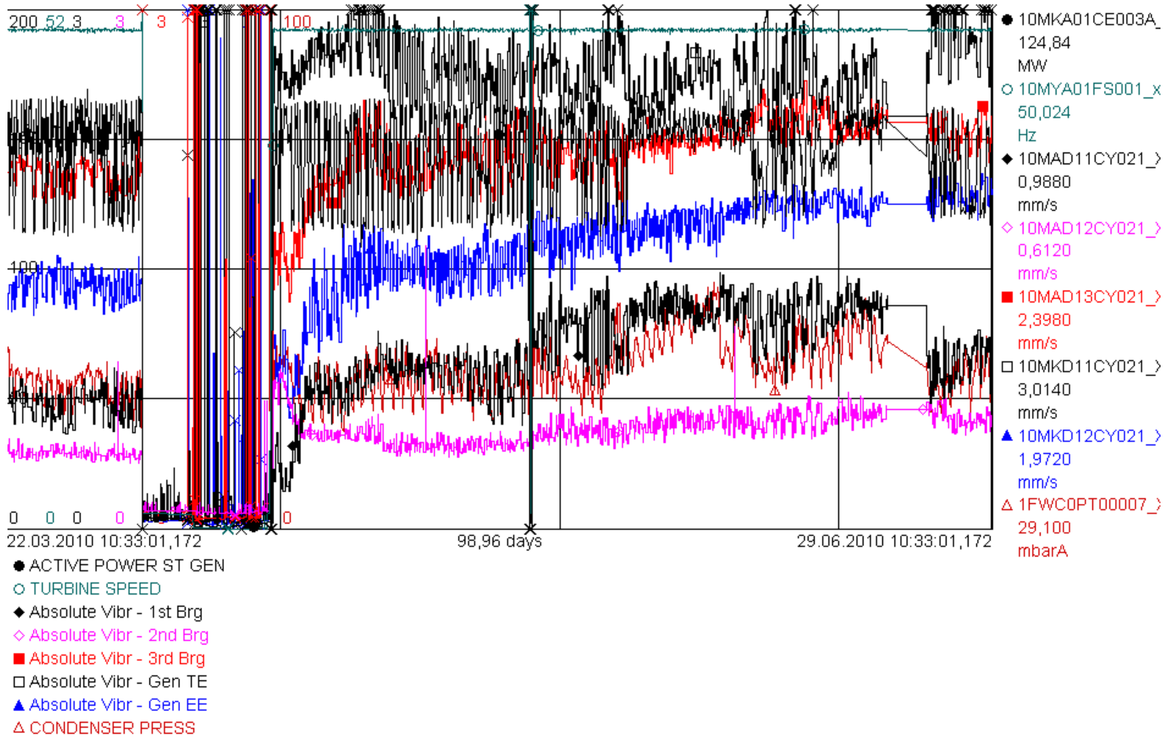

Figure 6: Vibration Monitoring Trends

These techniques can detect the problems. However, for a complete performance and condition monitoring additional systems are required. These performance monitoring systems consist of the following: 1) Data acquisition system that gathers on-site measurements. 2) A computer-based heat and mass balance model that calculate the performance of the plant as well as the missing measurements 3) A time logged data storage. The data loggers have analyzing tools such as trend graphs. By observing these trends, changes in the plant parameters can be detected. The deviation of the operating parameters indicate degradation and/or possible failures. Making correct judgments on these trends (Figure 7) requires talent and experience.

A decrease of the performance can have natural causes. For example a gas turbine power output decreases with the increasing ambient temperature [13]. Discharge temperature at the lower reservoir increases, and the density of the air decreases at high temperatures so that the mass flow will reduce as the volume of the machine is constant.

This is an expected result, it has no relation with a degradation mechanism. Performance engineer should be able to distinguish these facts.

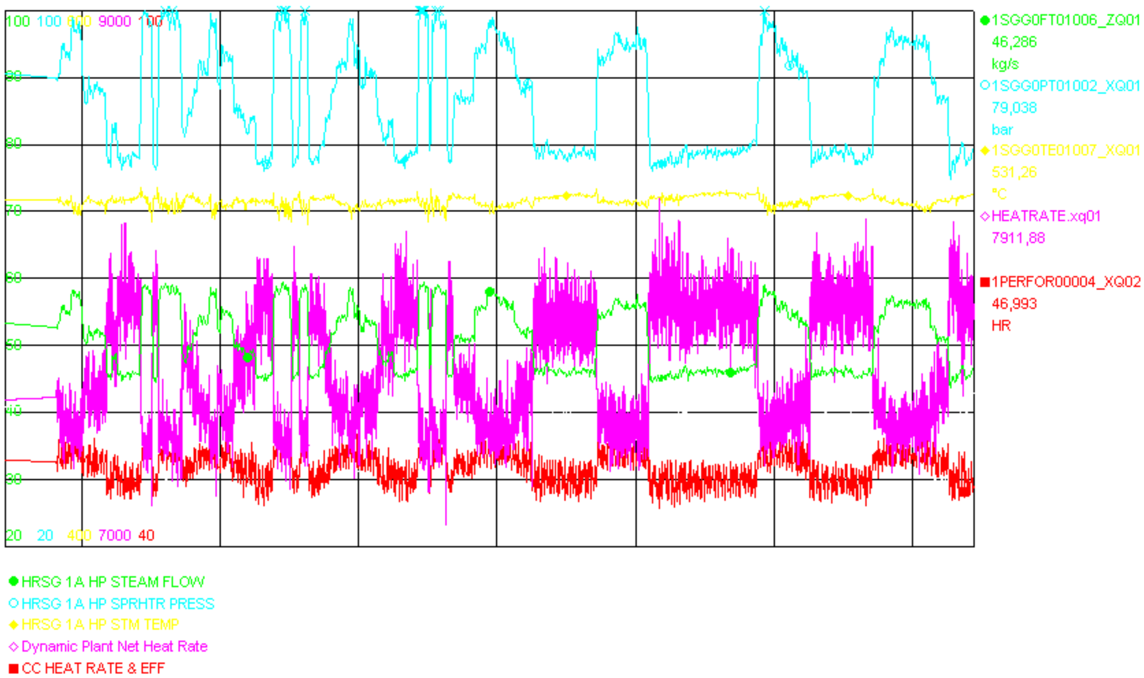

Figure 7: Trends of Selected Operating Parameters on the data logger.

Detecting the absolute values of the power plant performance is not so important for on-line performance monitoring applications. Because, the on-line performance monitoring is generally used to determine the "deviations" of the operating parameters and the performance [62]. In fact, finding the absolute values is very difficult. As there can be imprecise measurements in a power plant.

3.3. Expected-corrected performance

As mentioned earlier the ambient conditions like the ambient temperature, cooling water temperature, etc. have significant effect on the performance parameters. For example, the same gas turbine, which produces 120 MW at 30 °C ambient temperature, may produce over 130 MW at 0°C. (Figure 8) For a fair comparison of the performance of different gas turbines, analysis must be carried out for the same ambient conditions. International Organization for Standardization (ISO) has declared the ISO standard ambient conditions as: Ambient Temperature: 15 °C, Relative Humidity: 60 % and under Ambient Pressure at Sea Level (1.013 bar). Generally, the acceptance tests are performed at these conditions. If the site conditions are different than these during the tests, the results are "corrected" to ISO conditions.

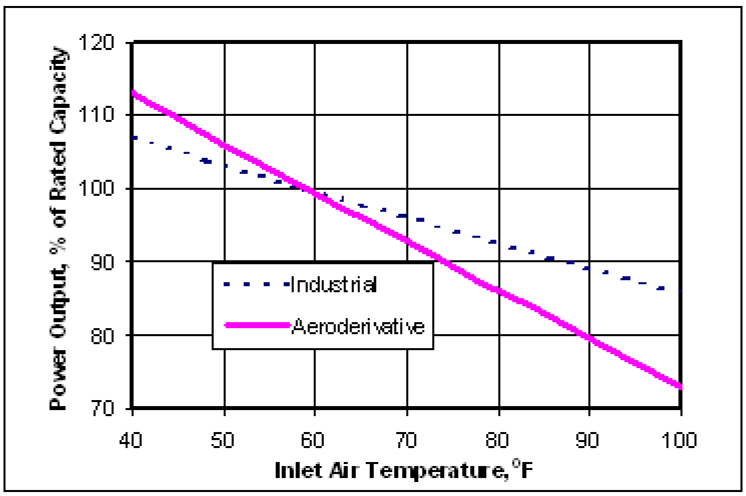

Figure 8: Power Output vs. Inlet Air Temperature
(http://www.turbineinletcooling.org)

The fact that the performance changes due to the ambient conditions is a main concern for performance analyses. It should be determined whether the performance changes occur because of the ambient condition variations or the deteriorations. Powerful and well-established performance monitoring systems have additional tools that can help the operators to make correct judgments. These programs provide "expected and corrected" performance calculations, which really help during degradation observations. Solid understanding of these terms is indispensable to interpret the trends from the performance monitoring system.

The expected performance is the performance that the machine should perform at any ambient condition. For example, it is calculated with a heat and mass balance that a gas turbine which has 100 MW ISO rating can produce 110 MW at 5 °C. Then it can be said that this gas turbine without any deterioration is “expected” to generate 110 MW at 5 °C. If the actual power production of this gas turbine at 5 °C is measured 105 MW then it can be commented that the gas turbine has 5 MW degradation. The deviation of the actual performance from the expected performance indicates the magnitude of the degradations (Figure 9).

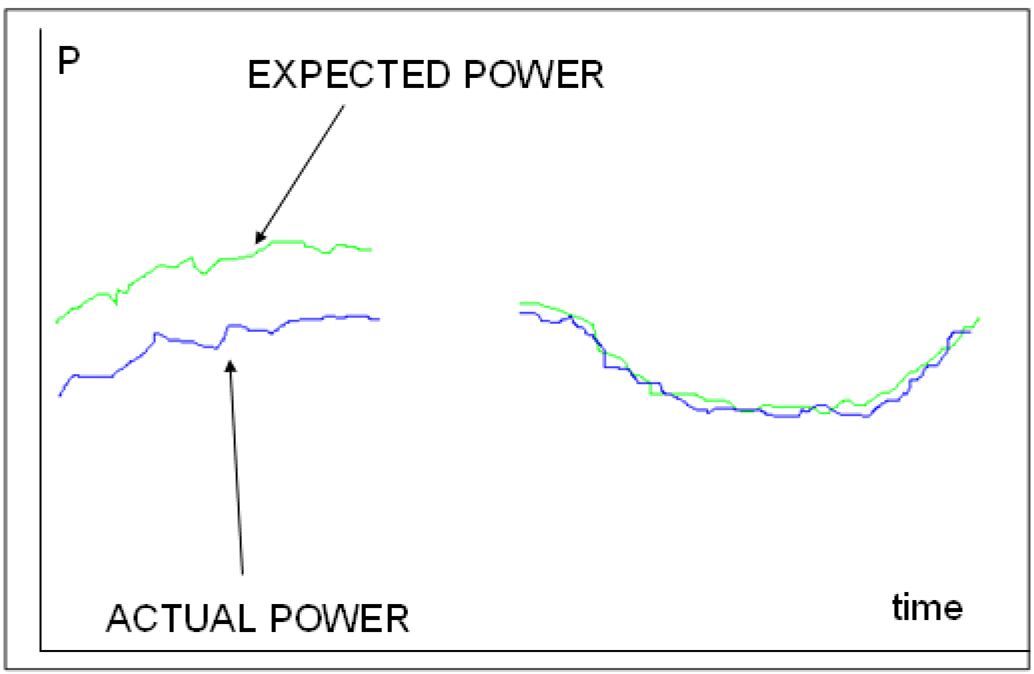

Figure 9: Expected and Actual Power Output Trends of a Gas Turbine

On the above figure the blank section indicates an overhaul. It can be easily seen that before the outage there is power degradation. It is observed that after the maintenance the machine had recovered its performance, and it started to operate as expected. This trend also reveals the success of the revision. Here, it is also noticed that the power output is decreasing with time. This is not due to a deterioration. Actually the increasing temperature of the air in a summer day causes the shortfall of the power output. The machine is expected to generate low power during higher summer temperatures.

The expected values of the other important operating parameters such as condenser pressure, steam shaft output, high pressure steam temperature & pressure, etc can be calculated, and these values can be used in performance analyses.

Because the expected performance of the machines changes continuously according to the ambient conditions, it may be difficult to conduct long term performance surveys by examining the expected performance. To overcome this issue, corrected performance of the machine is calculated [62].

"Correction" concept of the performance is confusing. In fact, to find the corrected performance, first of all, the performance of the machine in the current ambient conditions is determined. After that, it is calculated that; what would be the performance of the machine if it was running in ISO ambient conditions. For example, a new gas turbine generating 120 MW at 5 °C is considered. The calculations reveal that this machine can produce 110 MW at ISO conditions. This is the corrected output. The same gas turbine may degrade and its production may decrease to 115 MW at 5 °C. If the correction calculations say that this degraded machine would generate 107 MW in ISO ambient conditions, then according to corrected performance concept, this machine has 3 MW corrected power degradation. Therefore, in the corrected performance concept, the design output of this turbine is always 110 MW under ISO conditions. When the machine starts to deteriorate, this corrected output will start to decrease. This concept provides more straight forward indication to watch the gas turbine degradation. The correction trend of any equipment is the performance "gauge" of the component. The performance degradations and improvements can be determined on these graphs (Figure 10).

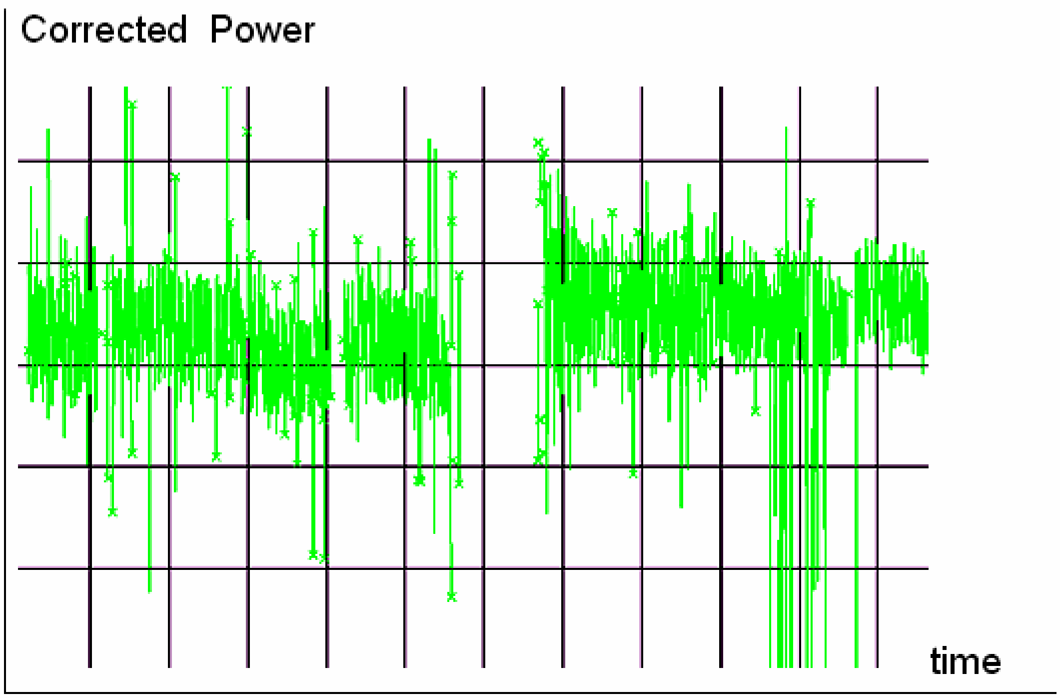

Figure 10: Corrected Power Output Trend of a Gas Turbine

The above figure displays the corrected power output of a gas turbine. It is observed that a gradually degrading gas turbine recovers its performance after an overhaul.

There are several ways to calculate the expected and corrected performance. The vendors generally supply correction curves to the corresponding hardware that shows the deviation of the performance according to the various ambient conditions. The expected and corrected performance can be found using these curves. Another way is to use fundamental thermodynamic principles. The performance of the machines can be calculated by applying the heat and mass balance equations. However, it would be difficult to solve these equations by hand because power plants are complex facilities, and numerous local heat and mass balance equations should be simultaneously solved to find their performance. As a result, the most efficient way is to develop computer based models to conduct performance analyses.

3.4. Power Plant Simulation Models

As discussed earlier, the efficient and functional way to analyze power plant performance is to create a computer based model that solves the heat and mass balance equations. To find correct performance results, the power plant should be modeled properly. A power plant can be simulated correctly only by using the real data. The following information should be available to prepare an accurate model:

1) The flow chart of the Power Plant,
2) The technical drawings that show the layout of the HRSG,
3) Acceptance Test Reports,
4) Gas turbine ISO Ratings,
5) The necessary actual operating parameters, etc.

The first step is to arrange the flow chart of the power plant. The power plant components should be modeled separately. Each component model should have its own code that calculates the local heat and mass balance. Finally, the component models are connected to each other according to the power plant flow chart. After this arrangement, the model should be "tuned" using the acceptance test data to get the system model per design specs of the power plant. This tuning is made by iterating the second law efficiencies such as steam turbine isentropic efficiency.

3.5. Design & Off-design Conditions

The understanding of the design and off-design concept is a must to be able to model, simulate and analyze the power plant performance.

Generally, the power plants are designed according to the ISO ambient conditions. However, the power plants mostly operate beyond the ISO conditions. The air temperature, humidity, sea temperature etc. are always changing. So it can be said that the power plants generally operate at "off-design" conditions.

The computer based power plant models should be able to calculate the performance under different off-design conditions in order to observe the behavior of the power plant in different ambient conditions. The off-design model is also used to examine the part-load performance. Furthermore, the degradation analyses are also carried out with the off-design models.

3.6. Gas turbine Performance

The gas turbine is the prime mover of a combined cycle power plant. It generates electricity by burning fuel. It also provides hot exhaust gas to the heat recovery steam generator where steam is produced for the steam turbine.

Gas turbine (Figure 11) is invented nearly hundred years ago. But its utilization in the power industry is relatively new. Before, it was widely used as engines for the aviation industry. It has become a favorable machine for the energy sector by the development of the advanced materials and cooling systems (Figure 12). These advances provide higher turbine inlet temperatures which means higher efficiency and output. Today, modern gas turbine firing temperatures have reached almost 1500 °C. These high temperatures boost the gas turbine performance and power output while decreasing the combined cycle heat rate.

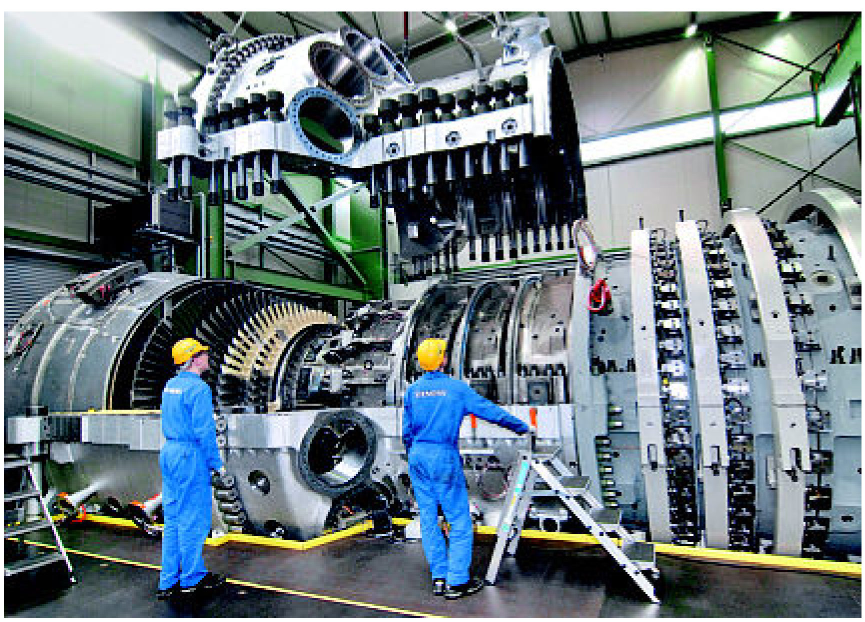

Figure 11: A Heavy Duty Gas Turbine (www.siemens.com)

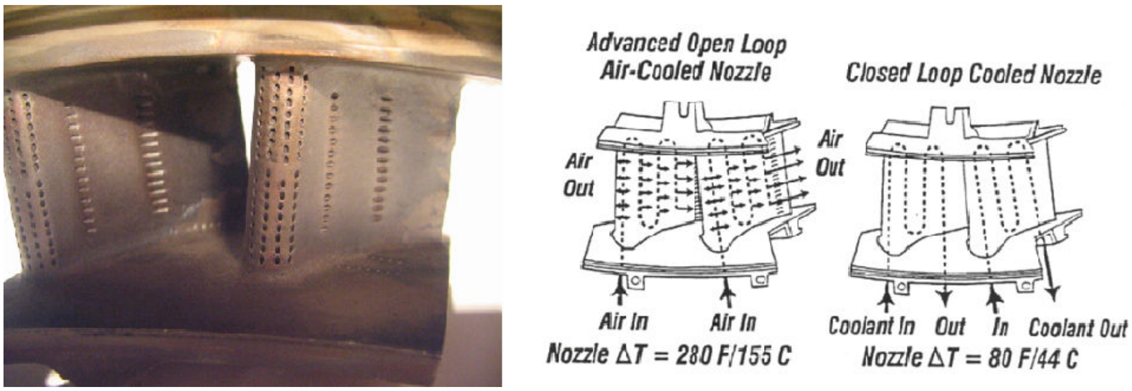

Figure 12: Advanced Cooling Techniques [5]

Today gas turbines are used in various industries for different purposes. Their capacities are varied from 0.05 MW to almost 400 MW. There are several manufacturers that offer industrial turbines, heavy duty gas turbines and aeroderivative gas turbines, which were initially designed as aircraft engines. The leaders in the market are General Electric, Siemens, Mitsubishi and Alstom.

The gas turbines are comprised of three sections. The compressor, the combustion chamber and the turbine (Figure 13). The compressor sucks and pressurizes the air. The pressurized air is mixed with fuel and burned in combustion chamber. After the combustor, the combustion products, which have high enthalpy, expand through the turbine to produce power. The compressor is typically on the same shaft with the turbine. It consumes nearly half of the shaft power output to compress the air [66].

A gas turbine has the following advantages that make it attractive [66]:

1) Very high specific power to weight ratio which is much higher than the other engines. That is why it is used in the aero planes instead of the other engines.

2) It is a very simple machine compared to the other internal combustion machines. As a result its maintenance cost is low.

3) It can start and connect to the electricity grid in minutes while it takes hours for a steam turbine to take load.

4) It can burn different fuels such as natural gas, diesel, gasified coal etc.

5) Its capital cost is low, and it requires minimal installation area.

6) Its construction time is very short compared to the conventional power plants.

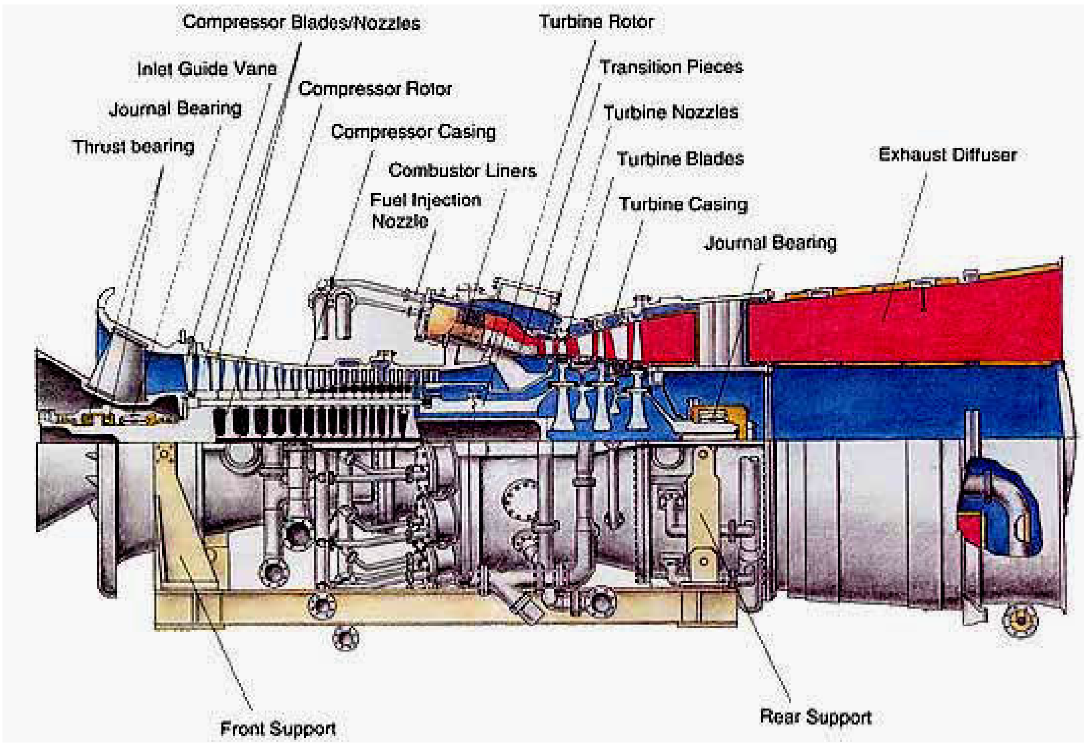

Figure 13: The Components of a Gas Turbine (www.tic.toshiba.com.au)

The gas turbine exhaust gases are generally very hot (up to 600°C) that contain high waste energy. Discharging these gases directly to the atmosphere leads high losses. Therefore, the heat recovery steam generators are designed to utilize this high waste energy to produce steam, which turns another power engine, the steam turbine.

A gas turbine performance is determined by its efficiency and power output. In performance degradation analyses, decrease of these parameters indicate deterioration of the machine. As described earlier, the “corrected efficiency” and “corrected power output” values should be used to track the changes in order to make correct judgments. In addition “expected power output” and “expected efficiency” can be compared with the actual performance to track the degradations and recoveries after the outages [62].

The efficiency of the machine is calculated by the First Law of Thermodynamics as: [65]

$$\eta_{GT} = \frac{\dot{W}_e}{\dot{Q}_{in}} \quad ; \qquad [8]$$

η_{GT} = The efficiency of the Gas Turbine.

$\dot{W}_e$ = Power Generated by the Gas Turbine

$\dot{Q}_{in}$ = Heat Input from the Fuel

The First Law Efficiency can be used effectively in Performance Monitoring and Performance Degradation studies by tracking its changes during time which indicates the deterioration. Calculating its absolute value does not provide much useful information.

The success of a retrofit can also be evaluated by observing the change of the performance parameters. The level of efficiency and work output increase shows the rehabilitation success.

"Heat Rate" is also used widely in energy sector as a first law efficiency parameter. Heat rate is defined as the required heat to produce one kWh electricity. In the literature BTU/kWh, kcal/kWh and kJ/kWh are used as Heat Rate units.

Unlike first law efficiency , absolute value of the Second Law Efficiency of a Gas Turbine is important for performance analysis [65]:

$$\eta_{II,GT} = \frac{\eta_{GT}}{\eta_{carnot}} \qquad [9]$$

where;

$\eta_{II,GT}$ = The Second Law Efficiency of the Gas Turbine

η_{GT} = The Efficiency of the Gas Turbine.

η_{carnot} = The Efficiency of the Carnot machine.

The second law efficiency is especially useful to compare the thermodynamic success of different gas turbine models operating under the same condition. It also reveals the possible achievable gap to improve the efficiency. However, closing this gap is generally challenging. Various constraints such as high temperature material strength and oxidation resistance limit the optimization studies.

The second law efficiency is mainly used at design phase of the machine. It is also widely used in optimization studies. For performance degradation and performance monitoring purposes, tracking the relative change of the first law efficiency is commonly used. On the other hand, the second law efficiency is used to determine the gas turbine component performance. Compressor and turbine efficiencies are determined by calculating the isentropic efficiencies of these components. These parameters can be also used to track the equipment performance degradation. These efficiency levels are used for calibration of the computer-based gas turbine models. Degradation of the machine can be reflected to the gas turbine model by adjusting these parameters.

3.7. Steam turbine Performance

Steam turbine is the other turbo-machine in a combined cycle power plant that generate electricity. It is driven by steam (Figure 14).

A steam turbine is an external combustion machine that is used widely in power industry for many years. No matter how a steam is produced, the steam turbine has the same structure in coal-fired, nuclear, geothermal and combined cycle power plants. Of course, there are some characteristic differences for various power plants.

In a steam turbine, only expansion of the working fluid (steam) takes place. There is no combustion or compression. It is just an expander (Figure 15) Therefore, it is accepted as the simplest machine that converts heat energy to mechanical energy. On the other hand, it is the most efficient machine [67].

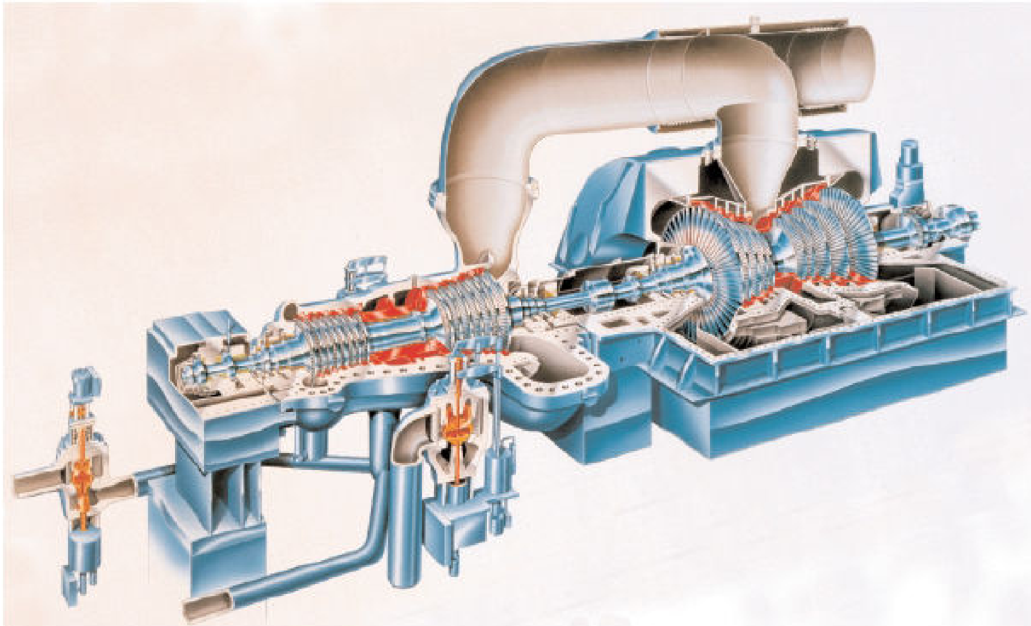

Figure 14: A Steam Turbine [5]

Figure 15: Steam Turbine Rotor (www.arabianoilandgas.com)

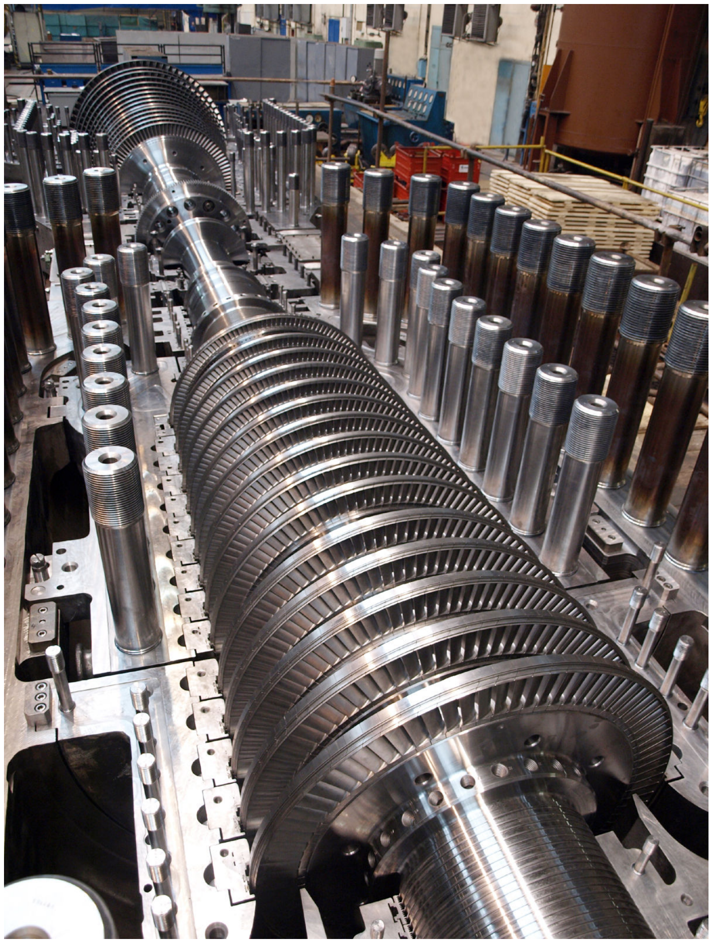

Figure 16: HP and IP Sections of a Steam Turbine (www.skoda.cz)

The steam turbine can have multiple pressure expanders. It can also have more than one casing according to the design (Figure 16).

The steam turbines have two different control methods: Sliding pressure and Throttling pressure. In sliding control, the inlet valves are wide open and the steam inlet pressure is determined by the steam generator. This is the most common control for combined cycle power plants. It is preferred because it provides high quality steam that increase the Rankine efficiency. It also keeps the liquid mass fraction of the steam in acceptable limits at the last stages [62].

In throttling control the inlet pressure is set to a certain pressure which is controlled by a throttling valve.

In a steam turbine, steam enters the turbine, and flows through the stationary guide vanes and the rotating blades. Throughout this process the steam is expanded and its high enthalpy is converted to mechanical shaft power.

The steam turbines are designed to provide maximum possible efficiency by minimizing pressure drops. The guide vanes and the rotating blades are designed to get the optimum steam velocity. Sealing design is another important design consideration to assure high performance.

In performance studies, the steam turbine performance is scored by its corrected shaft output. The first law efficiency is not a performance parameter like gas turbines because steam turbine does not burn fuel.

The change of the corrected power output is observed to analyze the degradation of the steam turbine. An upgrade shows its success by a rise of the corrected power generation. Expected power output and actual power may be compared to observe the degradation and performance improvements after the overhauls (Figure 17).

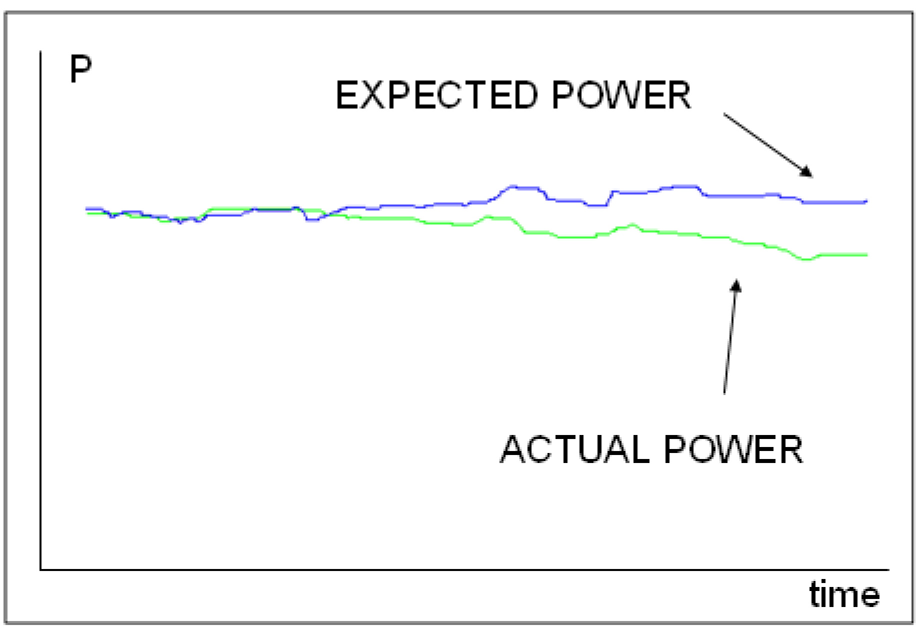

Figure 17: Expected and Actual Power Output of a Steam Turbine

In the above figure, the steam turbine degradation can be observed. The gap between the expected and actual power output increases with time because of the deteriorations.

The second law of thermodynamics is used to calculate the isentropic efficiency of a steam turbine. This parameter indicates the success of the design. It can also be used to track deterioration of the steam turbine. The isentropic expansion is the ideal turbine without leakage, friction, heat loss, mechanical losses, etc. It indicates the maximum possible work output. As mentioned earlier, isentropic efficiency is the ratio of the actual output to the isentropic expansion output.

3.8. HRSG Performance

Heat Recovery Steam Generator (HRSG) consists of various heat exchangers that transfer the gas turbine exhaust waste energy to liquid water in order to convert it to steam to be utilized in a steam turbine. There are different HRSG types. But generally they are classified according to the gas flow. If the exhaust gas flows horizontally through the HRSG, it is called horizontal HRSG. Otherwise the exhaust gas flow is upwards. This type is called vertical HRSG.

The HRSG generally have several pressure levels to optimize the cycle efficiency (Figure 18). The other distinct parameter of the HRSG is the evaporator circulation type, which can be natural or forced circulation. In forced circulation, the water-steam mixture circulation is maintained by a pump. On the other hand, the buoyancy forces drive the circulation in the natural types.

Basically, each HRSG pressure level contains three different heat exchangers (Figure 19). The water is pressurized in boiler feed pump and enters the economizer. In an economizer, the water is heated close to the saturation point without vaporization. Evaporation is avoided in the economizer as the two-phase flow can have destructive effects on the flow path, on tubes, elbows, etc. Thus, the economizer outlet temperature should be lower than the saturation temperature at that pressure. The difference between saturation temperature and the actual outlet temperature is called approach temperature or economizer exit subcooling. In fact, it is one of the most important design parameters of the economizer and the HRSG.

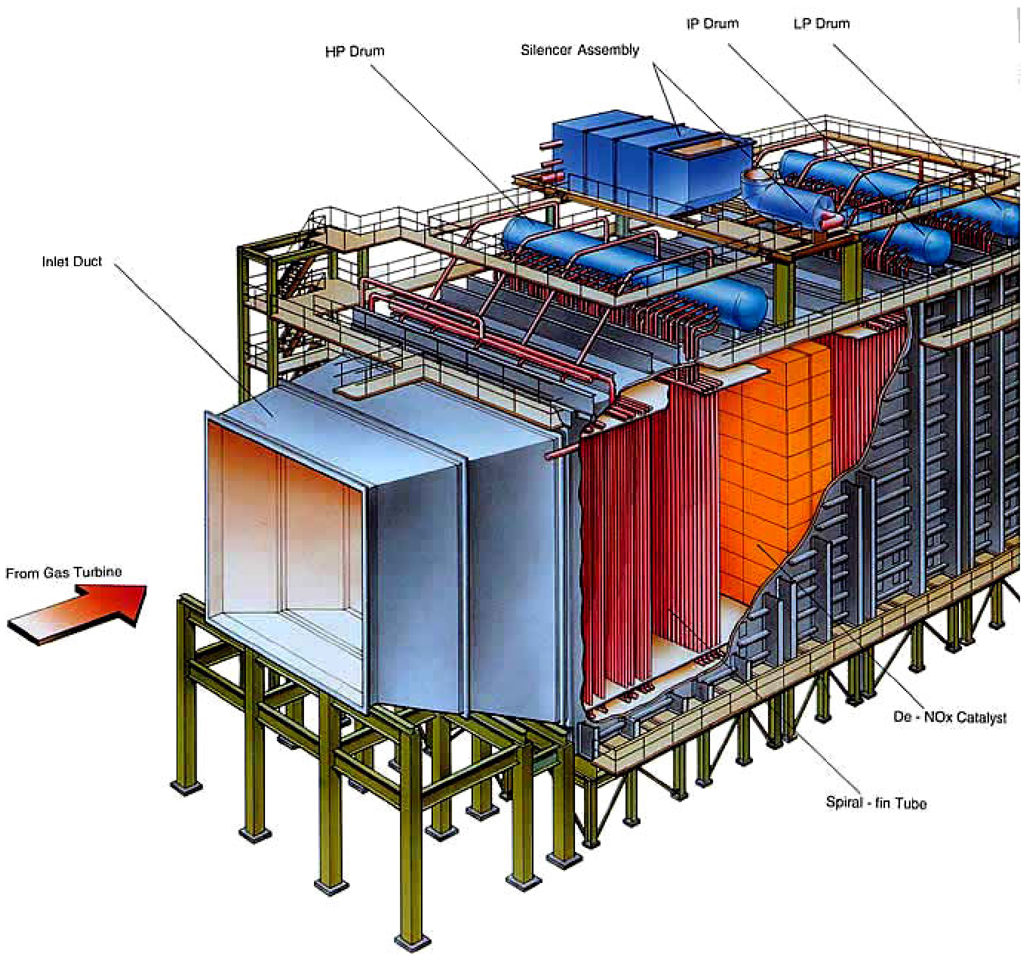

Figure 18: Triple Pressure HRSG (www.tic.toshiba.com.au)

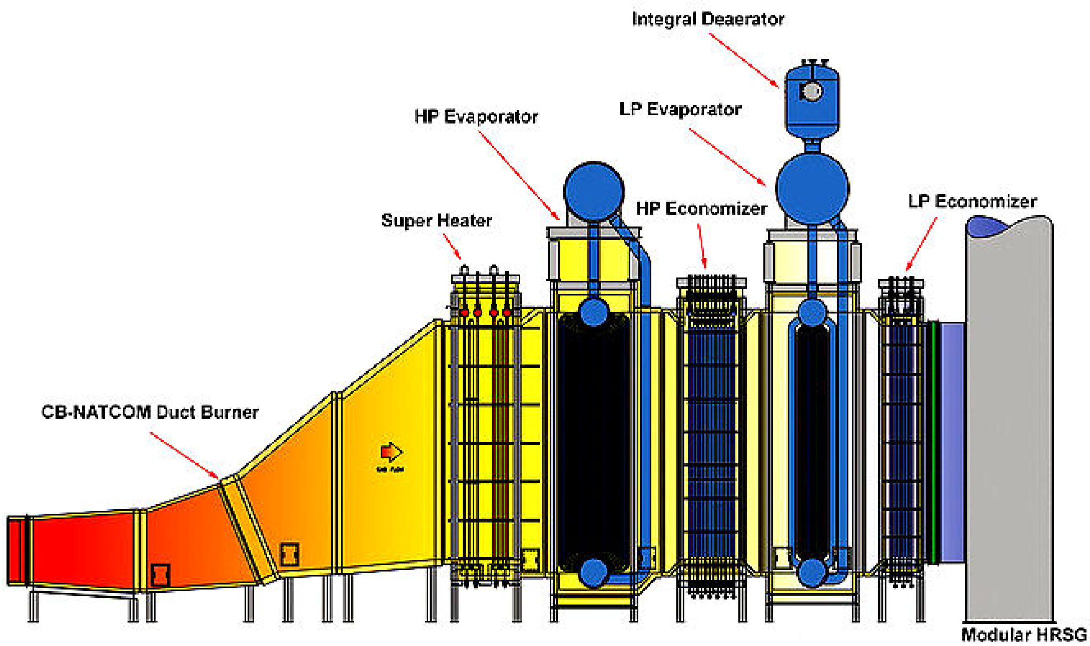

Figure 19: HRSG Sections (en.wikipedia.org)

After the economizer the water pours into the drum. Drum is a pressure vessel where the steam is separated from the water. The water and steam is saturated in the drum. The bottom half contains water. This water flows through the downcomers to the evaporator. In the evaporator the water flows through the risers to return the drum. During this flow a fraction of the water is evaporated. The evaporated saturated steam stays at the top half of the drum. This steam flows to the superheater. There, the saturated steam becomes superheated. After the superheater, the steam goes to the steam turbine.

Another important HRSG design parameter is the pinch point temperature. The pinch is the difference between the exhaust gas temperature leaving the evaporator and the saturation temperature of the water-steam mixture.

The cycle efficiency will improve as far as the approach temperature and the pinch is reduced. To reduce these temperatures the heat transfer areas must be increased. For an infinite surface area heat exchanger these temperatures approach to zero. Therefore, there is a trade-off here. The increase of the heat transfer typically means additional heat transfer area and consequently additional cost [68] (Figure 20). The efficiency gain must justify the cost.

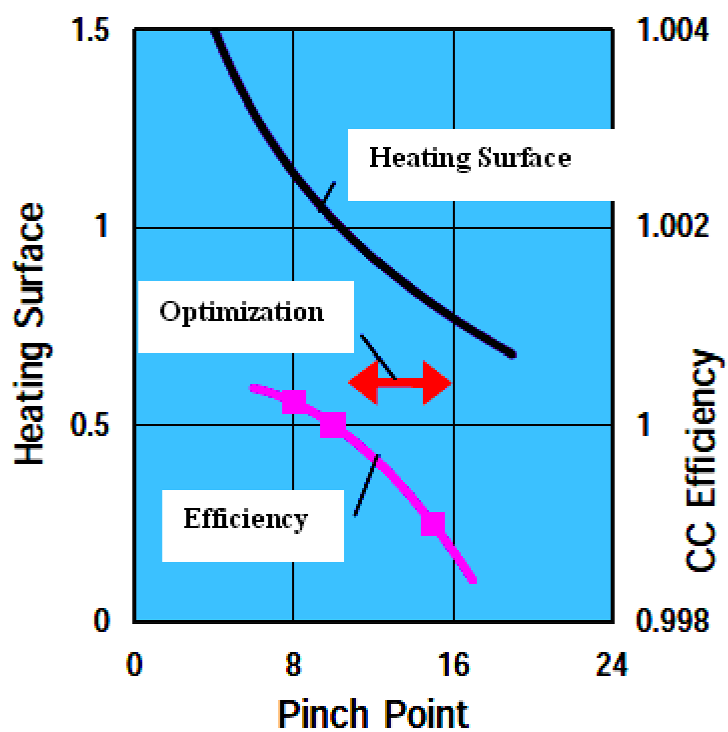

Figure 20: Heat Transfer Surface vs. Pinch Point [68]

The HRSG design process does not involve only the thermodynamic optimization. The design involving complex heat transfer and two-phase flow is rather challenging. For example, the height of the HRSG is designed so that a natural circulation can occur in evaporators. In fact there are many variables that should be considered during the design phase to avoid operational problems.

In some plants, reheat is applied to enhance the efficiency. The triple pressure with reheat offers the most efficient configuration. However, it is also the most expensive and complex one.

A modern HRSG can produce more than 100 bars steam at 560 °C . Actually, the steam operating constraints are determined by the steam turbine material limits.

The water that is utilized in HRSG must be purified with defined properties to keep the HRSG healthy. Otherwise, the HRSG can be affected by different degradation and failure mechanisms. Cycling operation also causes mechanical and thermal stresses on HRSG that shortens its life.

Most of the HRSG units are designed as unfired. However, there is also supplementary firing option. The tubes are generally finned. It is a challenge to provide minimum gas side pressure drop. If the pressure drop increases, backpressure of the gas turbine will increase reducing gas turbine performance.

Effectiveness and efficiency are used as performance indicators to analyze the performance of the HRSG and its components.

The efficiency is the ratio of the absorbed energy of the water to the input energy of the exhaust gas. The First Law of thermodynamics applies. The second law efficiency reveals effectiveness of the heat exchanger. It is calculated by dividing the actual heat transfer to the maximum possible heat transfer. The effectiveness of a HRSG component is calculated as follows [62]:

$$\text{Effectiveness} = \frac{(h_{gas,in} - h_{gas,out})}{(h_{gas,in} - h_{gas,\min})} \qquad [10]$$

where;

$h_{gas,in}$ = enthalpy of the inlet exhaust gas

$h_{gas,out}$ = enthalpy of the outlet exhaust gas

$h_{gas,\min}$ = enthalpy of the exhaust gas at the minimum possible gas outlet temperature

The decrease in the effectiveness reveals deterioration of the heat exchanger. Therefore, it is used in performance degradation analyses.

Minimum possible gas outlet temperature is the temperature that would be the outlet temperature of the exhaust gas if the exit subcooling and pinch point temperatures were zero [62].

Overall HRSG efficiency and effectiveness calculation is not a meaningful way to analyze the HRSG performance. The HRSG components should be evaluated separately to make true judgments. If an HRSG unit is selected as the control volume, the performance calculations will deal with the input and output flows. However, if the high pressure side is degraded, the heat transfer will move to the low pressure side. Therefore, the total heat transfer may not change.

3.9. Condenser Performance

The bottom pressure line of the Rankine cycle takes place in condensers. The low pressure exhaust steam enters the condenser where the steam rejects its heat to another cooling media such as sea water, cooling water or air according to the design and power plant location. In the condenser, the cooling fluid flows through thousands of thin tubes and the steam flows vertically through outside of these tubes. At the end, the condensed water fall down to the hotwell which is at the bottom of the condenser. Finally, the water in the hotwell is pumped again to the steam generator. This concept is same for all kind of power plants that include a condensing steam turbine (Figure 21).

A condenser is one of the key elements in a power plant for the performance. The reason lies on the Rankine cycle graph. Both shaft output and the efficiency will improve if the low pressure line of the cycle decrease as much as possible. However, it has a limit because the last stage exit can have supersonic (choked) flow. If a supersonic flow starts, the decrease of the backpressure in the condenser will not affect the pressure at the exhaust. This means that after a point, decreasing the condenser pressure will not increase the output [54]. In fact, after choked flow conditions occur, decreasing the condenser pressure further reduces the efficiency. This is because when the condenser pressure decreases, the condensate water temperature will also decrease as it is saturated. As a result, the condensate water will go to the HRSG unit cooler. Therefore, HRSG will extract more exhaust gas heat to the condensate water which reduces the efficiency.

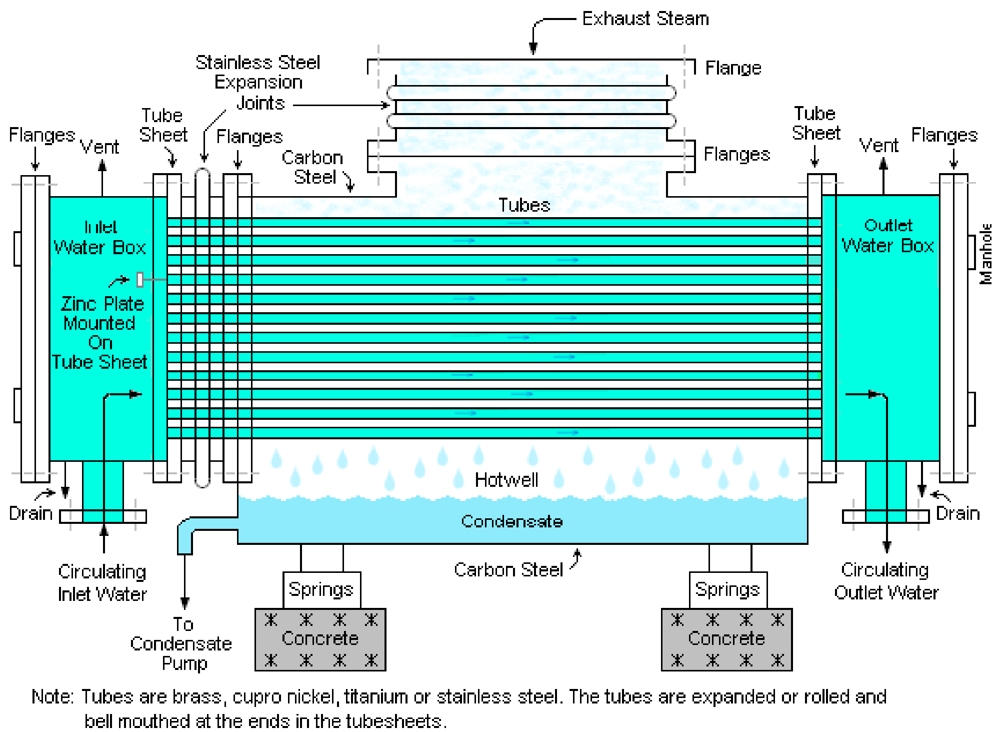

Figure 21: A Condenser (medlibrary.org)

There are three types of condensers: Direct contact condensers which spray water directly into the steam flow, air-cooled condensers and water cooled surface condensers where steam flows over tubes containing cooling water.

The condenser performance is determined by the condenser shell pressure. The decrease in the corrected condenser pressure reveals the performance degradation of the condenser.

As a heat exchanger, effectiveness is also used as a performance indicator for condensers. It is calculated as the same manner as the HRSG components.

3.10. Other equipments

In addition to the major equipments mentioned in the previous sections, there are many other auxiliary systems and components in a power plant. These equipments connect main equipments and complete the thermodynamic cycle. These equipments include heat exchangers, feed water heaters, pumps and valves. These equipments have minor effects on performance compared to the main components. However, in some occasions they can make significant effects on the performance. For example, a leakage in a high pressure condenser by-pass valve can cause shaft output reductions that can reach MWs. Inefficient pumps can also bring additional losses.

3.11. Overall Combined Cycle Power Plant Performance

The main performance parameters of a Combined Cycle Power Plant are the efficiency and total power generation. The performance degradation of a power plant can be identified by the shortfall in these parameters [62].

$$\eta_{CCPP} = \frac{\dot{W}_e}{\dot{Q}_{in}} \quad ; \qquad [11]$$

η_{CCPP} = The efficiency of the Combined Cycle Power Plant

$\dot{W}_e$ = Power Generated by the Combined Cycle Power Plant

$\dot{Q}_{in}$ = Heat Input from the Fuel

The changes of the corrected power output and corrected efficiency are used to find the performance deterioration magnitude. These parameters are also used to see the effects of the rehabilitation projects. A comparison of the expected and current performance can be made as well to determine the performance deviations.

4. COMBINED CYCLE POWER PLANT DEGRADATION

Degradation is the decrease of the equipment performance which is caused by destructive mechanisms such as erosion, corrosion, fatigue, creep, wear, fouling, scaling, etc. These mechanisms appear as a result of different factors such as the transient operation conditions during start-ups, shutdowns and cycling operations. Even in normal operation, the power plant components run under extremely aggressive conditions which cause equipment aging. The contaminants in the fuel and the particles in the air entering the machine also cause degradation of the power plants [69] (Figure 22). All equipments in a power plant suffer degradation in their performance during their service life

Figure 22: A Degraded Blade

Power plant performance degradation is a major problem for the power plant industry as it has an adverse impact on plant output and efficiency. The shortfall of the power output reduces the revenues of the power plant. Degradation also increases the fuel consumption and consequently, it increases the operation costs. It also brings significant environmental impacts. Power plant operators undertake large investments to overcome degradations.

Degradation of a power plant starts as soon as the plant starts commercial operation. Degradation occurs in all parts of the plant, but the focus is especially on gas and steam turbine degradation. Performance recovery efforts are more complicated and expensive on turbines than the other parts of the plant. Performance recovery potential with turbine retrofits is also higher than the other equipments.

There are two main types of degradation: recoverable degradation and the non-recoverable degradation [70].

Recoverable degradation is the formation of deposits, scaling and clogging on the working surfaces of the equipments which are in direct contact with the working fluids. This type of degradation can be recovered with appropriate cleaning techniques.

The other type is the non-recoverable degradation This degradation happens because of the destructive mechanisms like wear & tear, creep, fatigue, aging, loss of working surface, corrosion, oxidation, erosion and other damaging mechanisms. This type of degradation can only be recovered by repair or replacement of the components during overhauls. Examples of non-recoverable degradations are increase in blade tip clearances, sealing leakage of the steam turbine, and combustion system component damage that causes flame instabilities.

Contaminants that degrade the power plant enter the system with air, fuel and water. Air filtration systems, fuel scrubbers, filters and water treatment facilities are used to reduce their impact as much as possible. Microscopic particles and dust is the main air-borne contaminants. Sodium, potassium calcium, lead, sulfur are the typical fuel contaminants. Sodium, potassium calcium, silica, carbonate, sulfate, chloride are the contaminants that come into the plant with water [70].

The main power plant components like turbine and steam generator components are exposed to extremely harsh operating conditions like high temperatures, supercritical pressures and supersonic-speed flows. These conditions cause degradation and aging of the components [59].

The power plants overall performance decreases continuously because of the degradation of the equipments. It is very important for the operator to identify the magnitude and the locations of this degradation to carry out necessary actions.

It is an important task for researchers to find out how each components degradation effect the overall performance of the power plant. This information can be used while deciding which equipment should be overhauled, replaced or upgraded. In Figure 23 different equipment degradation effects on a combined cycle power plant are displayed [42].

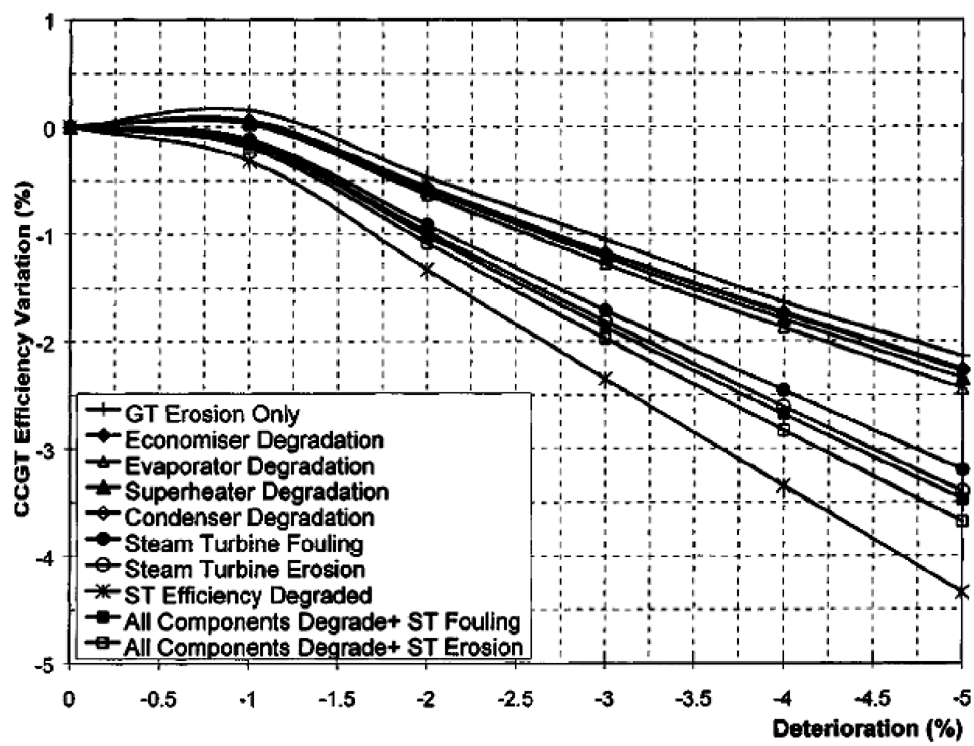

Figure 23: CCPP Power Variation with Gas and Steam Cycles Component Degradation [42]

In Figure 24, a typical degradation behavior of a combined cycle power plant can be seen. The sudden increase of the output and the sudden fall of the heat rate reveals the effect of the overhauls. It is observed that there is a continuous aging. Therefore, maintenance activities can not recover completely the effects of the degradation. A characteristic aging curve occurs for the combined cycle power plants. A complete recovery can only be achieved with life extension or upgrade activities [60].

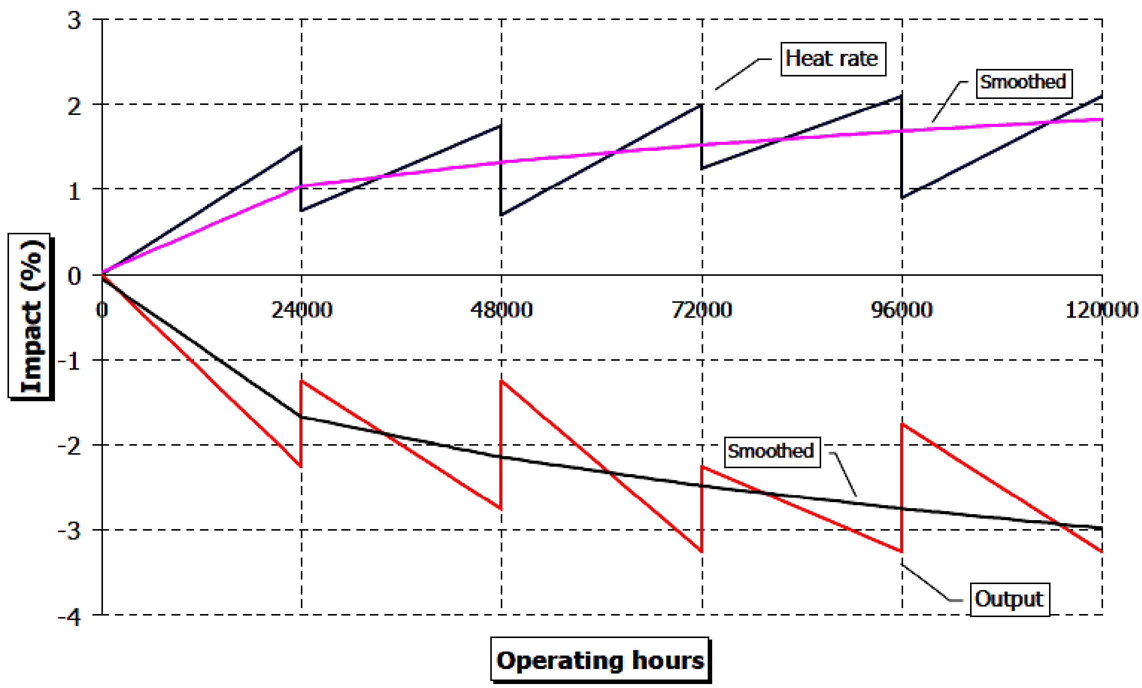

Figure 24: Representative Impact of Aging on CCPP Heat Rate and Capacity (www.ema.gov.sg)

4.1. Gas Turbine Degradation

The gas turbine components, especially the hot gas path components run under very harsh operating conditions. The temperatures can reach up to 1500 °C. There is also high speed flows (supersonic in some components) inside the machine. Because of these effects, gas turbines degrade during operation. Contaminants in air and fuel also degrade the machine.

Gas turbine degradation causes flow path deteriorations, which induce flow reductions or flow increases that cause flow instabilities. These reduce the component efficiencies. As a result, the gas turbine power and efficiency decreases.

Degradation occurs as a result of many different mechanisms. These are: fouling, erosion, corrosion/oxidation, foreign object damage, worn bearings, rubbed seals, blade tip clearances that are out of tolerances, burned or dirty vanes or blades, partially or wholly missing blades or vanes, clogged fuel nozzles, etc. [43].

Compressor fouling (Figure 25) is a recoverable degradation of the gas turbine. It can be recovered by cleaning the compressor blades with on-line water washing, or washing them with brushes during off-line washing shutdowns [71].

Figure 25: Fouled Compressor

The most common non-recoverable losses occur due to the increased turbine and compressor blade tip clearances, increased sealing gaps, damage on the flow path surfaces. These deteriorations can only be recovered by refurbishments and life extension procedures that are applied during the overhauls.

In the open literature, general degradation principles and their effects on the performance do not exist because degradation is actually engine specific. The operating temperatures, pressures and the other operating parameters change differently for different designs of various vendors.

Efficiency reduction of the turbine section affects the performance more than the compressor side. However, compressor deterioration is regarded as a greater performance problem because the compressor degrades much more easily than the turbine section. Compressor degradation occurs as fouling, increased tip clearance, erosion and corrosion on the blades. These non-recoverable degradation mechanisms reduce the compressor section efficiency and mass flow capability. Compressors can easily foul as soon as the engine starts to operate. Fouling is caused by dust particles in the air. The dust deposits adhere on the compressor rotating and stationary blades. Humid air and oil leak from the bearings enhance the fouling.

Fouling degradation increases the blade surface roughness. It also changes the blade profiles. As a result, it reduces both the efficiency and the mass flow capacity of the compressor. In combined cycle power plants, generally constant speed gas turbines are used. The fouling is a bigger problem for these engines as the compressor can not raise its speed to compensate the lost flow. Reduction of the compressor efficiency increase the compressor input shaft power that reduce the unit output and efficiency. The fouling indicates itself with a drop at the compressor discharge pressure.

The reduction of the mass flow causes lower air mass flow to the combustion. It decreases the compressor discharge pressure that means lower pressure ratio and consequently a lower turbine expansion. As the combustion gas mass flow is reduced, this also decreases the gas turbine output power and efficiency. Extreme compressor fouling can decrease the gas turbine output up to 20 %. In combined cycle process, the reduced mass flow also decreases the steam cycle efficiency [72].

Air filters are used to clean the intake air from contaminants because of the significant effect of the compressor fouling. On-line and off-line washings are also applied in appropriate intervals. These are very important to maintain the performance of the machine as much as possible. However sometimes, during on-line washing processes the removed contaminants can clog the cooling channels. This can bring hot corrosion on turbine blades.

The seals prevent the leakage of the working fluids between the rotating and stationary parts. These leakage flows cause decrease on the performance of the machine because the leaked fluid does not do any useful work. Parasitic leakage also distorts the air flow through the compressor stages. The sealing components can be easily worn because of rubbing, The rubbing occurs because of the relative movement of the rotor to the casing. The rubbing generally happens during start-ups and shutdowns. Part-load operation could be another reason. During overhauls correct alignment of the rotor with the casing is very important to prevent rubbing. Nowadays advanced sealing technologies like abradable coatings and brush seals are used in modern gas turbines to overcome performance losses due to rubbing.

Abrasive particles like hard sand and dust cause erosion (Figure 26). All of the component surfaces that face flow can erode because of these particles. The flow speed in the machine is very high. Therefore, a tiny particle can hit on the blades with high energy and cut away small metal particles. Industry experience reveals that particles with 20 μm diameter and above can cause erosion. Blade erosion increases the surface roughness. The turbine nozzles operates near or at the choked flow so they are very sensitive to the changes in the flow area. Increased surface roughness causes thicker boundary layers on the blades and sidewalls, near choking flows. This reduce the flow capacity. Over time blade profiles can change because of excessive material loss as well. This will reduce the aerodynamic characteristics of the blades.

Figure 26: Eroded Coatings of the First Stage Nozzle

On the other hand, if erosion causes material loss especially on nozzles, it will introduce flow increase. This disturbs all of the compressor and turbine flows which leads to yet another performance loss. As an extreme case, the erosion can cause a failure of a blade that will lead to a catastrophic damage to the turbine. If just one blade breaks, it will damage all of the rotating and stationary blades including the other parts in the flow path (Figure 27).

Figure 27: Catastrophic Failure of a Gas Turbine

Compressor blades can have corrosion that increases the surface roughness with decrease in the aerodynamic performance. Sea salt, acids and other corrosive materials which enter into the machine with air can cause corrosion.

Fouling of the turbine section of the machine is not a big concern if it burns natural gas. It contains low contaminants compared to the liquid fuels.

The turbine blades face hot gas for long periods. Their surface roughness may increase with time. The corrosive metals like sodium, potassium, vanadium and lead can react with sulfur and oxygen that deposit on the hot gas path components. Dust and other abrasive particles also erode the turbine blades [73].

The combustion system degradations generally do not decrease the performance significantly. However, their degradation can disturb the temperature profile. This can cause control problems. Generally, the turbine control systems use turbine exit temperatures to control the unit. A set of thermocouples are implemented annularly at the turbine exhaust. Gas turbine control systems use the average of their measurements as a control parameter. If the exhaust temperature is measured incorrectly, it can cause over firing or under firing. Both of these decrease efficiency. Variation of the temperature profile can also damage the turbine blades.

There are blow-off valves in gas turbines to prevent surge during start-ups. If these valves leak during normal operation, it has adverse effect on the performance.

The air filters foul with time that decreases the performance. This degradation is recovered by changing the filters.

The flow in the turbine is generally near Mach numbers. The Mach numbers become higher at low temperatures. This makes the compressor efficiency more sensitive to deviations of the mass flow capacity. Therefore, the effect of degradation gets more significant at low temperatures.

Compressor degradation has more adverse effects on the power output than the heat rate, because generally the turbine runs according to a fixed rotor inlet temperature or constant exhaust temperature. As a result, if the air mass flow reduces, the turbine control system also reduces the fuel flow to fix the control temperatures [43].

The turbine blades operate under high temperatures that are above 1000 °C. There are advanced cooling technologies to protect these components from the extreme temperatures to provide a long operating life. Protective coatings are used for this purpose. Destructive wear and tear mechanisms damage these protective coatings (Figure 28).

Figure 28: Worn Turbine Vane Coating

Another deterioration mechanism is foreign object damage (FOD). Ice particles, loose bolts, forgotten tools after the overhauls that hit to the blades or vanes can make this damage [66].

In Figures 29-33, the effects of various Gas Turbine degradation mechanisms are displayed [12,70].

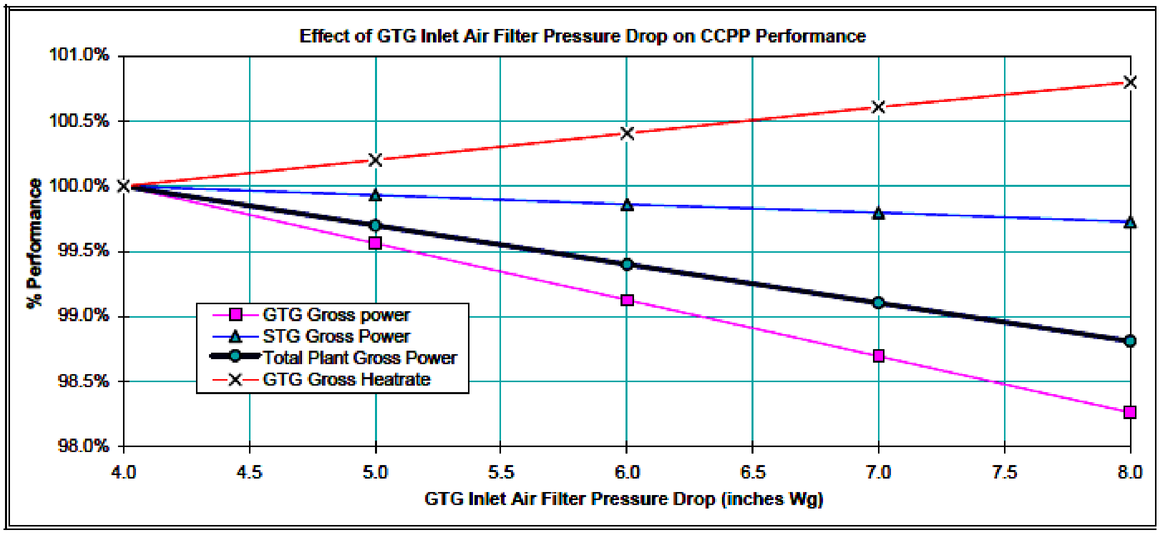

Figure 29: Effect of Inlet Air Filter Pressure Drop on CCPP Performance [70]

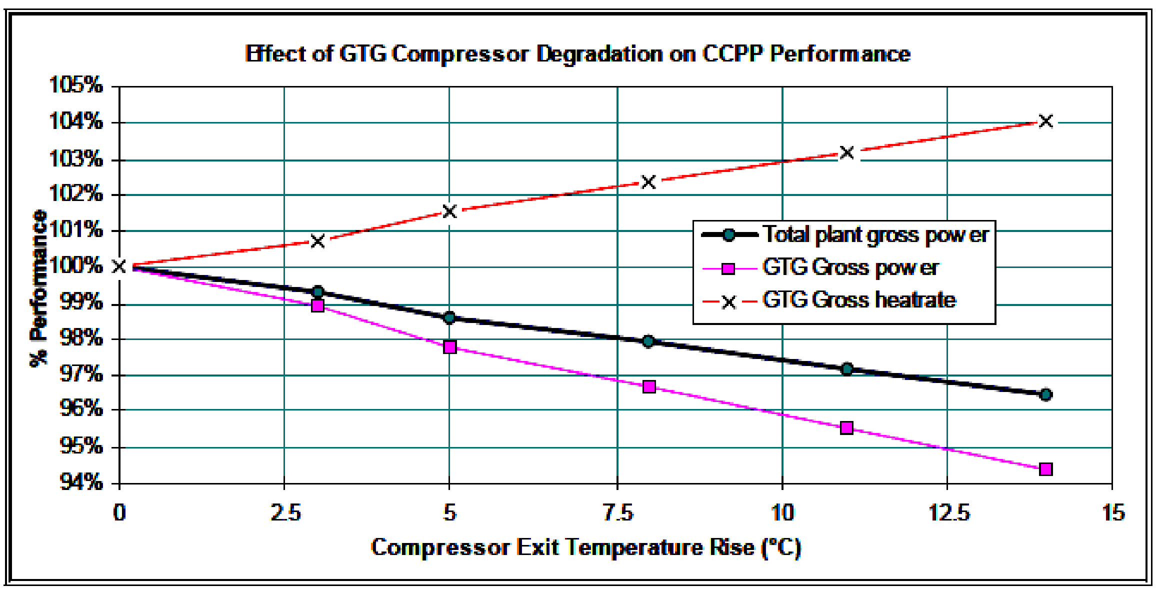

Figure 30: Effect of Compressor Degradation on CCPP Performance [70]

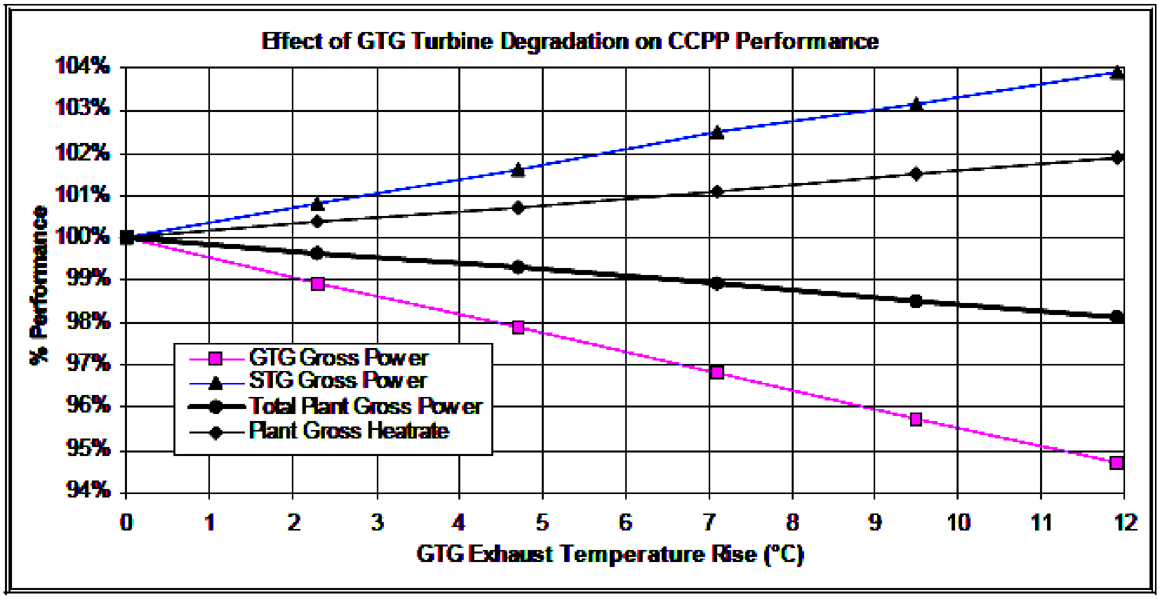

Figure 31: Effect of Turbine Degradation on CCPP Performance [70]

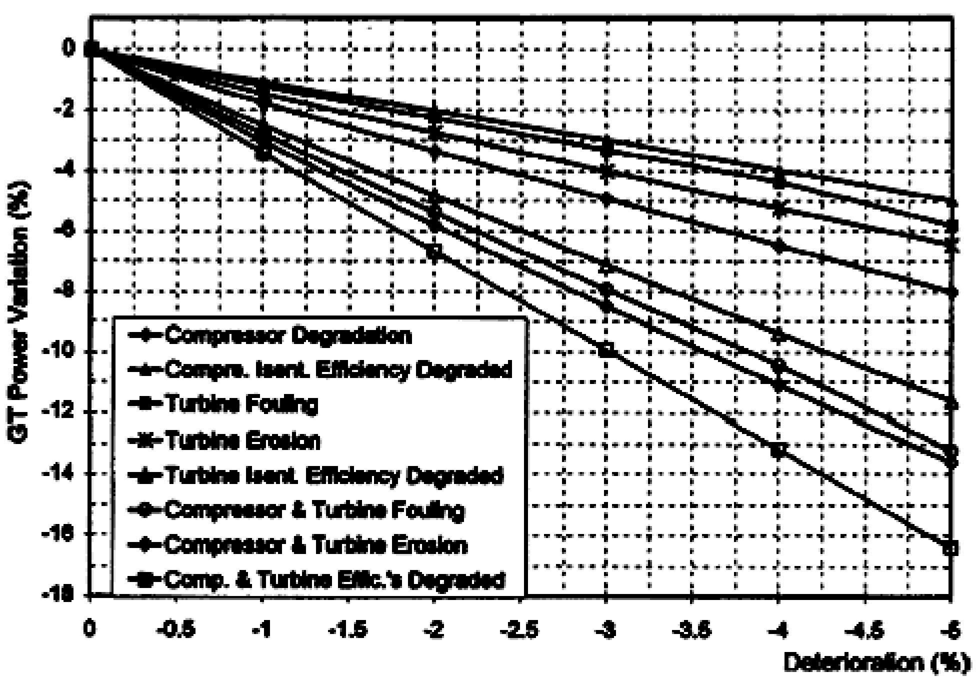

Figure 32: Gas Turbine Power Variation with Component Deterioration [12]

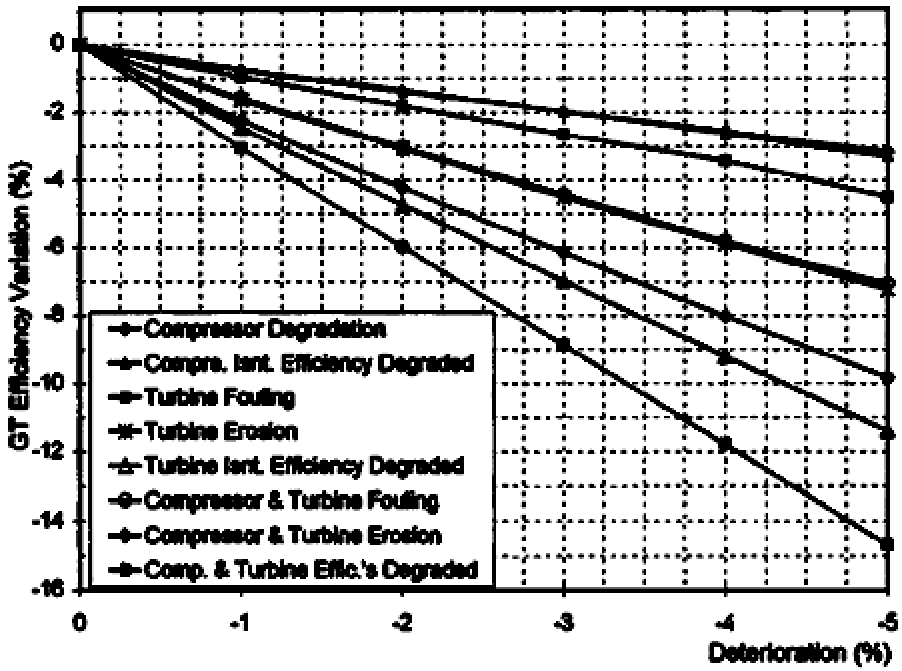

Figure 33: Gas Turbine Variation with Component Deterioration [12]

4.2. Steam Turbine Degradation

Steam turbine degradation (Figure 34) is not a big concern like degradation in gas turbines, because steam turbines are not as complex machines as gas turbines. Their operating environments are not so severe like the gas turbines either. Another reason is that the working fluid of the steam turbine is purified water, which contains negligible contaminants with good water treatment systems.

The main degradation mechanisms for steam turbines are deposition and surface roughness on the flow path, sealing leakage, erosion and internal leakage. The most important problem is the sealing leakages [74].

Figure 34: Degraded HP&IP Sections of a Steam Turbine

Deposition is a consequence of the carry-over mechanism. Carry over means the undesired transportation of materials with steam from the steam generator to the turbine. Typical deposits are silica, copper oxides, chlorides, iron oxides, sulfates, organic or inorganic acids. These materials deposit on surfaces of different sections of the turbine. For example, copper deposits accumulate at the High Pressure Throttle and decrease the flow area. Silica generally deposits on the blades at the low pressure side [67]. The main adverse impacts of the deposits are: raising surface roughness, changing blade profiles and reducing flow capacity by restricting the steam path [75].

Some of these deposits can be cleaned with water if they are water soluble. The others can only be cleaned with chemical treatment or mechanical cleaning. However, the turbine should be opened to be able carry out this cleaning.

Steam turbines can experience foreign object damage like gas turbines which degrades the turbine. The source of the foreign objects is again the carry over from the heat recovery steam generators. Welding leftovers can also cause these damages.

The carry-overs also cause solid particle erosion at inlet stages. The steam turbine erosion (Figure 35) increases the mass flow capacity at the turbine inlet and causes a reduction in turbine isentropic efficiency.

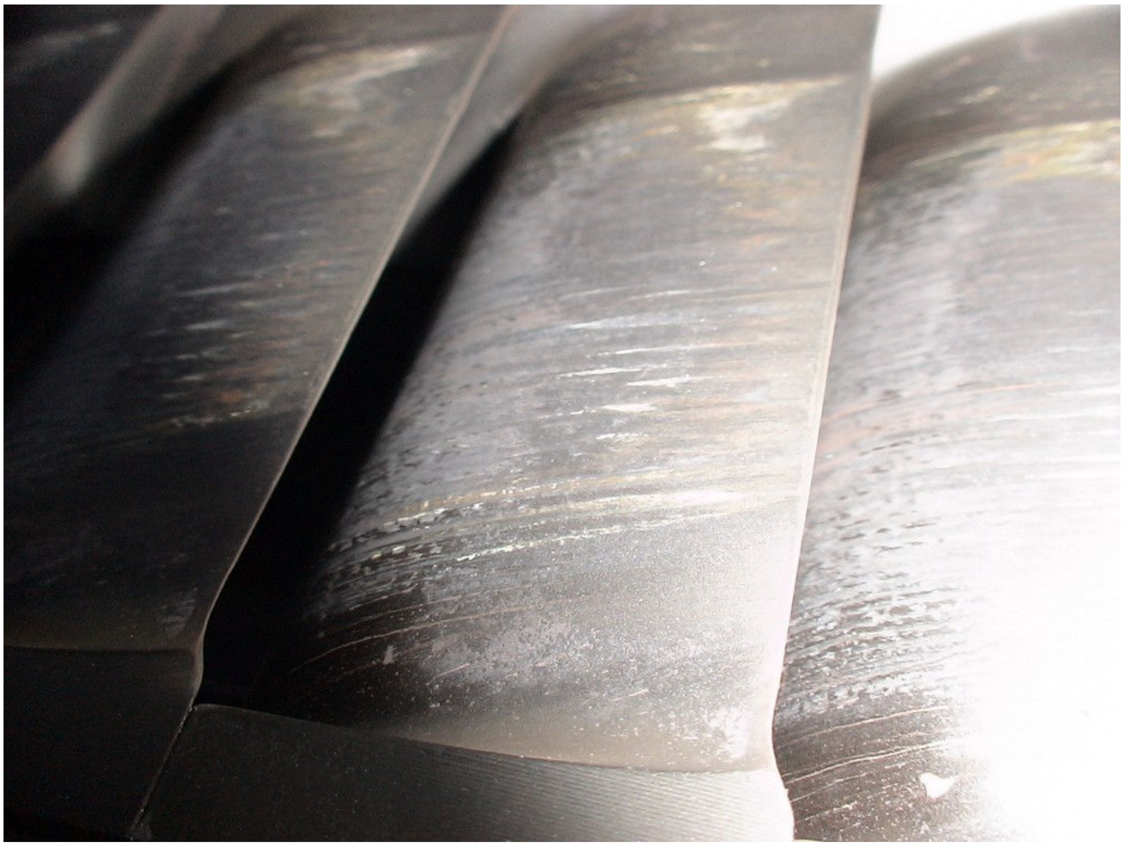

Figure 35: Eroded Steam Turbine Guide Vanes

Erosion is a big problem for the last stages of the low pressure turbine. At these stages, the steam become saturated so it contains droplets. These droplets impinge on the blades that cause erosion. Like deposition this phenomenon increase the surface roughness. It also changes the blade profiles. Turbine manufacturers take protective measures, such as moisture separation and hardening of the leading edge of the blades to prevent erosion as much as possible [76].

Rubbing is the main cause of the sealing problems. The sealing elements can be damaged because of rubbing (Figure 36). The cause of the rubbing can be incorrect alignment, excessive vibration, and high relative movement of the rotor with respect to the casing at the start-ups or turbine trips. Proper operation generally prevents rubbing in steam turbines. The operators should follow the manufacturer manuals at start-ups and shutdowns [77].

Figure 36: Worn Labyrinth Seals

Some of the steam passes through the stages without doing useful work because of the leakage. This leakage also disturbs the steam flow. As a rule of thumb, 0.5 mm additional clearance brings around %1 to %3 efficiency loss [67]. Of course, this can not be generalized. This loss changes according to the design.

The steam turbine fouling reduces the mass flow as well as the isentropic efficiency of the turbine. In the graphs presented below different component degradation mechanisms in steam turbines are displayed [41]:

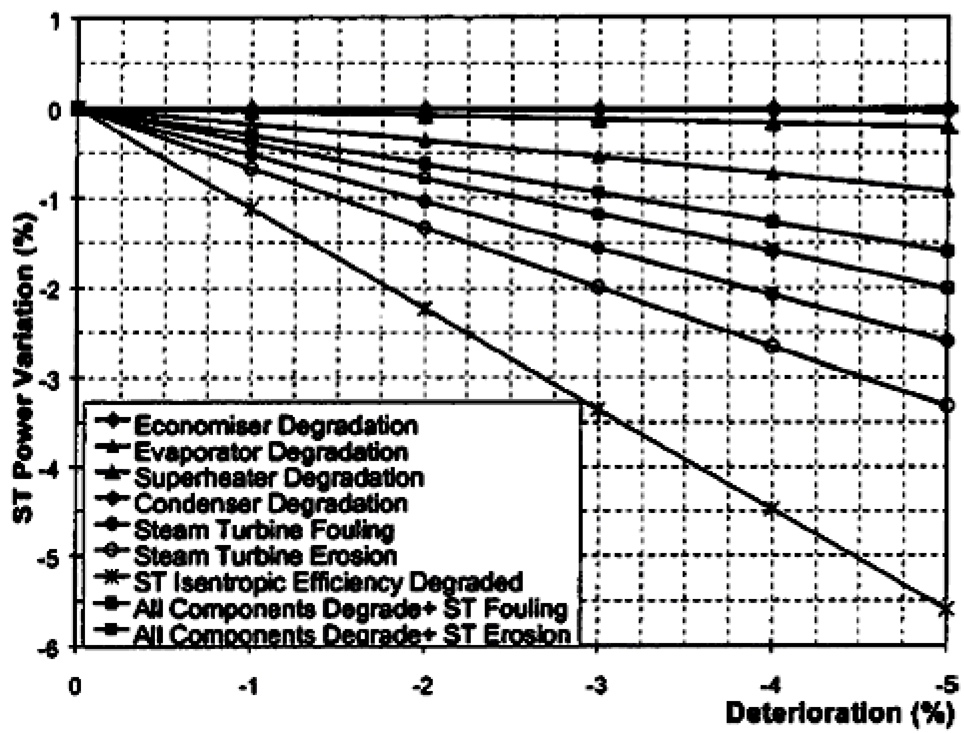

Figure 37: Steam Turbine Power Variation with Component Deterioration [41]

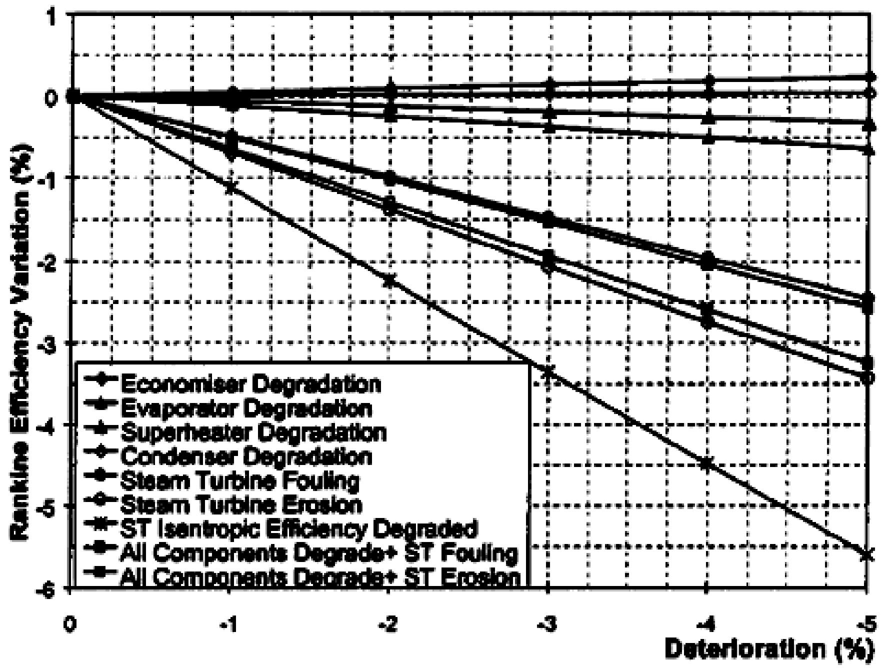

Figure 38: Rankine Efficiency Variation with Component Deterioration [41]

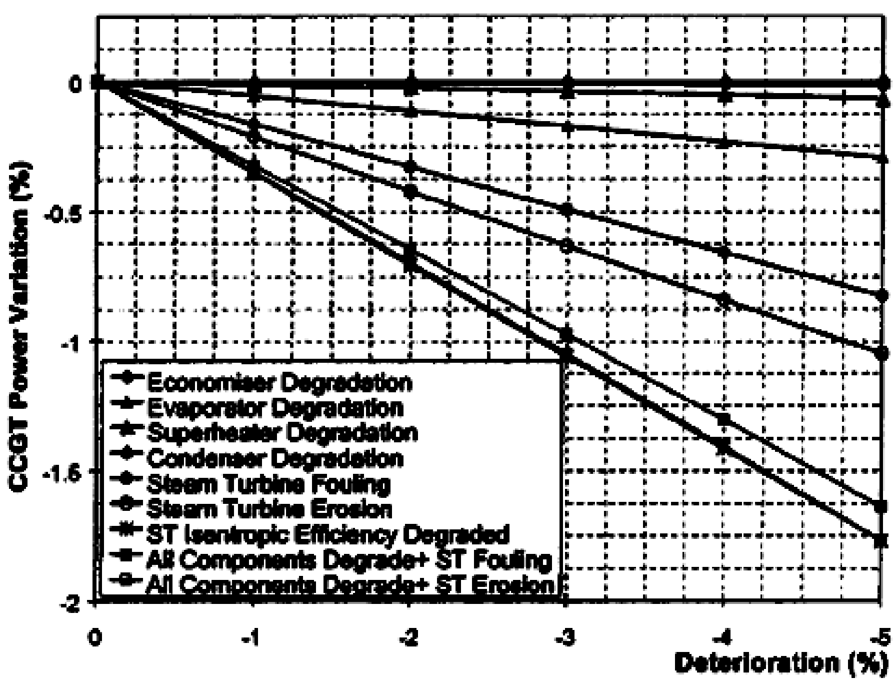

Figure 39: CCPP Power Variation with Steam Component Degradation [41]

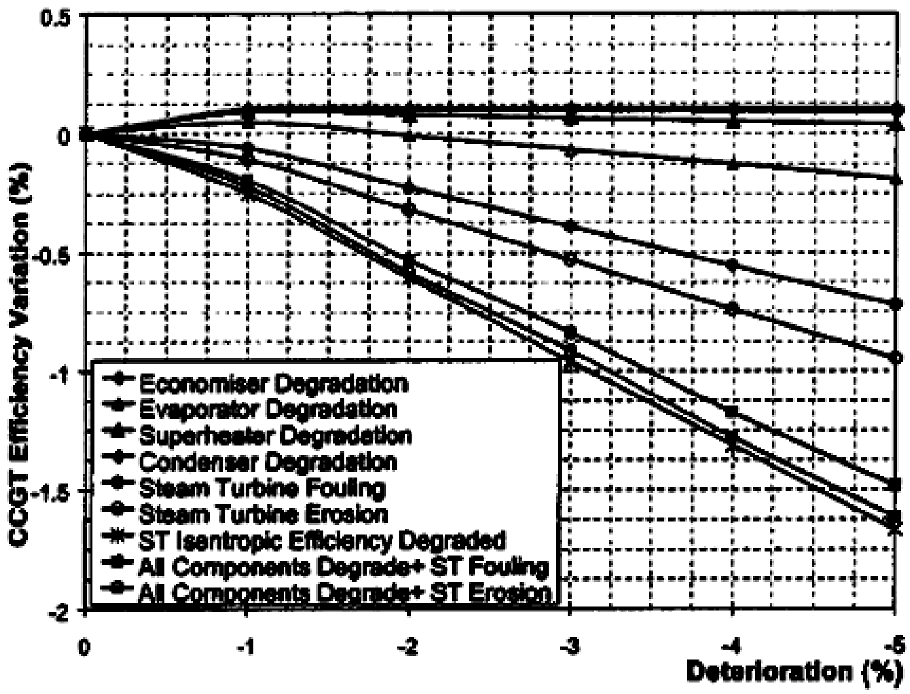

Figure 40: CCPP Efficiency Variation with Steam Component Degradation [41]

4.3. HRSG Degradation

Generally, degradation of the heat recovery steam generator is not considered as a big problem like the gas and steam turbines [78]. There are few studies about the HRSG degradation. However, the degradation may cause significant adverse impacts on the performance of the combined cycle power plants.

The heat recovery steam generators can degrade very rapidly (Figure 41). The cyclic operations, and burning liquid fuel instead of natural gas are the main causes of the heat recovery steam generator deterioration. The liquid fuel combustion products foul the fins of the pipes [79].

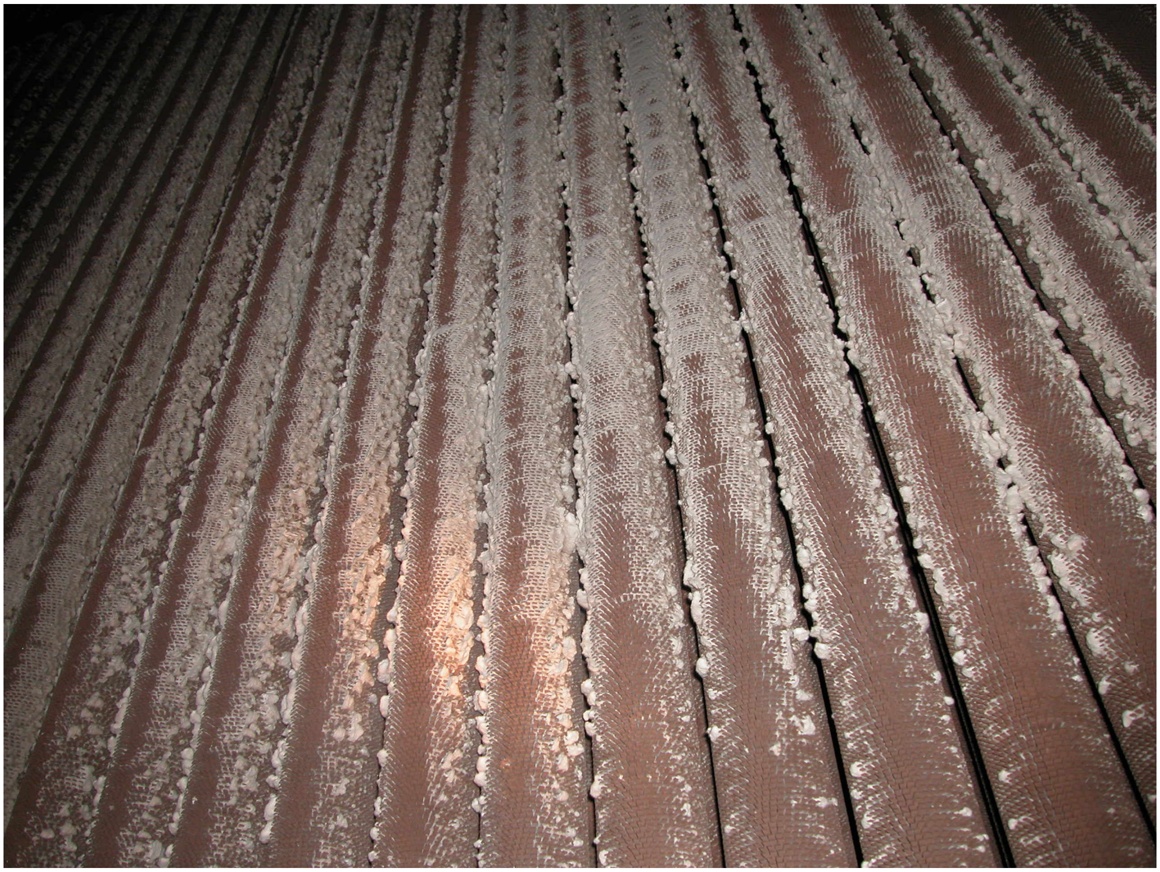

Figure 41: Fouled HRSG Tubes

The typical indication of the heat recovery steam generator degradation is the increase of the stack temperature. The fall of the performance of the heat recovery steam generator brings lower steam generation, lower superheated steam temperatures, and lower steam pressures. All of these operating parameters decrease the power output of the steam turbine which decreases the combined cycle performance.

The boiler feed water pre-heaters are located generally at the end of the heat recovery steam generators for an optimum cycle. The exhaust gas contains gaseous water as a product of combustion. The temperature of the exhaust gas near the tubes can drop below the water dew point while it is passing through the pre-heater. This can cause condensation of the water on the tubes. The liquid water immediately combines with the other materials like sulfur which are also present in the combustion gas. As a result destructive compounds like sulfur oxides arise.

The deposits increase the outside surface roughness of the pipes which leads to higher pressure drops of the exhaust gas. These pressure drops increase the back pressure of the gas turbine. Consequently the output of the gas turbine decreases.

The deposition can also take place inside the tubes. Both inside or outside fouling of the heat exchanger pipes increase the thermal resistance which reduces the effectiveness of the steam generator.

Water treatment facilities, which produce highly purified water are used at power plants to prevent fouling inside the tubes. Water content and pH is controlled at every stage. However, it is impossible to prevent fouling completely.

The following graphs show the effects of the HRSG degradation mechanisms [42, 70].

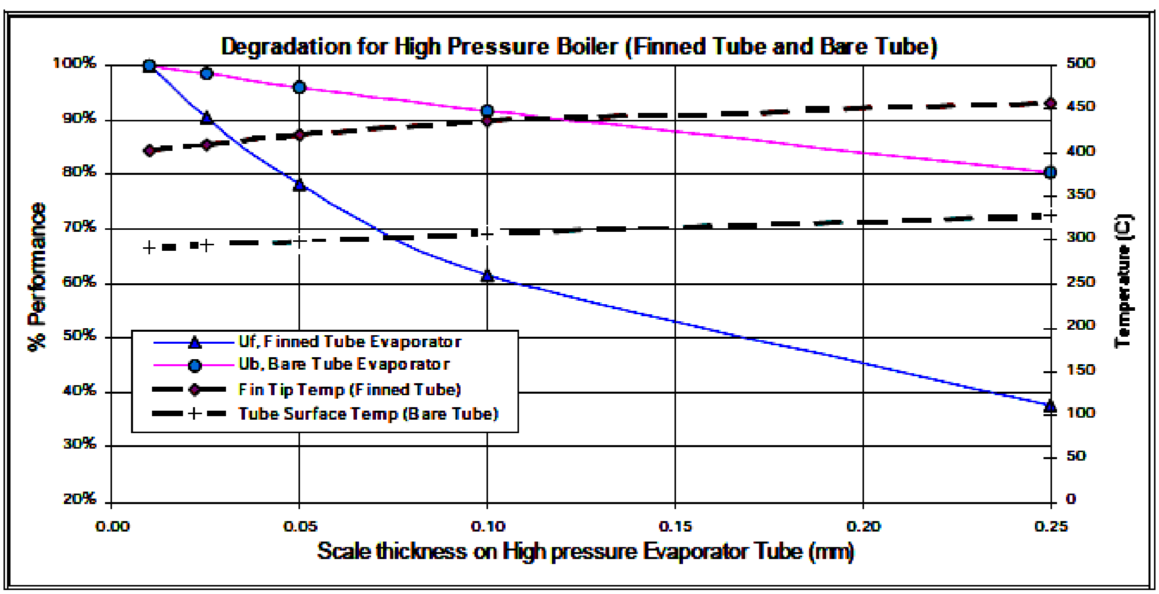

Figure 42: Effect of Fouling on High Pressure Evaporator Performance [70]

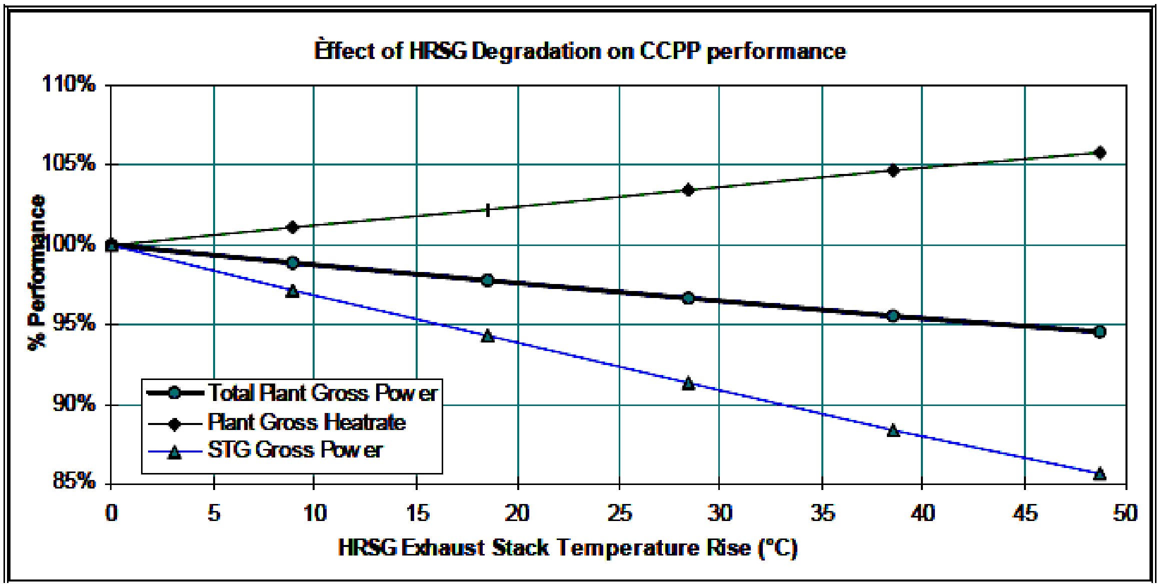

Figure 43: Effect of HRSG Degradation on CCPP Performance [70]

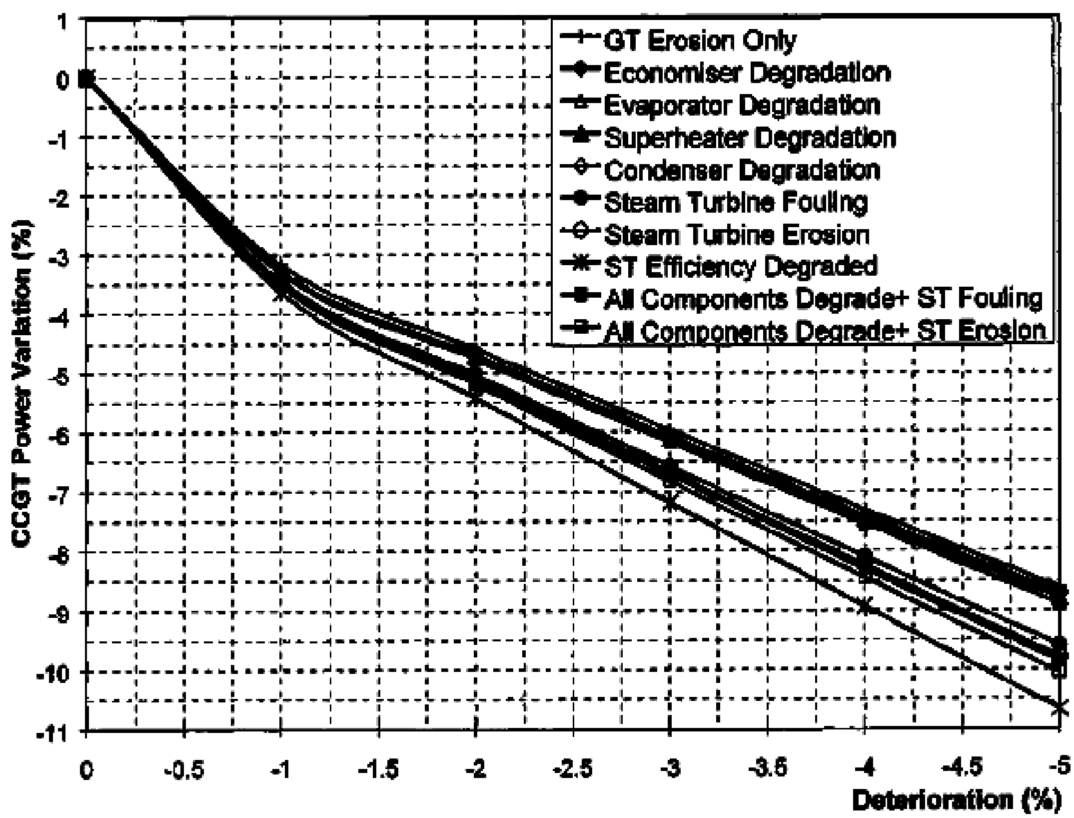

Figure 44: HRSG Efficiency Variation with Gas Steam Cycles
Component Degradation [42]

4.4. Condenser Degradation

A condenser degradation has a major effect on the combined cycle power plant performance (Figure 45) [73].

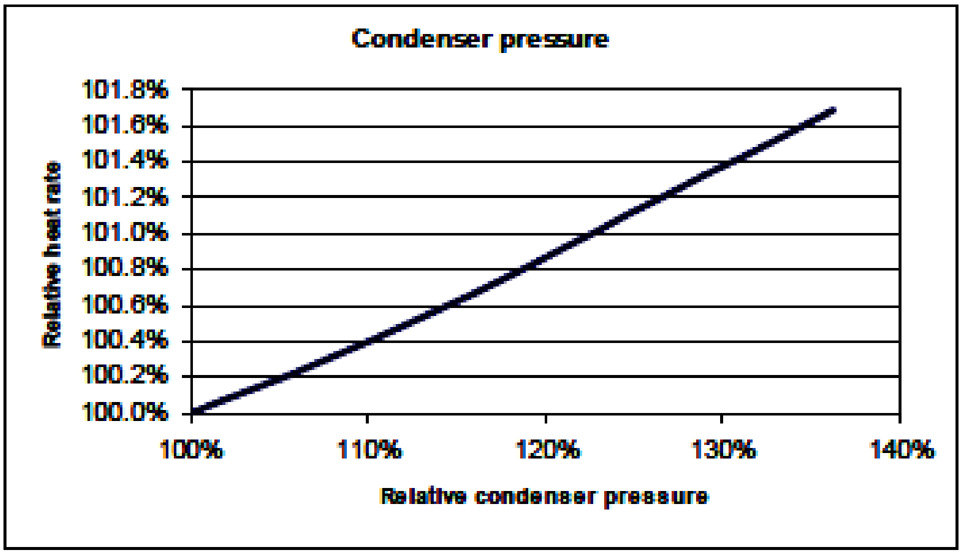

Figure 45: Heat Rate vs. Condenser Pressure [73]

The main mechanisms of condenser deterioration are clogging (Figure 46), fouling of the tubes, and air leakages. Cooling water flow decrease can be accepted as condenser degradation as well.

Figure 46: Clogged Condenser Tubes (www.tubetech.com)

The air leakages disturb the vacuum of the condenser. The air leakages can be identified by controlling the dissolved oxygen content in the condensate water.

The fouling of the tubes decrease the heat transfer rate between the condensing steam and the cooling water. If the cooling water is taken from the sea, biological fouling could be a problem. Generally, sponge balls are used to prevent this fouling. These balls flow through the tubes scraping the deposits. It also prevents the plugging of the tubes. The most scaling materials are manganese and iron in a condenser.

Vacuum equipment deterioration can be another reason of a condenser degradation. The effects of the condenser deterioration on CCPP Performance is displayed in the diagram below [70].

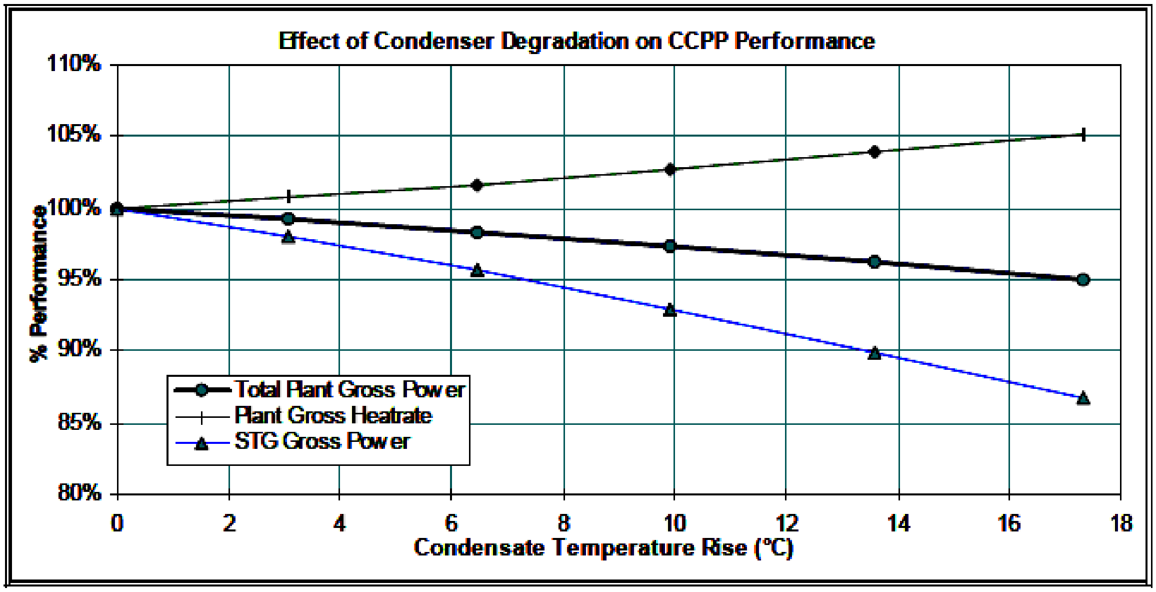

Figure 47: Effect of Condenser Degradation on CCPP Performance [70]

5. REHABILITATION OF THE COMBINED CYCLE POWER PLANTS

In the previous chapter, it is stated that the power plant components degrade with time. The degradation is an unavoidable phenomenon, and the environmental impacts will make it worse. At some point, the operators must perform corrective actions. The reasons for these actions may be site specific. For example, because of degradation, power plant emissions may start to exceed the environmental limits, which can lead to expensive fines and punishment. The contracts of the operators with their customers may be another reason. The contract may dictate a heat rate limit or guaranteed annual production. If the operator can not achieve this contractual clauses because of deterioration, they may face severe penalties. Safety concerns can be another cause as degraded equipments can cause catastrophic failures, which may harm, even kill the power plant staff (Figure 48). Unfortunately, there are many fatal experiences in the industry. A failed gear box, a punctured high pressure pipe with hundreds of bars, or a cracked rotor are some examples. In fact, almost all of the equipments in the power plants are dangerous after their degradation exceeds the safety limits.

Nowadays, another important fact to decide on upgrades or modernizations is the current requirements of the deregulated energy market [2]. The market has become very competitive. As a result, the power plants should run more efficiently to reduce the costs. In this market, the power plants, which are originally designed to operate at base-load, should also run at peak loads with frequent starts and stops. This cycling operation generally needs rehabilitation of the power plants.

Nevertheless, the main reason is the loss of the profits to decide on doing something about degradation. As the degradation gets worse, the plant output decreases which means a reduction in the profit. When efficiency drops, the fuel consumption increases dramatically that has significant effect on the operating expenses. Just a small decrease in the efficiency like one percent can cost millions of dollars per year to the power plant operator. This fact becomes more crucial as the fuel prices increase continuously.

Figure 48: A Disastrous Failure of a Steam Turbine & Generator (www.steamforum.com)

The recovery of the degraded equipments can be made by life-extension or retrofit activities. Life extension generally refers to replacing the degraded components with the brand new ones (Figure 49). Life extension activities recover the performance of the machine to the design conditions.

On the other hand, retrofits can also increase the performance with the state-of-the art advanced technology. Here, it should be noted that the retrofits can be also accepted as life-extension process.

In the energy industry the retrofit term is also named as upgrade. Modernization and rehabilitation terms include both the life extension and upgrade activities.

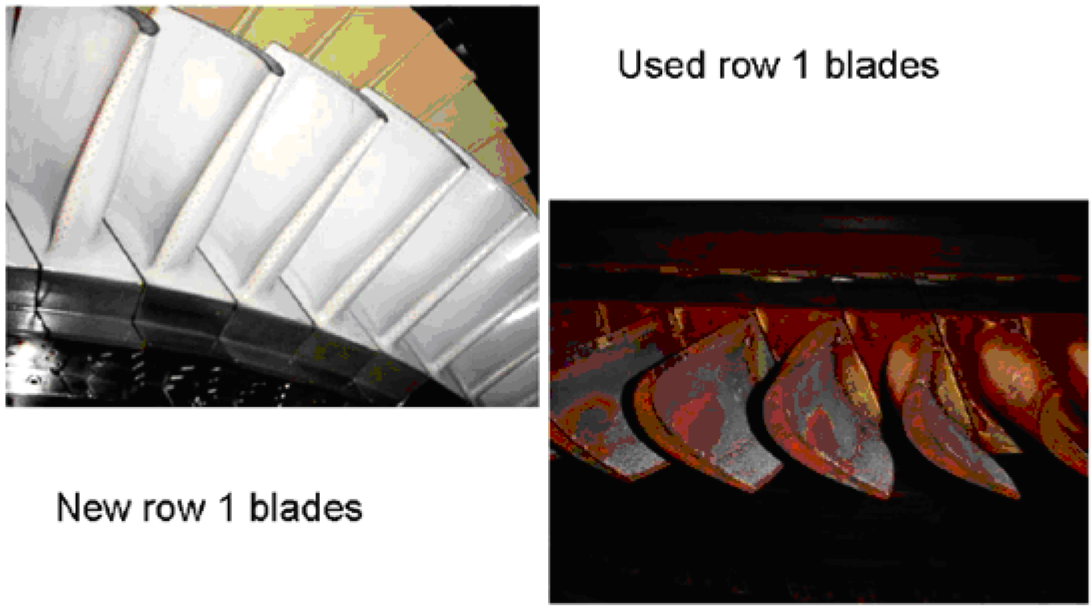

Figure 49: Turbine Blade Replacement [80]

The upgrade projects should be evaluated carefully both technically and economically as the power plant rehabilitations are very expensive investments and they must justify their costs. Significant performance rise can be achieved with low cost by applying a comprehensive performance improvement study.

Technically, it is possible to rehabilitate all of the degraded equipments. However, the upgrade efforts are generally focused on the gas turbines. There are several reasons: First of all the gas turbines have many sections that can be optimized with the evolving advanced technology. For example, the latest advanced materials can resist to higher turbine inlet temperatures which boost both the output and the thermodynamic efficiency. Advanced blade designs provide higher pressure ratios that provide higher performance. As long as the materials are improved and able to survive higher temperatures and pressures, the gas turbine performance will increase.

The heat recovery steam generators consist of tubes on which you can not make any significant upgrades. Generally, the HRSG tubes are replaced because of maintenance and safety considerations. The tubes are changed when the tubes puncture or their wall thickness become thinner than the safety limits. Performance is the secondary consideration for an HRSG rehabilitation.

The manufacturers offer steam turbine modernization projects that they claim to bring significant performance improvements. Replacing the worn sealing rings can have substantial performance recoveries.

Other systems that can be implemented to the plant could have crucial effects on the performance. Some of these upgrade options are: compressor inlet cooling systems, heat recovery steam generator duct burners, etc. The following table indicates some of the modernizations, and their potential efficiency improvements [73].

Table 2: Rehabilitations and Potential Efficiency Improvements

Action	**Description**	**Potential Efficiency Improvement (%HHV)**
Restore the Plant to Design Condition		
Replace/Clean fouled air filters	Fouled air filters increase the pressure drop across the air inlet to the gas turbine compressor	Up to 0.4%
Change Operational Settings		
Increase frequency of compressor cleaning	Compressor washing restores compressor efficiency	Up to 0.5% per wash
Check control system settings	Check IGV angles, instrument calibration and hardware for correct operation	Up to 0.5%
Retrofit Improvements		
Consider inlet air conditioning	Options are evaporative cooler, mist/fog system	Up to 0.5%
Upgrade components to increase turbine inlet temperature	Increased turbine inlet temperature increases efficiency	Application specific
Review air inlet and GT exhaust arrangements	Revised inlet and exhaust duct arrangements may reduce pressure drop.	Up to 0.3%

5.1. Maintenance

The equipment degradation is unavoidable; however, its effects can be reduced with proper maintenance activities. Experienced field personnel, original spare parts and following the required maintenance intervals are essential necessities to prevent the power plant from degradation as much as possible.

A professional fact finding should be carried out before the overhaul and during the overhaul to detect the degraded components. Performance monitoring and condition monitoring programs also provide valuable information before the revisions.

Generally, steam turbines operate without opening for long service intervals. On the other hand, the gas turbine hot gas path must be controlled and overhauled frequently. This is not surprising as the gas turbine components run under much more severe conditions than a steam turbine. All of the gas turbine manufacturers have maintenance guidelines that show the operating life of the gas turbine components. It is very important to follow these instructions to provide safe and profitable operation.

The heat recovery steam generator may need repairs because of the failed tubes that cause water leakage. Condenser can also face tube leaks and clogging that must be recovered during overhauls.

5.2. Cost benefit analysis

Technically, it is possible to carry out life extensions or modernizations to all of the equipments in a power plant. However, it may be economically infeasible. Generally, power plants have limited budgets, and these corrective actions are very expensive. Therefore, power plant operators need to know which recovery activities are more cost effective than the others. Reliable calculations and engineering information are needed to decide which equipment should be rehabilitated. Therefore, cost benefit analyses should be carried out before a modernization or a life extension project. The cost benefit analyses reveal the most cost effective rehabilitation combinations. Nevertheless these analyses are not so easy to perform. Every plant may have custom configuration combination to suit to the site conditions. Deriving the necessary information from an aging plant is very difficult, and there are not many professionals who can perform these analyses.

For a cost benefit analysis, benefits and costs should be obtained for rehabilitation projects. Costs can be obtained relatively easy. The power plant will ask for a quotation from the manufacturers for a possible rehabilitation project. However, calculating the benefit accurately is very difficult. First of all, the power plant should be simulated with a computer-based model. The simulation model should represent the power plant correctly.

The model must be tuned to get the accurate model. The heat recovery steam generator tube bundle drawings should be studied carefully, and their characteristics must be applied to the model correctly. The plant commissioning/acceptance test records or the first operating parameter logs can be used for calibration purposes. The plant model should reflect the design performance that will be the baseline point to find the performance deteriorations.

After the design performance of the power plant is obtained, the existing operating data measurements will be used to calculate the degradations of the equipments. The effect of each component on the power plant performance should be obtained with the simulation model. This information is used to calculate the benefit if the degraded equipment is rehabilitated.

There are three different arguments that are used in cost benefit analysis studies. These are the net present value, the internal rate of return and the payback time. These parameters are widely used in engineering economics in many different industries.

- The net present value is the most transparent parameter. It reflects the total present values of the annual cash flow.

- Internal rate of return indicates the discount rate which makes the net present value equal to zero.

- Pay back time reveals that in how many years an investment will justify its cost

.

5.3. Gas turbine Rehabilitation

Gas turbines are first equipments to be considered for life extension and upgrade efforts. The causes of this fact is already presented. Increasing the turbine inlet temperature is the main concern for the manufacturers and operators. It is the most important parameter for the gas turbine performance. Both the output and the efficiency boost as much as the turbine inlet temperature increases. Therefore, many studies are focused to develop advanced materials and high-tech cooling techniques to provide higher turbine inlet temperatures. It should be noted that increasing the gas turbine performance will also improve the combined cycle performance.

State of the art sealing systems such as abradable coatings and brush sealing are another option for a gas turbine retrofit. Other upgrade options are implementing a fuel heating system, changing the blade profiles etc.

The control system upgrades are important especially in cycling operation. In control system modernizations, IGV (Inlet Guide Vanes) and the gas control valve is scheduled in order to prevent under-firing or over-firing. The under-firing will cause the machine to run below its performance capabilities. On the other hand, over-firing will reduce the life of the hot gas path components. These modernizations optimize the gas turbine operation.

Figure 50 shows the typical degradation diagram of a gas turbine, and the effect of the retrofits on the performance [60].

Table 3 reveals several potential gas turbine retrofit options [10].

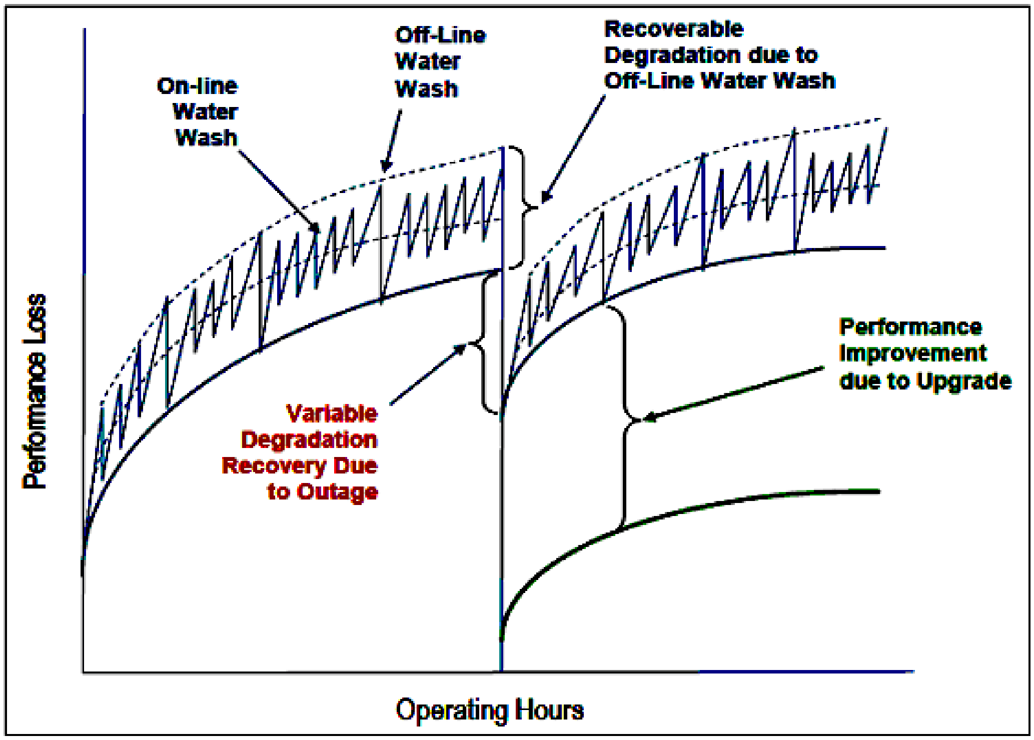

Figure 50: Typical Gas Turbine Degradation Trend [60]

5.4. Steam turbine

Steam Turbine is the second important equipment in a combined cycle power plant to consider an upgrade. Life extension is not a common application since the steam turbine can operate without opening for many years.

There are steam turbines operating for more than 50 years. This is not surprising because the working fluid is purified water that has negligible contaminants. Another reason is that the highest temperatures are around 600 °C. While gas turbines operate under much more aggressive conditions such as operating temperatures up to 1500 °C, and working fluid contains high amount of contaminants. Therefore, life extension is an important consideration for gas turbines.

Table 3: Gas Turbine Retrofit Options

Modification	**Remarks**	**Potential Performance Improvement**
Turbine inlet Temperature Increase	Increased turbine inlet temperature + upgraded hot parts + upgraded low emission combustion system = performance (power output, heat rate) and improved reliability.	- GT Power: 5.2% better - GT Heat Rate: 1.3% better - CCPP Power: 5.0% better - CCPP Heat Rate: 1.2% better
Improved seal technologies	Advanced sealing technologies (i.e., Brush seals, E-seals) reduce leakage air loss significantly resulting in performance (power output, heat rate) benefit.	- GT Power: 1.7% better - GT Heat Rate: 0.7% better - CCPP Power: 1.5% better - CCPP Heat Rate: 0.5% better
Abradable coating	Abradable coating on ring segments (facing rotating blades) reduces hot gas leakage between blade tips and stator significantly resulting in performance (power output, heat rate) benefit.	- GT Power: 1.3% better - GT Heat Rate: 1.3% better - CCPP Power: 0.4% better - CCPP Heat Rate: 0.4% better
FGH(Fuel Gas Heating) system	Waste heat from the turbine cooling air cooler is utilized to heat the fuel gas, resulting in a heat rate improvement	- GT Power: 0.0% better - GT Heat Rate: 0.6% better -CCPP Power: 0.0% better -CCPP Heat Rate: 0.6% better
Air intake filter	HEPA filters effectively removes dusts from inlet air. Provides significant benefit on gas turbine performance deterioration as compared to conventional filters.	
DLN (Dry Low NOx) combustion system	Emissions are reduced significantly without water/steam injection by multi-premixed combustion system.	

The steam turbine can be recovered to its as-new condition or it can be upgraded with the latest technology. The retrofits can bring high performance gains because most of the installed turbines are very old. They can perform much better with the state of the art technology that offers advanced blade shapes and sealing systems (Figure 51).

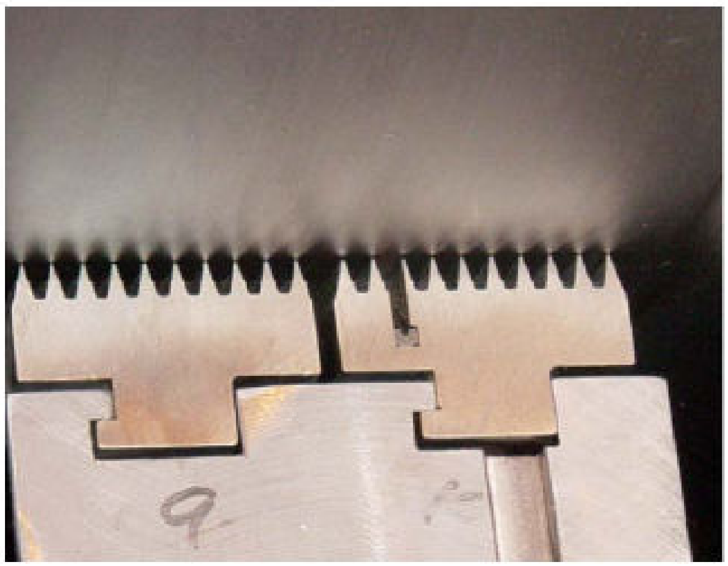

Figure 51: New Sealing Components [77]

The isentropic efficiency of a steam turbine has a direct effect on the overall power plant heat rate because it does not burn fuel. The operators pay special attention for the performance of the steam turbine trying to recover lost performance during the overhauls by replacing the sealing rings, repairing the eroded blades, removing the deposits on the steam path, etc. However, sometimes the lost performance can not be recovered completely without changing blades or the other components.

5.5. HRSG Rehabilitation

The degraded sections can be recovered by cleaning the scaling and deposits. There are different cleaning applications such as pressurized water cleaning, acid cleaning and dry-ice blasting. However, these methods can not clean completely especially the finned tubes. Therefore, replacing the tube bundles may be considered that will recover the HRSG to its design performance.

The HRSG sections consist of ordinary finned tubes so there are not many upgrade opportunities like gas turbines with high technology. Increasing the heat exchanger surface area may be thought as a retrofit that provide lower pinch point and approach temperatures. However it is generally impossible because the outer casing of the HRSG is already constructed and it is impossible to change it to obtain additional space.

Optimization of the operating parameters of the HRSG will be a good practice to improve the performance. However a highly skilled and experienced engineer, who can both perform a thermodynamic optimization and generate a control logic according to that, should adjust the control system.

5.6. Condenser Rehabilitation

Condenser degradation is one of the main reasons for a performance loss in a combined cycle power plant. The shell pressure of the condenser highly affects the performance. If the condenser begins to deteriorate, the condenser shell pressure will start to rise that reduce the expansion line of the steam turbine. As a result, both the shaft output and the efficiency decrease.

A Condenser performance can be recovered by cleaning the outside and inside of the tubes. This may bring the performance close to its design values. However a replacement of the tubes may be considered to recover completely. Like HRSG, Condenser does not have many upgrade options. Replacing the vacuum system to provide lower pressures may be a choice. Also the cooling water system may be rehabilitated to improve the condenser performance.

6. MODELING AND ANALYSIS OF AMBARLI COMBINED CYCLE POWER PLANT

Combined cycle power plant performance and degradation issues discussed above have been applied to model and analyze a real power plant as a case study. The Ambarlı Combined Cycle Power Plant that is the subject of this case study is located near Istanbul (Figure 52). It was commissioned in 1990. It has 1350 MW installed power. When it started operation, it was the most efficient combined cycle power plant in the world. Both the steam turbines and gas turbines were manufactured by Siemens AG, a well-known German company. The heat recovery steam generators were constructed by an Austrian company, Simmering-Graz-Pauker. The power plant consists of three blocks, each block has two gas turbines, two heat recovery steam generators that supply one steam turbine. Therefore, the power plant has six gas turbines (Figure 53), six heat recovery steam generators, three steam turbines and three condensers. In addition there are lots of auxiliary equipments like pumps, valves etc. In this study, one of these blocks has been analyzed.

Figure 52: Ambarlı Combined Cycle Power Plant

Figure 53: Ambarlı Siemens V94.2 Gas Turbines

In order to perform the performance analyses all necessary information has been obtained with the help from the power plant engineering department. Once a computer-based model of the power plant has been developed, the model has been tuned using these actual plant design specs. These efforts have provided a calibrated model to analyze design performance of the power plant.

In the next step, heat and mass balance and performance evaluation of the complete block has been performed using the actual operation data. The current performance of the plant has been calculated by this analysis. The performance degradation of the power plant has been observed by comparing current and design performance results.

In order to identify individual component contributions to overall performance degradation a sensitivity analysis has been performed as well. This analysis provided the magnitude and impact of the individual component degradations on the combined cycle power plant performance.

After the model has been built, calibrated, and component based sensitivity analyses have been performed, various rehabilitation scenarios have been also considered, and their potential improvements have been studied.

Finally, a cost benefit analysis has been carried out to identify cost effective life extension and upgrade opportunities. This analysis aims to serve as a good tool to guide the power plant operators through their performance improvement projects.

6.1. Simulation Model of the Power Plant

The power plant model has been constructed according to the flow chart of the power plant (Figure 54), HRSG drawings (Figure 55, Figure 56), technical data of the equipments. Then, this model has been tuned using the acceptance/comissioning test data. This information has been obtained by the help of the Ambarlı Power Plant Engineering Department.

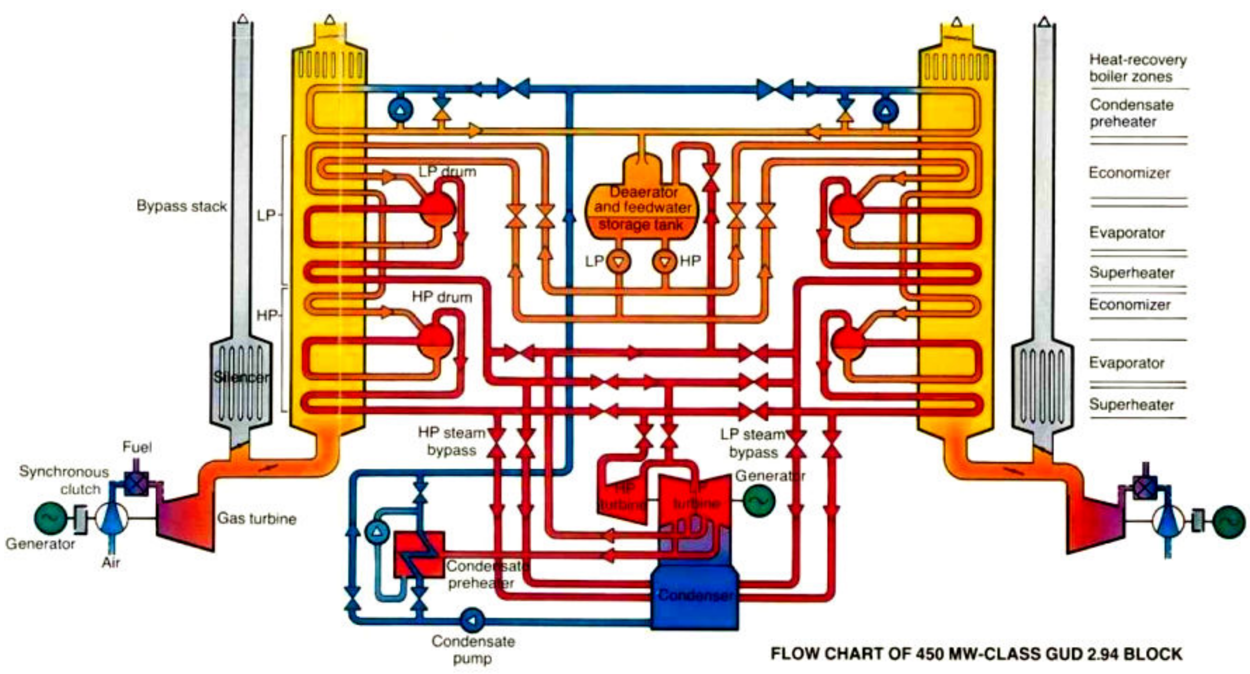

Figure 54: The flow chart of a block at Ambarlı Combined Cycle Power Plant

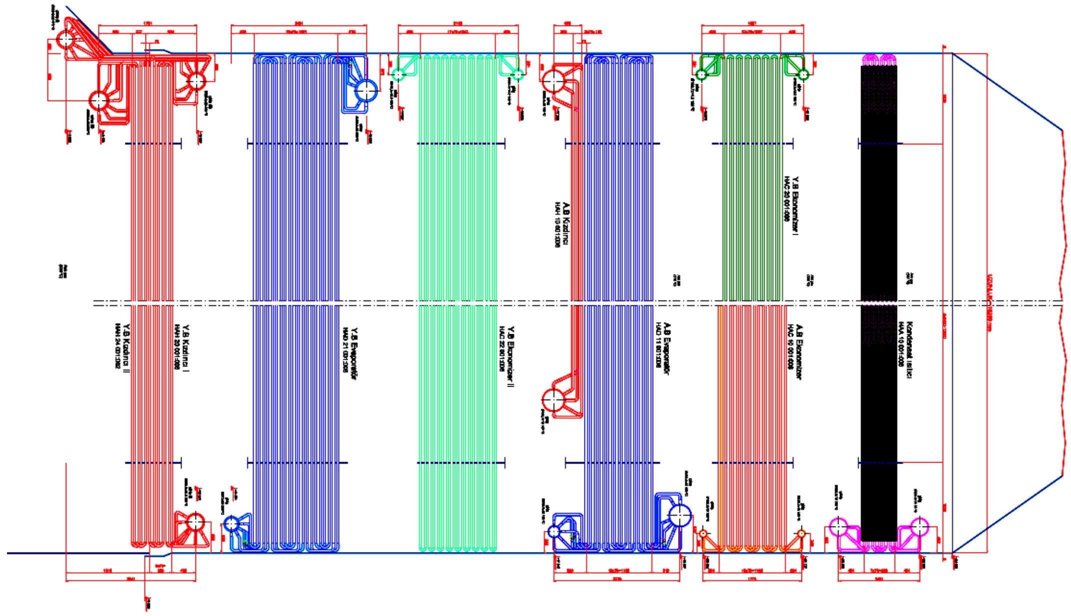

Figure 55: HRSG Tube Bundle Layout Drawing

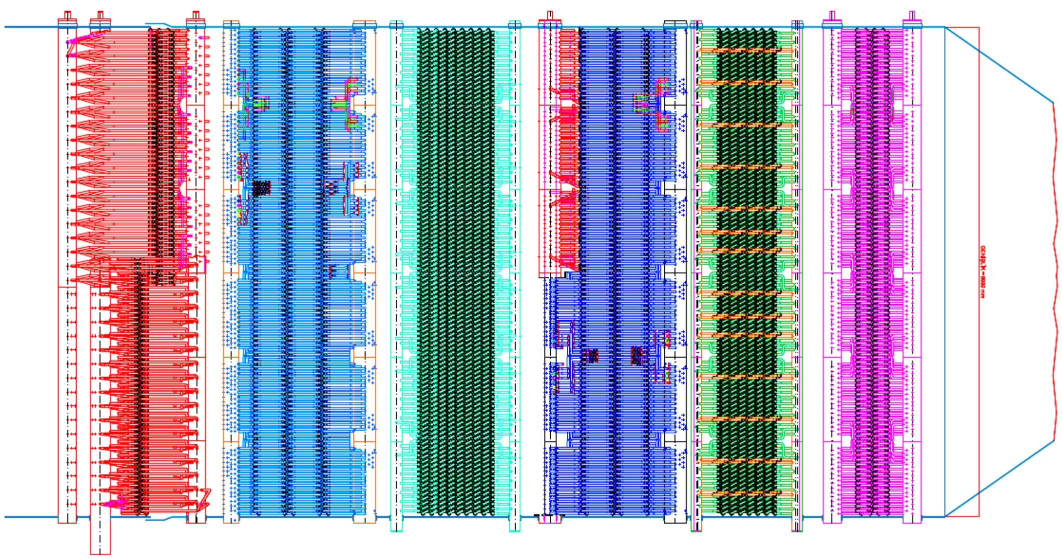

Figure 56: HRSG Tube Bundle Layout Drawing (Other View)

In this study, Gatecycle has been used as a tool to prepare the computer based models. Gatecycle is a commercially available code that provides templates and subroutines to solve thermodynamic and heat transfer equations. The templates and library routines can be customized and configured to build flexible power plant models to analyze unique plant features and customized plant operations tailored to site conditions,

In order to achieve a successful and complete plant model, every component has been modeled, customized and connected to each other. Using actual plant specific component design data the local heat and mass balance equations have been solved for each component. Every system component in the model should be in mass and energy balance. Otherwise, the analysis would not converge. Therefore, great effort has been spent to model the power plant. Once a comprehensive model that includes all components down to small pumps has been constructed, the overall heat and mass balance equations have been solved to calculate the overall power plant performance.

The gas turbines have been modeled according to the acceptance test data. After adjusting the gas turbine models, the HRSG has been modeled according to the HRSG layout drawings (Figure 55, Figure 56). Then, steam turbine and condenser have been modeled. Finally, the remaining components have been added to complete the cycle. These equipments are auxiliary equipments like condensate pump, boiler feed pumps, feed water heater, etc. The modeling has been finished by connecting the equipments according to the flow chart (Figure 57).

Every power plant component has its own model that solves local heat and mass balance equations for that component. The Second Law of Thermodynamics is used to tune these models. For example, steam turbine isentropic efficiency is used to adjust steam turbine model.

The acceptance test results have been used for tuning and calibrating these models, because it reflects the power plant performance when it was new. The manufacturer guaranteed performance data may lead to incorrect results because the manufacturers leave a margin on the expected performance to be on the safe side. If the equipments can not reach the guaranteed performance, manufacturers may get heavy fines according to the contract. Therefore, using the acceptance test data is more accurate. If a performance monitoring system is available on site, the necessary data can be obtained from the data logger code. In order to obtain as-new plant parameters, the data should be taken from the beginning of the operation. The Ambarlı Power Plant does not have a performance monitoring system. Therefore, in this study the necessary information has been taken from the acceptance test results. According to the acceptance tests, the component models have been tuned. When the model has been validated against the acceptance tests, the as-new model of the power plant has been obtained. This model calculates the design performance of the plant. This is the performance that the power plant was performing when it was new. This obtained performance information is used as the baseline point of the performance degradation analyses. The degradation of the equipments are determined by comparing the current performance of the components with the baseline performance.

6.1.1. Gas Turbine Model

The gas turbine component models have been tuned to simulate Siemens V94.2 Gas Turbines of the Ambarlı Power Plant. The model has been started per design specs. Then, the efficiency of compressor and turbine sections have been adjusted to match the acceptance test results.

Ambarlı Gas Turbine acceptance tests were conducted during simple cycle operation. This means that the HRSG had not been constructed at the test period. Unfortunately, the gas turbine as-new operating parameters in combined cycle operation mode are not available. As a result, the gas turbine model has been designed and calibrated separately using the simple cycle acceptance test data. The acceptance test results (Table 4) are used to model the gas turbine:

The isentropic efficiency of compressor and turbine are the main design parameters of the gas turbine model. These values have been iterated until the local heat and mass balance calculations of the gas turbines match the acceptance test data. After the tuning, the gas turbine performance calculations have yielded the same results with the acceptance test data.

Table 4: Gas Turbine Simple Cycle Acceptance Test Data

Operating Parameter	**Value**	**Unit**
Compressor Inlet Temperature	15.59	°C
Compressor Inlet Pressure	1.0114	bar
Compressor Inlet Air Mass Flow	505.36	kg/s
Compressor Discharge Pressure	9.9	Bar
Compressor Discharge Temperature	328.1	°C
Natural Gas Mass Flow	9.025	kg/s
Natural Gas Lower Heating Value	49311	kj/kg
Combustion Chamber Pressure Drop	189.1	mbar
Exhaust Mass Flow	514.38	kg/s
Turbine Inlet Temperature	1039.3	°C
Exhaust Temperature	535.5	°C
Gross Power	149.15	MW
Efficiency	33.46	%
Heat Rate	1653	kcal/kW

The available information about the combined cycle acceptance test results are:

Table 5: Gas Turbine Combined Cycle Acceptance Test Data

Operating Parameter	**Value**	**Unit**
Gross Power	146.09	MW
Efficiency	32.86	%
Turbine Inlet Temperature	1050	°C

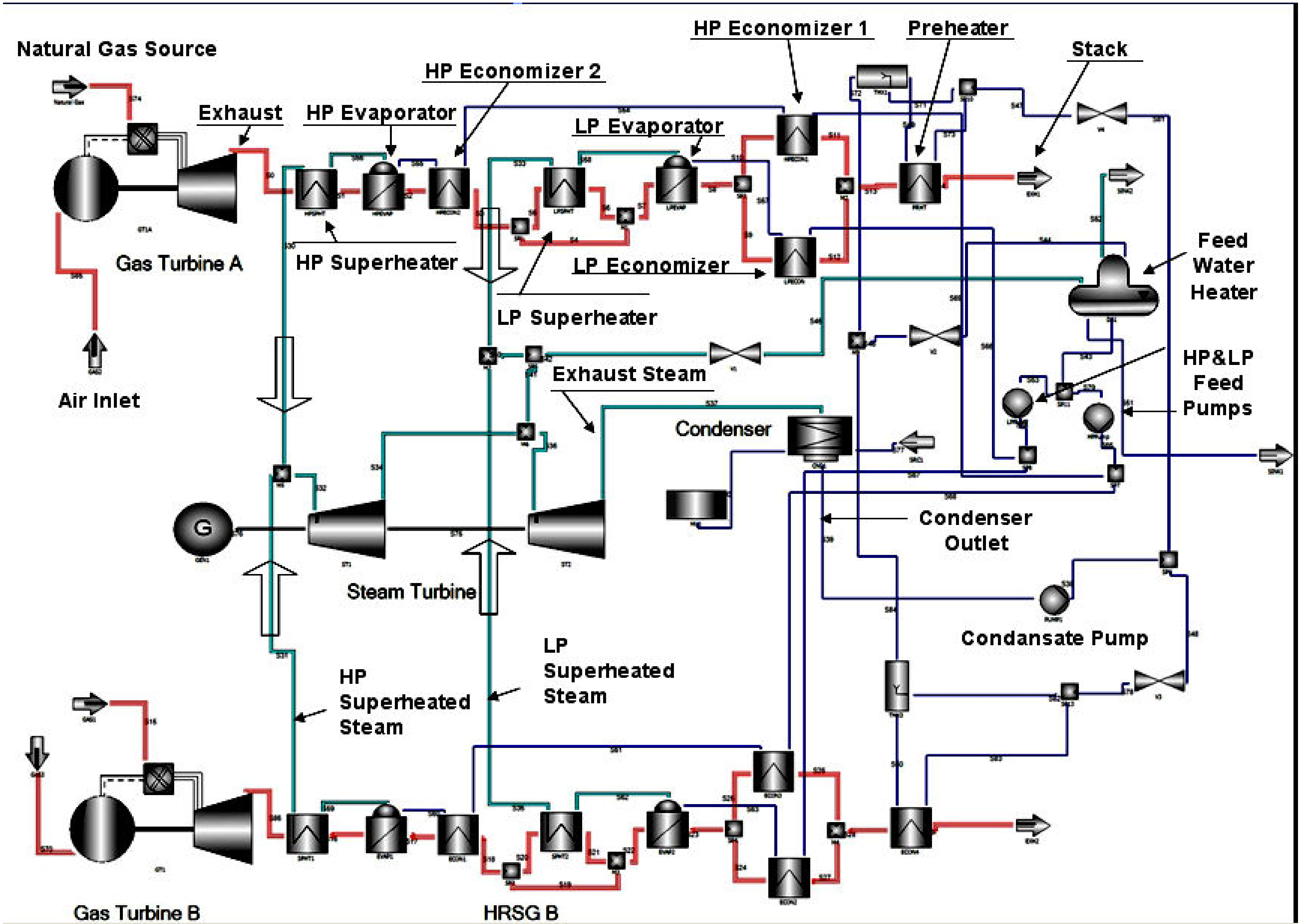

Figure 57: The Complete Model of the Ambarlı Power Plant

Using the turbine inlet temperature data, the exhaust temperature during the combined cycle operation has been determined as 544 °C. It was 535.5 °C in simple cycle operation. The exhaust temperature has been iterated until the turbine inlet temperature has risen to 1050 °C from 1039.3 °C. When exhaust temperature is changed to 544 °C, the second law efficiencies of the gas turbine components are not changed. It means the hardware of the gas turbine is fixed. Therefore, this change just reflects the machine behavior in a different off-design condition.

Detailed study of the combined cycle acceptance test data revealed that the performance of the gas turbine had reduced. The reason for this is the backpressure of the gas turbine exhaust duct. This backpressure occurs as HRSG tubes produce a resistance to the exhaust gas flow of the gas turbine. This exhaust flow resistance increases the backpressure, which decreases the gas turbine performance. During the combined cycle design, this shortfall of the performance is adapted on the gas turbine model by adding an energy loss parameter to the local energy balance equation. The magnitude of this energy loss has been found by iteration. Its value has been iterated until the results validate the combined cycle acceptance test data.

The acceptance tests for the second gas turbine were conducted one year later than the first one's tests. However, detailed operating parameters are not available in the acceptance test report like the first gas turbine. Its gross power and fuel consumption values are the only available information. These values are very close to the other machine. Therefore, the first gas turbine model design parameters have been also used for the second gas turbine. The slight performance difference between the gas turbines has been adapted to the models.

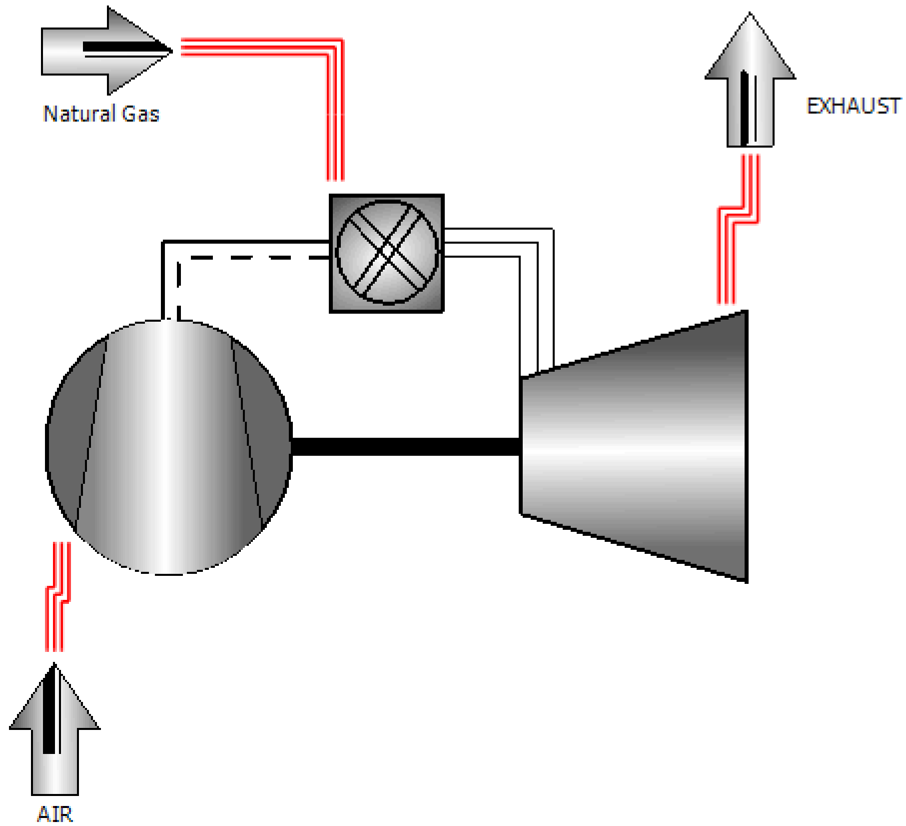

Figure 58: Gas Turbine Model

Local heat and mass balance equations for gas turbines are stated as follows:

Mass balance:

$$\dot{m}_{air} + \dot{m}_{fuel} = \dot{m}_{exh} \qquad [12]$$

Energy Balance:

$$\dot{m}_{air}\, h_{air} + \dot{m}_{fuel}\, \eta_{fuel} (LHV_{NG} + h_{fuel}) = \frac{Power_{GT}}{\eta_{generator}} + \dot{m}_{exh}\, h_{exh} \qquad [13]$$

The operating parameters of the gas turbine have been found by solving these heat and mass balance equations.

The isentropic efficiency of the compressor has been found by:

$$\eta_{ise,comp} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad [14]$$

The isentropic efficiency of the turbine has been calculated using the following relation:

$$\eta_{ise,\exp} = \frac{h_3 - h_4}{h_3 - h_{4s}} \quad [15]$$

These isentropic efficiency levels have been used to tune the gas turbine model. These second law parameters define the actual operation of the engine. The ideal expansion power in turbine is calculated by subtracting the turbine exhaust gas enthalpy that is determined according to Ideal Brayton Cycle, from turbine inlet gas enthalpy. This is defined as:

$$P_{ideal} = h_3 - h_{4s} \quad [16]$$

The actual expansion power is determined by subtracting the actual exhaust gas enthalpy from the turbine inlet gas enthalpy:

$$P_{actual} = h_3 - h_4 \quad [17]$$

The ratio of the actual operation power to the ideal cycle has been used to calibrate the component models. For example, if the turbine inlet temperature is known, the desired exhaust temperature can be determined using the following equation:

$$h_4 = h_3 - \eta_{ise,\exp}(h_3 - h_{4s}) \quad [18]$$

The isentropic efficiency has been iterated until the measured exhaust temperature is reached. Here, the temperatures can be determined from enthalpy values by thermodynamic principles. All of the other operating parameters have been adjusted by applying the same procedure.

The compressor discharge temperature is found by:

$$h_2 = \frac{h_{2s} - h_1}{\eta_{ise,comp}} + h_1 \quad [19]$$

The compressor isentropic efficiency has been iterated until the desired compressor discharge temperature is obtained.

6.1.2. The HRSG Model

The most important and difficult part of the simulation has been the HRSG model construction. The HRSG has been modeled by applying the flow chart of the plant (Figure 54) and the HRSG layout. After that, the component models have been adjusted according to the acceptance test data.

The acceptance data used for HSRG model calibration are listed below.

Table 6: HRSG Acceptance Test Data

Operating Parameter	Value	Unit
HP Steam Temperature	521.7	°C
HP Steam Pressure	73.84	Bar
HP Steam Flow	65.4	kg/s
LP Steam Temperature	200.4	Bar
LP Steam Pressure	6.02	°C
LP Steam Flow	13.35	kg/s

The second law efficiency is the effectiveness of the heat exchangers. It is used to tune the HRSG components according to the acceptance test results.

The HRSG units at Ambarlı Power Plant have two pressure levels: high pressure and low pressure. According to the layout the HRSG subsystems have been lined up as follows: HP Superheater, HP Evaporator, HP Economizer 2, LP Superheater, LP Evaporator, HP Economizer 1 &LP Economizer (they are in parallel), Pre-heater. The exhaust gas of the gas turbine flows through these sections, respectively. During this flow configuration, the exhaust gas rejects heat to the water and steam that flow inside the HRSG tube bundles.

The condensate water, coming from the condenser become superheated steam at the superheater outlet in HRSG. First of all, the condensate water is pumped from the condenser to the preheater. Here, its temperature is raised to about 100 °C. The heated water pours into feed water tank. In the plant this tank is used for dearation. That is, the oxygen content of the water is removed by this operation to prevent oxidation and corrosion inside the tubes. This tank feeds two pumps: HP feed and LP feed pumps. LP feed pump pressurizes water to around 6 bar, and drive the water to the LP Economizer. Here, the water is further heated, and its temperature is raised close to the saturation temperature (~160 °C) at working pressure. The water exiting the Economizer pours into the LP Drum. Then, the water flows through the evaporator to vaporize. Vaporized saturated steam returns to the LP Drum. Then, the saturated steam exits the drum, and flows to the LP Superheater. There, saturated steam becomes superheated (~200 °C). Finally, superheated steam flows to the steam turbine. The HP water/steam circuit has a similar flow path as well. The HP boiler feed pump pressurizes feed water (~80 bar), and drives it to the HP Economizer 1. Unlike LP circuit, HP stage has two economizers. The liquid water is heated to around 170°C in Economizer 1. Then, heated liquid water flows to the HP Economizer 2, where its temperature is further raised to around 290 °C. The liquid water leaving the HP Economizer 2 pours into the HP drum. Then, it flows to the HP evaporator to become saturated steam. Finally, saturated steam becomes superheated in HP Superheater (~520 °C). The superheated steam is supplied to the steam turbine HP section. Both HP and LP steam stream expand in steam turbine to generate power. (Figure 59)

The local heat and mass balance for HRSG is derived as:

Mass balance:

$$\begin{aligned} &\dot{m}_{gas,in} + \dot{m}_{LP,feedwater} + \dot{m}_{HP,feedwater} + \dot{m}_{preheater,in} = \\ &\dot{m}_{gas,stack} + \dot{m}_{HP,Steam} + \dot{m}_{LP,Steam} + \dot{m}_{\Pr eheater,Out} \end{aligned} \qquad [20]$$

Energy Balance:

$$\begin{aligned} &\dot{m}_{gas,in}\, h_{gas,in} + \dot{m}_{LP,feedwater}\, h_{LP,feedwater} + \dot{m}_{HP,feedwater}\, h_{HP,feedwater} + \\ &\dot{m}_{preheater,in}\, h_{preheater,in} + P_{pump} = \dot{m}_{gas,stack}\, h_{gas,stack} + \dot{m}_{HP,Steam}\, h_{HP,Steam} + \\ &\dot{m}_{LP,Steam}\, h_{LP,Steam} + \dot{m}_{\Pr eheater,out}\, h_{\Pr eheater,out} \end{aligned} \qquad [21]$$

The effectiveness of the heat exchangers are calculated as:

$$\text{Effectiveness} = \frac{(h_{gas,in} - h_{gas,out})}{(h_{gas,in} - h_{gas,\min})} \quad [22]$$

In this study, effectiveness has been used as the design variable to calculate the performance of the HRSG components. The energy balance and effectiveness equations are used to calibrate the model. The effectiveness of the components have been iterated until the exit temperatures meet the desired values. In order to conduct local heat and mass balance equations the individual heat exchangers should be combined. For example, to calibrate the superheater, local heat, mass balance and effectiveness equations are calculated as follows:

Mass balance equation of the superheater :

$$\dot{m}_{gas,in} + \dot{m}_{HP,Steam,in} = \dot{m}_{HP,Steam,out} + \dot{m}_{gas,out} \quad [23]$$

Energy balance equation of the superheater :

$$\dot{m}_{gas,in}\, h_{gas,in} + \dot{m}_{HP,Steam,in}\, h_{HP,steam,in} = \dot{m}_{HP,Steam,out}\, h_{HP,Steam,out} + \dot{m}_{gas,out}\, h_{gas,out} \quad [24]$$

$$\text{Effectiveness} = \varepsilon = \frac{(h_{gas,in} - h_{gas,out})}{(h_{gas,in} - h_{gas,\min})} \quad [25]$$

To find the desired steam outlet temperature the following equations are used:

$$h_{HP,steam,inout} = \frac{\dot{m}_{gas,in}\, h_{gas,in} + \dot{m}_{HP,Steam,in}\, h_{HP,steam,in} - \dot{m}_{gas,out}\, h_{gas,out}}{\dot{m}_{HP,Steam,out}} \quad [26]$$

$$h_{gas,out} = h_{gas,in} + \varepsilon(h_{gas,in} - h_{gas,\min}) \quad [27]$$

$$h_{HP,steam,inout} = \frac{\dot{m}_{gas,in}\, h_{gas,in} + \dot{m}_{HP,Steam,in}\, h_{HP,steam,in} - \dot{m}_{gas,out}\,(h_{gas,in} + \varepsilon(h_{gas,in} - h_{gas,\min}))}{\dot{m}_{HP,Steam,out}} \quad [28]$$

The effectiveness value is iterated until the desired steam outlet temperature is obtained. The same procedure is repeated to model and calibrate the other HRSG components according to the acceptance test data.

In this study The HRSG model has been calibrated as follows:

6.1.2.1. Superheaters

The high pressure superheater is the first tube bundle that meets with the exhaust gas. The superheater effectiveness has been iterated until the steam outlet temperature matches the acceptance test results. The acceptance test reveals that the superheated steam was 521.7 °C at superheater outlet. Therefore, the effectiveness has been adjusted until the local heat and mass balance results match the same exit temperature value.

The low pressure superheater has been modeled and tuned similar to HP superheater. The steam outlet temperature was measured at 200.4 °C during acceptance test period. The effectiveness has been iterated until the model calculations yield the same result.

6.1.2.2. Evaporators

The similar procedure has been applied to model the HP Evaporator. The effectiveness has been adjusted until the results match with the test results. 65.4 kg/s steam production recorded as the acceptance test result. LP Evaporator has been modeled with the same manner. The acceptance test value was 13.35 kg/s steam production.

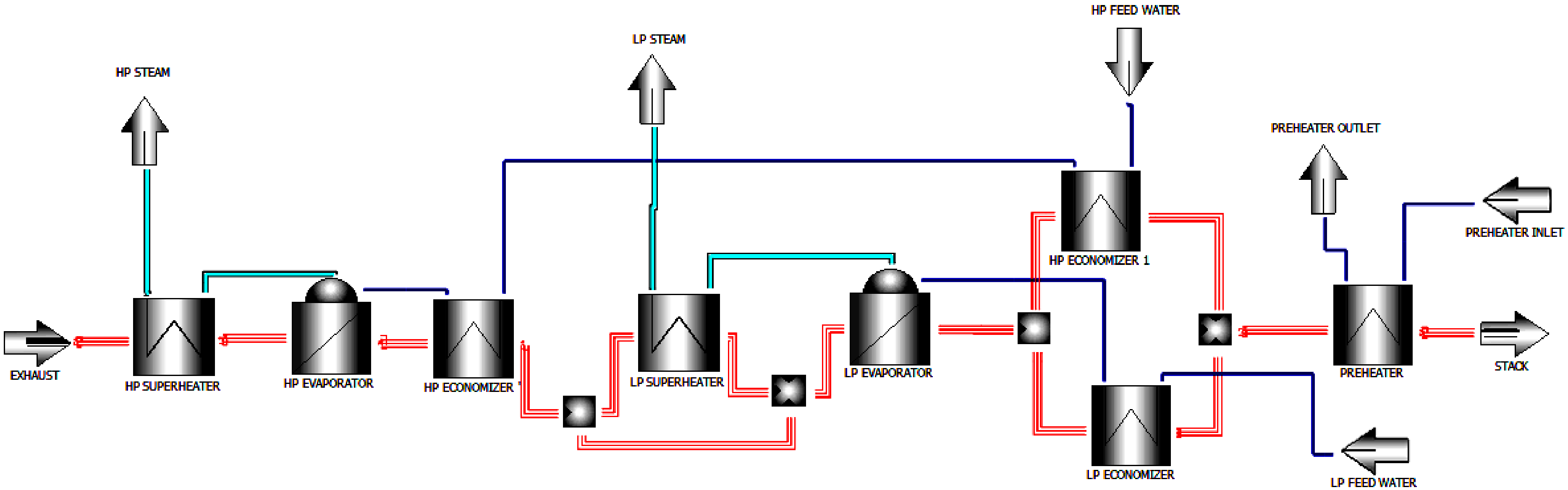

Figure 59: HRSG Model

6.1.2.3. Economizers and the Preheater

Ambarlı Power Plant has two economizers in the high pressure stage. The acceptance test data do not have any information about the economizer outlet temperatures. Therefore, the manufacturer design parameters have been used. These design information has been provided by the HRSG manufacturer to the power plant at the commissioning period. According to these information the outlet temperatures of the economizers have been adjusted as follows: Preheater Outlet: 100°C; LP Economizer Outlet: 158 °C; HP Economizer 1 Outlet: 168 °C; HP Economizer 2 Outlet: 292 °C.

6.1.3. Steam Turbine

The model calibration parameter of the steam turbine is the isentropic expansion efficiency. In the acceptance tests the steam turbine generated 175.144 MW. In this study the steam turbine isentropic efficiency has been calibrated to match the test data.

The local heat and mass balance equations for the steam turbine are derived as follows:

Mass balance:

$$\dot{m}_{HP,Steam} + \dot{m}_{LP,Steam} = \dot{m}_{exh,Steam} \quad [29]$$

Energy Balance:

$$\dot{m}_{HP,Steam}\, h_{HP,Steam} + \dot{m}_{LP,Steam}\, h_{LP,Steam} = \dot{m}_{exh,Steam}\, h_{exh,Steam} + P_{ST} \quad [30]$$

The isentropic efficiency of the turbine has been calculated by the following formula.

$$\eta_{ise,\exp} = \frac{h_3 - h_4}{h_3 - h_{4s}} \quad [31]$$

The steam turbine model has been adjusted by iterating this efficiency value.

Power output of the steam turbine is calculated by

$$P = h_3 - h_4 = \eta_{ise,\exp}(h_3 - h_{4s}) \quad [32]$$

The isentropic efficiency of the turbine has been iterated to find the desired power output.

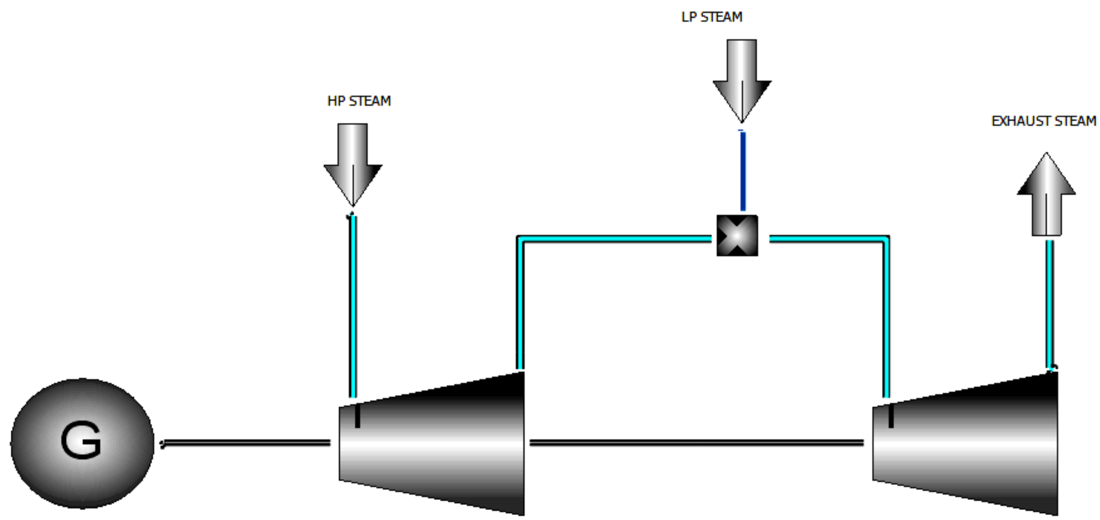

Figure 60: Steam Turbine Model

6.1.4. Condenser

The condenser is a heat exchanger. Its effectiveness has been also used as the design parameter for the condenser model. Unfortunately, there are no available acceptance test data about condenser performance. The manufacturer design value is 0.04 bar while the plant operation staff reported that the pressure was 0.042 bar when the plant was new. Therefore, 0.042 bar has been used as the reference condenser pressure. The water-steam mixture in the condenser is saturated. As a result, this condenser pressure yields a condenser outlet water temperature of around 29.8 °C. The effectiveness of the condenser has been iterated until the desired value is obtained.

The condenser heat and mass balance equations are:

Mass balance:

$$\dot{m}_{exh,Steam} + \dot{m}_{cooling,in} = \dot{m}_{Condenser,out} + \dot{m}_{Cooling,out} \qquad [33]$$

Energy Balance:

$$\dot{m}_{exh,Steam}\, h_{exh,steam} + \dot{m}_{cooling,in}\, h_{cooling,in} = \dot{m}_{Condenser,out}\, h_{Condenser,out} + \dot{m}_{cooling,out}\, h_{cooling,out} \qquad [34]$$

The condenser effectiveness is calculated as follows:

$$\text{Condenser Effectiveness} = \varepsilon = \frac{h_{cooling,out} - h_{cooling,in}}{h_{saturation} - h_{cooling,in}} \qquad [35]$$

This effectiveness has been used as the condenser design parameter in the model. To find the desired steam outlet temperature the following equations are utilized:

$$h_{condenser,out} = \frac{\dot{m}_{exh,steam}\, h_{exh,steam} + \dot{m}_{cooling,in}\, h_{cooling,in} - \dot{m}_{cooling,out}\, h_{cooling,out}}{\dot{m}_{condenser,out}} \qquad [36]$$

$$h_{cooling,out} = h_{cooling,in} + \varepsilon(h_{saturation} - h_{cooling,in}) \qquad [37]$$

$$h_{condenser,out} = \frac{\dot{m}_{exh,steam}\, h_{exh,steam} + \dot{m}_{cooling,in}\, h_{cooling,in} - \dot{m}_{cooling,out}\,(h_{cooling,in} + \varepsilon(h_{saturation} - h_{cooling,in}))}{\dot{m}_{condenser,out}} \qquad [38]$$

The effectiveness has been iterated until the desired condenser outlet temperature is obtained.

6.1.5. Other Equipments

The auxiliary equipments that are used to complete the cycle are: Feedwater Heater Tank , Pumps, Valves, Splitters, Mixers, Natural Gas Source and Exhaust Stack. These equipments have secondary effect on overall power plant performance. They have been used to arrange the mass balance of the cycle.

In this study, it has been assumed that the natural gas has 49311 kj/kg LHV. This value has been provided from the commissioning test reports.

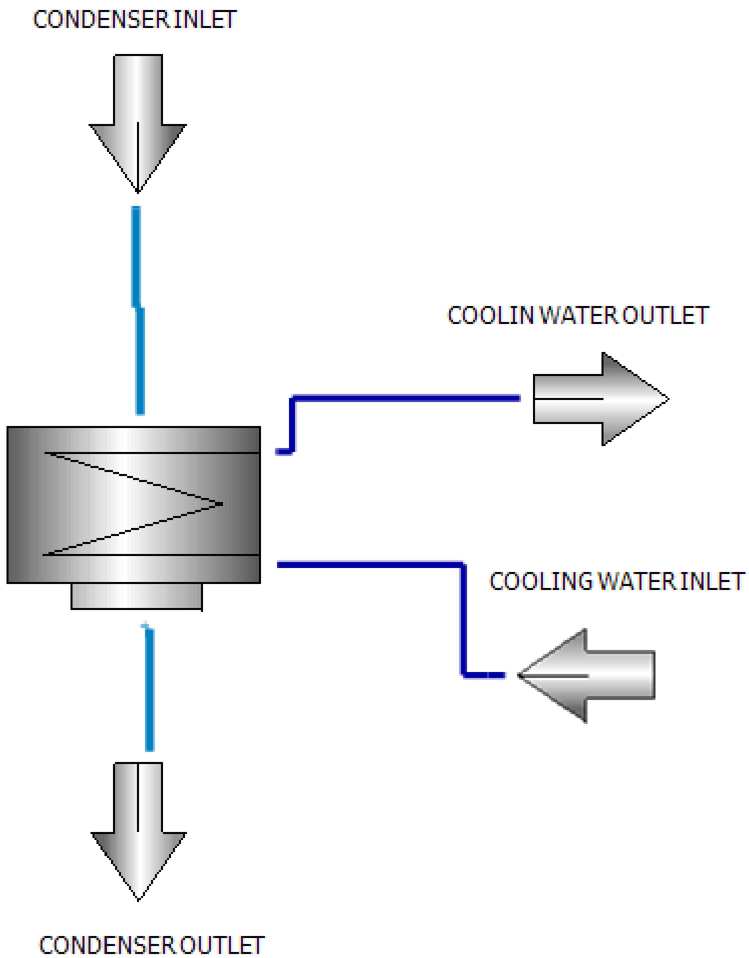

Figure 61: Condenser Model

6.2. Analysis of the Design/As-New Performance of the Power Plant

The design performance calculation procedure has been already explained. This performance has been used as the baseline performance for degradation analyses.

In the earlier chapters, description of the necessary adjustments on the component models have been presented regarding how performance calculations are validated by the acceptance test results. Once the computer model has been calibrated, the power plant model that reflects the as-new plant performance has been obtained.

The Table 7 displays the results of the as-new design performance analysis. The results match with the acceptance test data. The design parameters used to tune the component models have been presented in Table 7.

Table 7: The Design Performance and Operating Parameters of the Power Plant Model

Calculated Parameter	**Value**	**Unit**
Overall Plant Power	466.7	MW
Overall Cycle Efficiency (LHV)	52.006	%
Gas Turbine A Power	146.5	MW
Gas Turbine A Efficiency	32.65	%
Gas Turbine A Compressor Efficiency	83.03	%
Gas Turbine A Turbine Efficiency	92	%
Gas Turbine B Power	146.62	MW
Gas Turbine B Efficiency	32.68	%
Gas Turbine B Compressor Efficiency	83.03	%
Gas Turbine B Turbine Efficiency	92	%
HRSG A HP Steam Temperature	521.7	°C
HRSG A HP Steam Pressure	72.4	bar
HRSG A HP Steam Flow	65.4	kg/s
HRSG B HP Steam Temperature	521.7	°C
HRSG B HP Steam Pressure	72.4	bar
HRSG B HP Steam Flow	65.4	kg/s
HRSG A LP Steam Temperature	200.4	°C
HRSG A LP Steam Pressure	6.02	bar
HRSG A LP Steam Flow	13.35	kg/s
HRSG B LP Steam Temperature	200.4	°C
HRSG B LP Steam Pressure	6.02	bar
HRSG B LP Steam Flow	13.35	kg/s
HRSG A HP Superheater Effectiveness	0.9388	-
HRSG A HP Evaporator Effectiveness	0.9876	-
HRSG A HP Economizer 1 Effectiveness	0.9431	-
HRSG A HP Economizer 2 Effectiveness	0.9565	-
HRSG A LP Superheater Effectiveness	0.5853	-
HRSG A LP Evaporator Effectiveness	0.8498	-
HRSG A LP Economizer Effectiveness	0.7659	-

Calculated Parameter	Value	Unit
HRSG A Preheater Effectiveness	0.6556	-
HRSG B HP Superheater Effectiveness	0.9388	-
HRSG B HP Evaporator Effectiveness	0.9876	-
HRSG B HP Economizer 1 Effectiveness	0.9431	-
HRSG B HP Economizer 2 Effectiveness	0.9565	-
HRSG B LP Superheater Effectiveness	0.5853	-
HRSG B LP Evaporator Effectiveness	0.8498	-
HRSG B LP Economizer Effectiveness	0.7659	-
HRSG B Preheater Effectiveness	0.6556	-
Steam Turbine Power	175.16	MW
Steam Turbine HP Side Isentropic Efficiency	87.33	%
Steam Turbine LP Side Isentropic Efficiency	87.69	%
Condenser Effectiveness	0.618	-
Condenser Pressure	0.042	bar

6.3. Analysis of the Current Performance of the Power Plant

The current plant performance has been calculated by using actual operating measurements of the power plant. The measurements have been obtained from the distributed control system interface. Another power plant model has been prepared to calculate the current operating performance. The second law efficiency of the power plant component models have been adjusted to obtain the model that simulate the current degraded operation of the power plant.

The actual measurements that have been used to adjust the degraded model of the power plant are listed in Table 8.

Table 8: The Actual Measurements of the Operating Parameters

Operating Parameter	**Value**	**Unit**
Ambient Temperature	14	°C
Sea Water Temperature	11.25	°C
Gas Turbine A Power	133.414	MW
Gas Turbine B Power	132.047	MW
Steam Turbine Power	145	MW
HRSG A HP Steam Temperature	513	°C
HRSG A HP Steam Pressure	70.6	bar
HRSG A HP Steam Flow	59.244	kg/s
HRSG B HP Steam Temperature	513.3	°C
HRSG B HP Steam Pressure	70.5	bar
HRSG B HP Steam Flow	59.225	kg/s
HRSG A LP Steam Temperature	207	°C
HRSG A LP Steam Pressure	6.178	bar
HRSG A LP Steam Flow	14.099	kg/s
HRSG B LP Steam Temperature	202.3	°C
HRSG B LP Steam Pressure	6.207	bar
HRSG B LP Steam Flow	14.756	kg/s
Condenser Pressure	0.066	bar
HRSG A Preheater Outlet Temperature	97.5	°C
HRSG B Preheater Outlet Temperature	102.6	°C
Gas Turbine A Exhaust Temperature	527.906	°C
Gas Turbine B Exhaust Temperature	527.344	°C

Using the above information, the component models have been tuned, and the current performance of the power plant has been calculated. For example, the gas turbine design parameters like compressor and turbine isentropic efficiency have been adjusted until the model reflects the current performance. Reducing the compressor and turbine efficiency simulates the degradation of these gas turbine components.

Once the gas turbine performance matches with the actual performance, the HRSG components have been tuned to match the current performance. The original/as-new effectiveness values have been reduced to simulate the degradation of the HRSG components. The HP and LP economizers could not be evaluated because the actual measurements are not available on-site.

The condenser effectiveness has been iterated until the condenser outlet temperature matched with the current value. After the condenser degradation have been calibrated, it has been noted that the HRSG model should be tuned again. It has been observed that when the condenser deteriorates, the condensate water becomes hotter. Therefore, the steam properties change accordingly. The condensate water becomes hotter because the condensate water is saturated. As a result, when the condenser pressure increases because of the degradation, the condensate water temperature also increases accordingly.

Finally, the steam turbine isentropic efficiencies have been changed until the power output matched the actual current value.

After applying all of these steps, the current degraded model of the power plant has been obtained. Using the degraded plant model, the current performance of the power plant has been calculated. In Table 9, the calculated performance values and the design parameters that have been used to obtain the degraded model are shown.

Table 9: Degraded Power Plant Performance

Current Calculated Performance Parameters	**Value**	**Unit**
Overall Plant Power	410.41	MW
Overall Cycle Efficiency (LHV)	48.11	%
Gas Turbine A Efficiency	31.2	%
Gas Turbine A Compressor Efficiency	81.56	%
Gas Turbine A Turbine Efficiency	91.25	%
Gas Turbine B Efficiency	31.04	%
Gas Turbine B Compressor Efficiency	81.56	%
Gas Turbine B Turbine Efficiency	90.91	%
HRSG A HP Superheater Effectiveness	0.9607	-
HRSG A HP Evaporator Effectiveness	0.864	-
HRSG A HP Economizer 1 Effectiveness	0.9595	-
HRSG A HP Economizer 2 Effectiveness	0.9381	-
HRSG A LP Superheater Effectiveness	0.4897	-

Current Calculated Performance Parameters	Value	Unit
HRSG A LP Evaporator Effectiveness	0.5757	-
HRSG A LP Economizer Effectiveness	0.6915	-
HRSG A Preheater Effectiveness	0.5211	-
HRSG B HP Superheater Effectiveness	0.9608	-
HRSG B HP Evaporator Effectiveness	0.8667	-
HRSG B HP Economizer 1 Effectiveness	0.9601	-
HRSG B HP Economizer 2 Effectiveness	0.942	-
HRSG B LP Superheater Effectiveness	0.4591	-
HRSG B LP Evaporator Effectiveness	0.6256	-
HRSG B LP Economizer Effectiveness	0.6949	-
HRSG B Preheater Effectiveness	0.5782	-
HRSG A LP Superheater Effectiveness	0.5853	-
HRSG A LP Evaporator Effectiveness	0.8498	-
HRSG A LP Economizer Effectiveness	0.7659	-
HRSG A Preheater Effectiveness	0.6556	-
HRSG B HP Superheater Effectiveness	0.9388	-
HRSG B HP Evaporator Effectiveness	0.9876	-
HRSG B HP Economizer 1 Effectiveness	0.9431	-
HRSG B HP Economizer 2 Effectiveness	0.9565	-
HRSG B LP Superheater Effectiveness	0.5853	-
HRSG B LP Evaporator Effectiveness	0.8498	-
HRSG B LP Economizer Effectiveness	0.7659	-
HRSG B Preheater Effectiveness	0.6556	-
Condenser Effectiveness	0.3943	-
Steam Turbine HP Side Isentropic Efficiency	84.43	%
Steam Turbine LP Side Isentropic Efficiency	83.67	%

6.4. Analysis of the Power Plant Performance Degradation

In this section, the performance parameters of design and degraded models of the power plant have been compared to find out the magnitude of component deteriorations.

The calculated results are summarized on Table 10.

Table 10: Comparison of the Current and Design Performance

Observed Parameter	Design Value	Degraded Value	Degradation	Unit	Deviation (%)
Overall Plant Power	471.56	410.41	-61.15	MW	-12.9676
Overall Cycle Efficiency (LHV)	52.083	48.11	-3.973	%	-7.62821
Gas Turbine A Power	148.261	133.414	-14.847	MW	-10.0141
Gas Turbine A Efficiency	32.75	31.2	-1.55	%	-4.73282
Gas Turbine A Compressor Efficiency	83.03	81.56	-1.47	%	-1.77
Gas Turbine A Turbine Efficiency	92	91.25	-0.75	%	-0.815
Gas Turbine B Power	148.392	132.047	-16.345	MW	-11.0147
Gas Turbine B Efficiency	32.78	31.04	-1.74	%	-5.30811
Gas Turbine B Compressor Efficiency	83.03	81.56	-1.47	%	-1.77
Gas Turbine B Turbine Efficiency	92	90.91	-1.09	%	-1.185
Steam Turbine Power	176.5	145	-31.5	MW	-17.847
Steam Turbine HP Side Isentropic Efficiency	87.33	84.43	-2.9	%	-3.32
Steam Turbine LP Side Isentropic Efficiency	87.69	83.67	-4.02	%	-4.584
HRSG A HP Steam Temperature	521.57	513	-8.57	°C	-1.64312
HRSG A HP Steam Pressure	72.71	70.6	-2.11	bar	-2.90194
HRSG A HP Steam Flow	65.69	59.244	-6.456	kg/s	-9.82648
HRSG B HP Steam Temperature	521.58	513.3	-8.28	°C	-1.58748
HRSG B HP Steam Pressure	72.4	70.5	-2.21	bar	-3.03947
HRSG B HP Steam Flow	65.7	59.225	-6.465	kg/s	-9.84168
HRSG A LP Steam Temperature	200.56	207	6.44	°C	3.211009
HRSG A LP Steam Pressure	6.03	6.178	0.148	bar	2.454395
HRSG A LP Steam Flow	13.42	14.099	0.679	kg/s	5.059613
HRSG B LP Steam Temperature	200.56	202.3	1.74	°C	0.867571
HRSG B LP Steam Pressure	6.03	6.207	0.177	bar	2.935323
HRSG B LP Steam Flow	13.42	14.756	1.336	kg/s	9.955291
HRSG A HP Superheater Effectiveness	0.9383	0.9607	0.0224	-	2.387296
HRSG A HP Evaporator Effectiveness	0.9876	0.864	-0.1236	-	-12.5152
HRSG A HP Economizer 1 Effectiveness	0.9431	0.9595	0.0164	-	1.738946
HRSG A HP Economizer 2 Effectiveness	0.9565	0.9381	-0.0182	-	-1.90317
HRSG A LP Superheater Effectiveness	0.5845	0.4897	-0.0948	-	-16.219
HRSG A LP Evaporator Effectiveness	0.8497	0.5757	-0.274	-	-32.2467
HRSG A LP Economizer Effectiveness	0.7654	0.6915	-0.0739	-	-9.65508

Observed Parameter	Design Value	Degraded Value	Degradation	Unit	Deviation (%)
HRSG A Preheater Effectiveness	0.6525	0.5211	-0.1314	-	-20.1379
HRSG B HP Superheater Effectiveness	0.9384	0.9608	0.0224	-	2.387042
HRSG B HP Evaporator Effectiveness	0.9876	0.8667	-0.1209	-	-12.2418
HRSG B HP Economizer 1 Effectiveness	0.943	0.9601	0.0171	-	1.813362
HRSG B HP Economizer 2 Effectiveness	0.9563	0.942	-0.0143	-	-1.49535
HRSG B LP Superheater Effectiveness	0.5845	0.4591	-0.1254	-	-21.4542
HRSG B LP Evaporator Effectiveness	0.8496	0.6256	-0.224	-	-26.3653
HRSG B LP Economizer Effectiveness	0.7659	0.6949	-0.071	-	-9.27014
HRSG B Preheater Effectiveness	0.6528	0.5782	-0.0746	-	-11.4277
Condenser Effectiveness	0.618	0.3943	-0.2237	-	-36.1974
Condenser Pressure	0.042	0.066	0.024	bar	57.14286

The results reveal a substantial power degradation at the power plant. Significant degradations have been determined at condenser and HRSG components The contribution of the prime movers to the total plant power degradation is shown in the below figure:

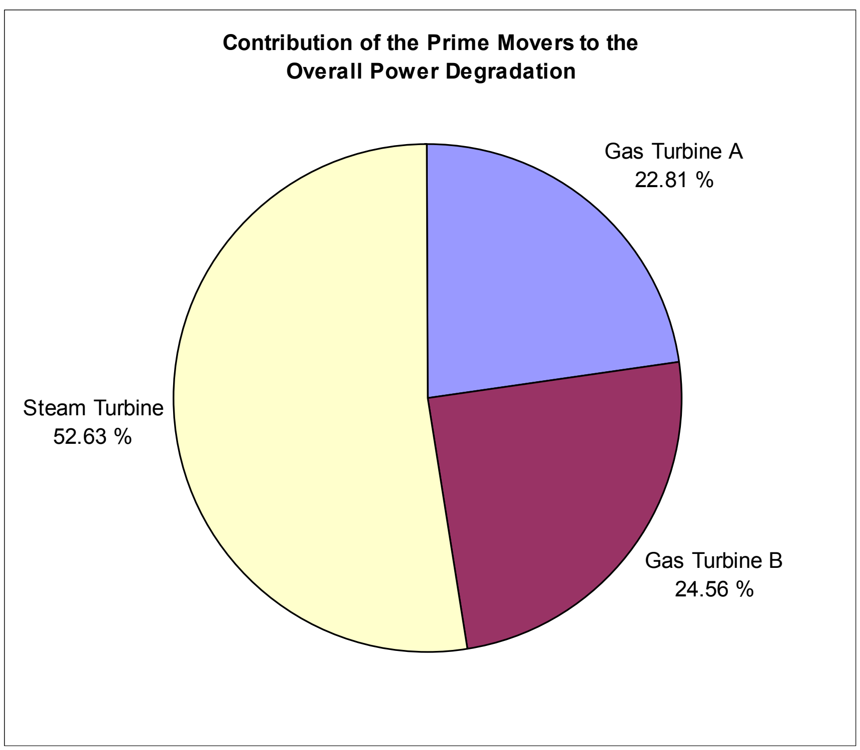

Figure 62: Contribution of the Prime Movers to the Overall Power Degradation

7. SENSITIVITY AND COST-BENEFIT ANALYSES

7.1. Sensitivity Analysis of the Individual Component Degradations on the Overall Plant Performance

A sensitivity analysis has been performed to find the effects of individual power plant components on the overall plant performance. The analyses have been conducted using the results from the model of the power plant which reflects the design performance as the benchmark. While keeping all other subsystems with as-new performance, analyses have been performed using degraded performance for only one component at a time. The second law efficiency values of each equipment on the degraded model have been implemented within the plant model calibrated for the design specs and the analysis has been performed assuming only the observed component has degraded. Running the simulation has provided the overall performance decrease if only the observed equipment would have been degraded. This analysis has been performed for every component, and the individual contributions have been calculated.

The design performance of the power plant has been selected as the baseline for these analyses. However, since operating condition and measurements were collected at different ambient conditions than the day when commissioning tests conducted and as-new performance data collected (the plant was producing 466 MW under those ambient conditions), as-new plant model has been run with the ambient conditions when the measurements were taken for the current degraded performance of the plant. In order to make a correct comparison, performance of the as-new plant has been calculated for the 14 ^{o}C air and 11.25 °C sea water temperature which recorded when degraded current performance data for the plant was collected. It has found that, if the entire plant operated in as-new condition with these ambient parameters, its output would be 471.56 MW with 52.083 % efficiency. These values have been used as the benchmark to compare the measured and calculated degraded performance under such operating conditions. After this, measured degraded performance values for individual components/subsystems like gas turbines have been introduced in this plant model to calculate the effect of individual component deteriorations on the overall plant performance.

Individual component contributions to the overall power plant performance degradation are displayed in Table 11 and Table 12.

Table 11: Component Degradation Effects on the Overall Power Output

Degraded Component	**Degraded Value of the Overall Power**	**Contribution of the Component to the Overall Power Output Degradation (MW)**	**Deviation (%)**
Gas Turbine B	450.37	-21.19	-4.4936
Gas Turbine A	451.53	-20.03	-4.2476
Condenser	461.95	-9.61	-2.0379
Steam Turbine	463.98	-7.58	-1.6074
HRSG A HP Evaporator	470.01	-1.55	-0.3287
HRSG B HP Evaporator	470.05	-1.51	-0.3202
HRSG A LP Evaporator	470.33	-1.23	-0.2608
HRSG A Preheater	470.4	-1.16	-0.246
HRSG B LP Evaporator	470.53	-1.03	-0.2184
HRSG B Preheater	470.67	-0.89	-0.1887
HRSG B LP Superheater	471.54	-0.02	-0.0042
HRSG A LP Superheater	471.55	-0.01	-0.0021

Table 12: Component Degradation Effects on the Overall Efficiency

Degraded Component	Degraded Value of the Overall Efficiency	Contribution of the Component to the Overall Efficiency Degradation	Deviation (%)
Condenser	51.022	-1.06	-2.0353
Steam Turbine	51.246	-0.836	-1.6052
Gas Turbine B	51.295	-0.787	-1.5111
Gas Turbine A	51.35	-0.732	-1.4055
HRSG A HP Evaporator	51.912	-0.17	-0.3264
HRSG B HP Evaporator	51.917	-0.165	-0.3168
HRSG A LP Evaporator	51.948	-0.134	-0.2573
HRSG A Preheater	51.956	-0.126	-0.2419
HRSG B LP Evaporator	51.97	-0.112	-0.2151
HRSG B Preheater	51.985	-0.097	-0.1862
HRSG B LP Superheater	52.0813	-0.0007	-0.0013
HRSG A LP Superheater	52.0819	-0.0001	-0.0002

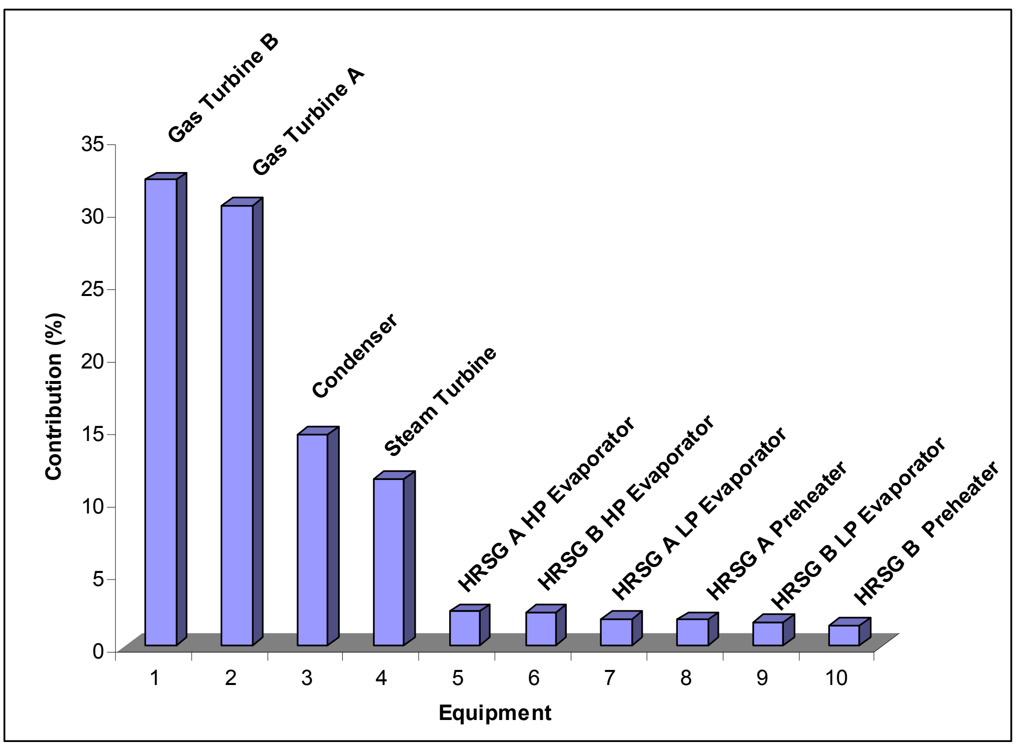

Figure 63: Contribution of the Equipments to the Overall Power Degradation

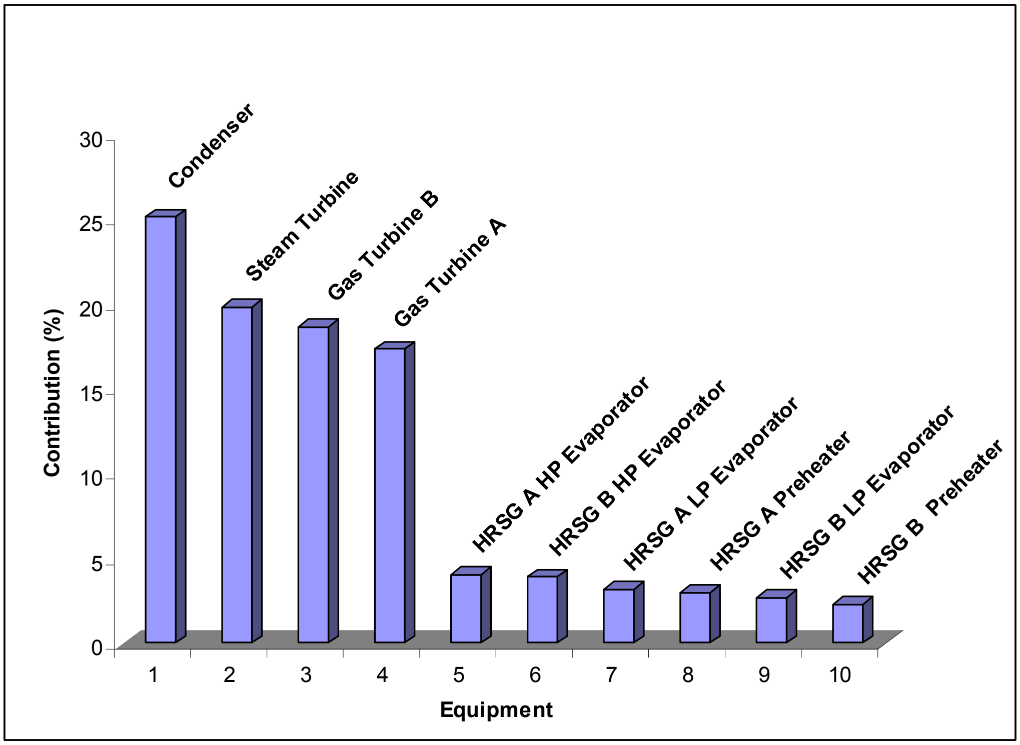

Figure 64: Contribution of the Equipments to the Overall Efficiency Degradation

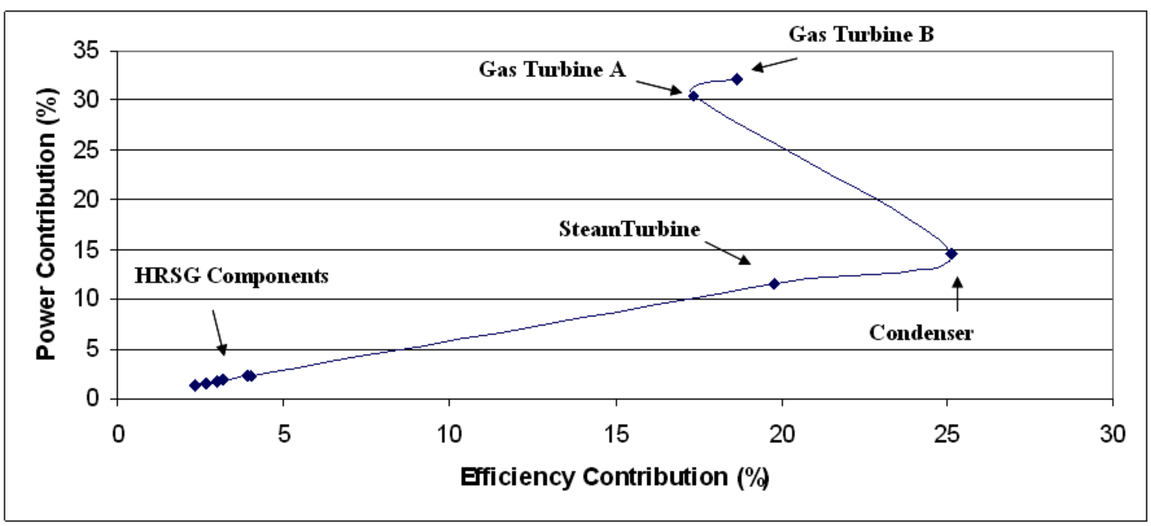

Figure 65: Pareto Chart of the Components Effect on Overall Performance

These sensitivity analyses revealed important consequences. For example; currently steam turbine is generating 145 MW while it was producing 175 MW when it was new. At a first glance, this shortfall of the power output may be judged as the steam turbine degradation. However, the analysis results indicated that individually the steam turbine has only 7.58 MW contribution to overall degradation. In fact, steam turbine can not generate the remaining 22 MW as a result of the other equipment degradations. This fact shows the difficulty of the performance evaluations. The equipments are connected to each other with tight interactions. Therefore, deviation of an equipment performance also effects the downstream component performances.

The results illustrated that the gas turbine degradations have the most detrimental effect to the overall power output of the plant. On the other hand, the condenser deteriorations affect overall efficiency more than the gas turbines. Steam turbine has also a high contribution to the overall efficiency degradation. The HRSG degradation has a secondary effect on overall performance.

7.2. Analyses of the Combined Cycle Power Plant Rehabilitation Scenarios

In this section possible rehabilitation scenarios have been evaluated to reveal how they can improve the overall power plant performance.

To perform these observations, the degraded model of the power plant has been used. The performance parameters of the rehabilitated equipments have been restored to as-new values in the model of the plant. The calculations have provided effects of the rehabilitations conducted on individual plant components. Various rehabilitation options have been analyzed with the same procedure.

7.2.1. Gas Turbine Rehabilitations

As discussed in the previous chapters, analyses showed that gas turbines have the greatest impact on the overall output performance. The results indicated that in Ambarlı Power Plant gas turbine rehabilitations should be considered before the other equipment modernizations. In the following sections the performance improvements of possible gas turbine rehabilitation projects have been analyzed. In these analyses, improvements on the overall power plant performance has been calculated assuming that only Gas Turbine A is rehabilitated.

7.2.1.1. Hot Gas Path Component Replacement

The easiest and common recovery option for a gas turbine is the replacement of the turbine blades and vanes. These components are running under the most aggressive conditions. As a result, they wear much more than the other components. Therefore, generally these components are replaced or refurbished most frequently. In this scenario it has been assumed that only the turbine section is replaced. Therefore, the degraded value of the turbine efficiency has been restored to its design value (Table 7). However, compressor section remains degraded. Considering this scenario the performance improvements have been calculated. The gas turbine power output has improved to 139.08 MW in this scenario.

Table 13: Hot Gas Path Component Replacement Effect on Overall Plant Performance

Performance Parameter of the Power Plant	Degraded Value	Improved Value	Improvement	Unit	Deviation (%)
Power	410.41	418.08	7.67	MW	1.868863
Efficiency	48.113	48.4	0.287	-	0.596512

7.2.1.2. Life Extension and Recovery of the Gas Turbine

Gas Turbine Components have limited life and the manufacturers determine component life according to their long term experience. Generally, after 100000 equivalent operating hours, comprehensive life extension activities should be carried out. A life extension project can include the replacement of the compressor and turbine rotating and stationary blades. Combustor can be also replaced or refurbished. Even the rotor should be changed or refurbished when its useful life is over.

A comprehensive life extension investment can recover the gas turbine performance to its design values. In this study, gas turbine has been assumed to recover to its design performance with 146.14 MW power output. To simulate this analysis, the degraded efficiency values of both the compressor and the turbine sections have been restored to their design values (Table 7). The analysis predicted the following overall plant performance for this scenario and the results are included in Table 14.

Table 14: Life Extension Effect on Overall Plant Performance

Performance Parameter of the Power Plant	Degraded Value	Improved Value	Improvement	Unit	Deviation (%)
Power	410.41	427.13	16.72	MW	4.073975
Efficiency	48.113	48.833	0.72	-	1.496477

7.2.1.3. Upgrading of the Gas Turbine

Gas Turbine technology is evolving continuously. Manufacturers offer new retrofit options to the plant operators. These retrofits include turbine blades made of advanced higher temperature superalloys, new burner systems, new blade designs, etc.

In this section such upgrade options have been evaluated. Analyses have been conducted assuming a complete upgrade to improve the gas turbine output to 155.8 MW. It has been assumed that this upgrade improved the turbine isentropic efficiency from 92% to 94 % and compressor efficiency from 83.03% to 86.2%. The effect of this upgrade to the overall plant performance has been calculated and displayed in Table 15.

Table 15: Upgrade Effect on Overall Plant Performance

Performance Parameter of the Power Plant	Degraded Value	Improved Value	Improvement	Unit	Deviation (%)
Power	410.41	437.38	26.97	MW	6.571477
Efficiency	48.113	49.27	1.157	-	2.404755

7.2.2. HRSG Rehabilitations

Heat Recovery Steam Generator Rehabilitation options have been evaluated in this section. Cleaning or replacing the tube bundles have been analyzed as rehabilitation options. In the performance degradation analyses, HP Evaporator and LP Evaporator have been determined as the most critical HRSG components on the overall plant performance. Therefore, their rehabilitations have been analyzed.

7.2.2.1. Cleaning of HP and LP Evaporator Tube Bundles

The model has been run with the assumption that cleaning recover the effectiveness of the evaporators. It has been assumed that the HP Evaporator Effectiveness improve to 93.52 %. The degraded value had been calculated as 86.4 % and the design value is 98.76 % (Table 10). Similarly for the LP Evaporator, it has been assumed that Effectiveness has improved to 69.04 %. Its degraded value and design values had been calculated as 57.57 % and 84.97 %, respectively. The results revealed the effects of cleaning applied on HP and LP Evaporators as presented in Table 16 and 17

Table 16: HP Evaporator Cleaning Effect on Overall Plant Performance

Performance Parameter of the Power Plant	Degraded Value	Improved Value	Improvement	Unit	Deviation (%)
Power	410.41	411.35	0.94	MW	0.229039
Efficiency	48.113	48.22	0.107	-	0.222393

Table 17: LP Evaporator Cleaning Effect on Overall Plant Performance

Performance Parameter of the Power Plant	Degraded Value	Improved Value	Improvement	Unit	Deviation (%)
Power	410.41	410.79	0.38	MW	0.09259
Efficiency	48.113	48.16	0.047	-	0.097687

7.2.2.2. Replacement of the HP and LP Evaporator Tube Bundles

The model has been run with the assumption that replacing the tube bundles recovers the evaporator performance to its design value. The LP and HP Evaporator Effectiveness have been restored to their design values (Table 7) to simulate these scenarios. The results revealed the effects of such a rehabilitation on overall plant performance as presented in Table 18 and 19:

Table 18: HP Evaporator Replacement Effect on Overall Plant Performance

Performance Parameter of the Power Plant	Degraded Value	Improved Value	Improvement	Unit	Deviation (%)
Power	410.41	412.06	1.65	MW	0.402037
Efficiency	48.113	48.307	0.194	-	0.403217

Table 19: LP Evaporator Replacement Effect on Overall Plant Performance

Performance Parameter of the Power Plant	Degraded Value	Improved Value	Improvement	Unit	Deviation (%)
Power	410.41	411.21	0.8	MW	0.194927
Efficiency	48.113	48.2	0.087	-	0.180824

7.2.3. Steam Turbine Rehabilitations

The steam turbine performance can be recovered by repairing the labyrinth seals, the blades and other steam path components. Manufacturers also offer upgrades for steam turbines.

7.2.3.1. Performance Recovery of the Steam Turbine

Table 20 summarizes the effect of the steam turbine recovery options. In this option, it has been assumed that the performance of the steam turbine has been completely restored to its design value. The isentropic efficiency of the turbine has been restored to its design value (Table 7).

Table 20: Effect of the Steam Turbine Recovery on the Overall Power Plant Performance

Performance Parameter of the Power Plant	Degraded Value	Improved Value	Improvement	Unit	Deviation (%)
Power	410.41	417.13	6.72	MW	1.637387
Efficiency	48.113	48.9	0.787	-	1.635733

7.2.3.2. Upgrade of the Steam Turbine

It has been assumed that the upgrade of the steam turbine improves the power output beyond original design value. It has been assumed that the isentropic efficiency values of HP and LP turbine improve from 87% to 91.5 % with the upgrade. The results are given in the Table 21:

Table 21: Effect of the Steam Turbine Retrofit on the Overall Power Plant Performance

Performance Parameter of the Power Plant	Degraded Value	Improved Value	Improvement	Unit	Deviation (%)
Power	410.41	422	11.59	MW	2.824005
Efficiency	48.113	49.47	1.357	-	2.820444

7.2.4. Condenser Rehabilitations

The degradation analyses indicated that condenser has great effect on overall performance. Therefore, considerable performance improvements can be achieved by condenser rehabilitations. Cleaning the condenser, or complete overhaul of the condenser that recovers the condenser performance to its design values have been considered as rehabilitation options.

7.2.4.1. Cleaning the Condenser Tubes

Cleaning the condenser can remove some of the accumulated deposits. The degraded and design effectiveness value had been calculated as 39.43 % and 61.8 % (Table 10) respectively. It has been assumed that the cleaning improves the condenser effectiveness to 49.01 %. This can improve the condenser performance:

Table 22: Effect of the Condenser Cleaning on the Overall Power Plant Performance

Performance Parameter of the Power Plant	Degraded Value	Improved Value	Improvement	Unit	Deviation (%)
Power	410.41	414.54	4.13	MW	1.006311
Efficiency	48.113	48.6	0.487	-	1.0122

7.2.4.2. Complete Recovery of the Vacuum System and the Condenser

It has been assumed that the condenser performance recovers completely by overhauling the vacuum system and replacing the condenser tubes. The degraded condenser value has been restored to its design value (Table 7) to analyze this rehabilitation option. The results are shown in the Table 23:

Table 23: Effect of the Condenser Recovery on the Overall Power Plant Performance

Performance Parameter of the Power Plant	**Degraded Value**	**Improved Value**	**Improvement**	**Unit**	**Deviation (%)**
Power	410.41	417.68	7.27	MW	1.771399
Efficiency	48.113	48.956	0.843	-	1.752125

7.2.5. Sensitivity Analysis of the Rehabilitation Projects

Sensitivity analysis of the rehabilitation improvements have been carried out. The following tables and figures display the power output and efficiency improvements of the analyzed rehabilitation projects.

Table 24: Effects of Rehabilitations on the Overall Power Plant Power Output

Rehabilitation	**Improved Value**	**Improvement**	**Deviation (%)**
GT Upgrade	437.38	26.97	6.57148
GT Life Extension	427.13	16.72	4.07398
ST Upgrade	422	11.59	2.82401
GT HGP	418.08	7.67	1.86886
Condenser Recovery	417.68	7.27	1.7714
ST Recovery	417.13	6.72	1.63739
Condenser Cleaning	414.54	4.13	1.00631
HP Evaporator Replacement	412.06	1.65	0.40204
HP Evaporator Cleaning	411.35	0.94	0.22904
LP Evaporator Replacement	411.21	0.8	0.19493
LP Evaporator Cleaning	410.79	0.38	0.09259

Table 25: Effects of Rehabilitation on the Overall Power Plant Efficiency

Rehabilitation	Improved Value	Improvement	Deviation (%)
ST Upgrade	49.47	1.357	2.82044
GT Upgrade	49.27	1.157	2.40476
Condenser Recovery	48.956	0.843	1.75213
ST Recovery	48.9	0.787	1.63573
GT Life Extension	48.833	0.72	1.49648
Condenser Cleaning	48.6	0.487	1.0122
GT HGP	48.4	0.287	0.59651
HP Evaporator Replacement	48.307	0.194	0.40322
HP Evaporator Cleaning	48.22	0.107	0.22239
LP Evaporator Replacement	48.2	0.087	0.18082
LP Evaporator Cleaning	48.16	0.047	0.09769

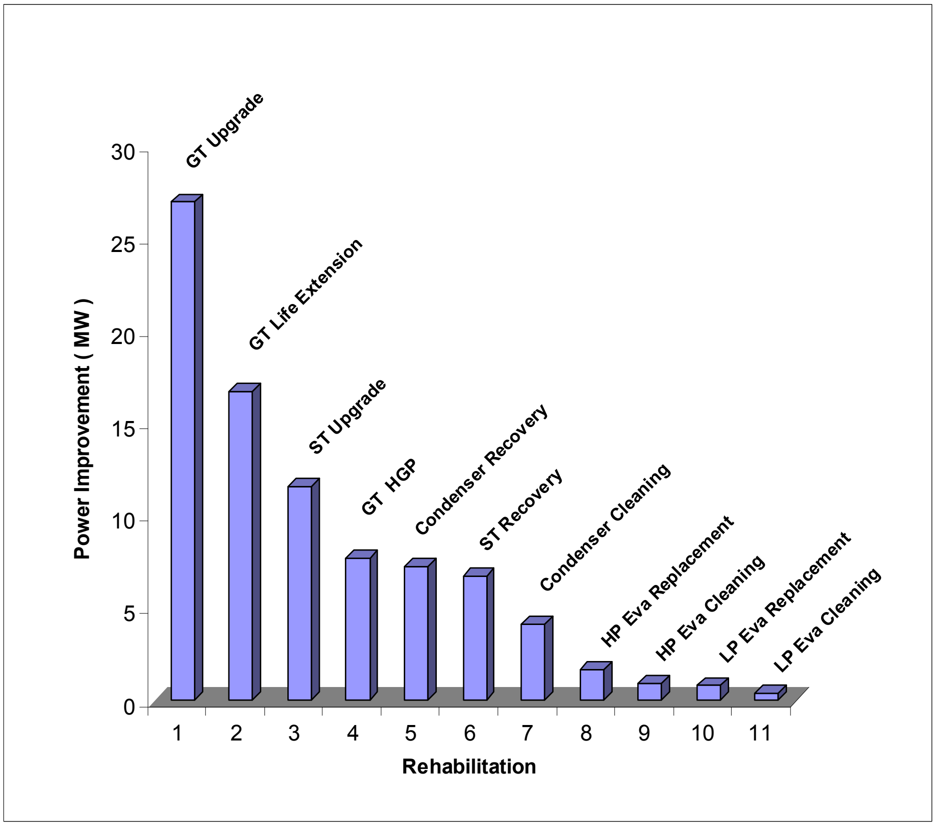

Figure 66: Power Output Improvements with Rehabilitation Projects

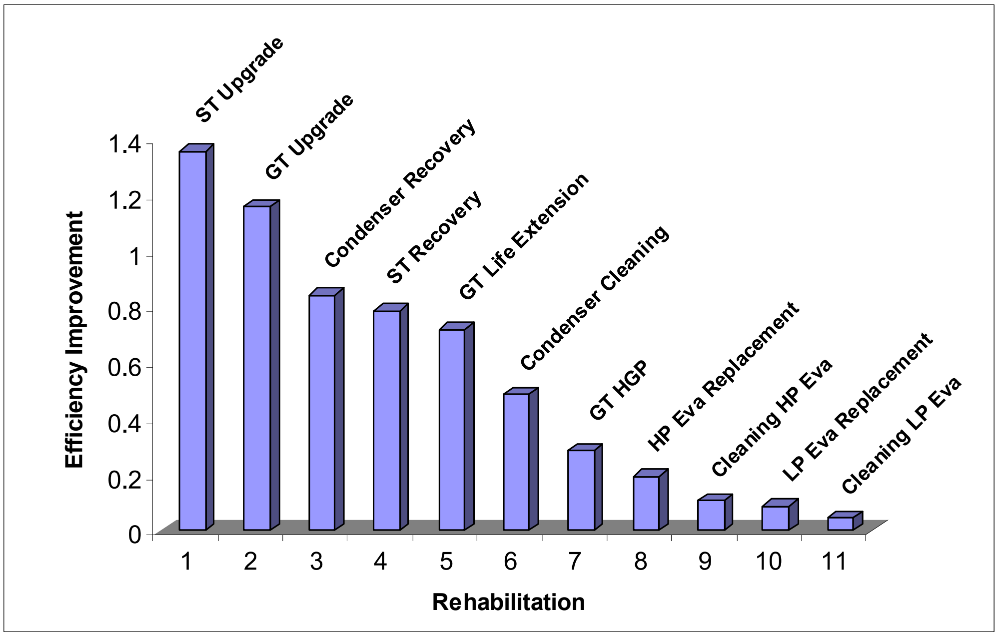

Figure 67: Efficiency Improvements with Rehabilitation Projects

7.3. Cost Benefit Analysis

It has been observed that the rehabilitations improve the performance of the power plant. However, this information by itself is not sufficient to decide or to plan a rehabilitation project. The operators also need to know whether the rehabilitations can justify their cost, and the estimated payback time. As a result, the benefits and the costs of the rehabilitation projects should be compared before a rehabilitation investment.

Cost-benefit analyses have been performed using the information obtained earlier. The rehabilitation costs depend on extent of the rehabilitation and the manufacturer. In this study approximate rehabilitation investment costs are used from the market. It has been assumed that average electricity whole sale price is 100 \$/MWh, and natural gas cost is 5 \$/GJ. It has been assumed that the power plant is operating 8000 hours per year at base load. These are typical current values.

The hourly benefit has been calculated with the following equation:

$$HB = P_{imp} x C_e - \dot{m}_{add,NG} \, x LHV_{NG} x C_{NG} \qquad [39]$$

where;

HB = Hourly Benefit,

P_{imp} = Power Improvement

C_e = Electricity Price

$\dot{m}_{add,NG}$ = Additional fuel consumption

LHV_{NG} = Lower Heating Value of Natural Gas

C_{NG} = Natural Gas Price

Here Additional Fuel Consumption is calculated as follows:

$$\dot{m}_{add,NG} = \dot{m}_{aft,Rhb} - \dot{m}_{bef,Rhb} \qquad [40]$$

Where;

$\dot{m}_{aft,Rhb}$ = Fuel consumption after the rehabilitation

$\dot{m}_{bef,Rhb}$ = Fuel consumption before the rehabilitation

These two term is obtained with the following formulas:

$$\dot{m}_{aft,Rhb} = \frac{\dot{W}_{aft,Rhb}}{\eta_{aft,Rhb} x LHV_{NG}} \qquad [41]$$

$$\dot{m}_{bef,Rhb} = \frac{\dot{W}_{bef,Rhb}}{\eta_{bef,Rhb} x LHV_{NG}} \qquad [42]$$

where;

$\dot{W}_{aft,Rhb}$ = Power output after the rehabilitation

$\dot{W}_{bef,Rhb}$ = Power output before the rehabilitation

$\eta_{aft,Rhb}$ = Efficiency after the rehabilitation

$\eta_{bef,Rhb}$ = Efficiency before the rehabilitation

The hourly benefits have been multiplied by the annual operating hours to find the annual benefits. Detailed economic analyses can be performed by calculating the net present value of a determined operation period. Additional parameters such as degradation factors can be added for long-time economic analyses. Internal rate of return and pay back time can be also calculated for rehabilitation investment analyses. These detailed economic calculations are beyond this study. In this study, the net present value of the annual benefits have been calculated to compare the rehabilitation scenarios.

The results are given in Table 26:

Table 26: Cost Benefit Analysis of the Rehabilitation Projects

Rehabilitation Project	**Power Gain (MW)**	**Efficiency Gain**	**Additional Fuel (kg/s)**	**Hourly Benefit ($)**	**Annual Benefit ($)**	**Investment Cost ($)**	**Cost-Benefit Ratio**
Gas Turbine Upgrade	26.97	1.157	0.70386	2269.7	18,157,399	20,000,000	0.90787
Gas Turbine Life Extension	16.72	0.72	0.4393	1405.3	11,242,359	15,000,000	0.74949
Steam Turbine Upgrade	11.59	1.357	0	1158.6	9,269,090	10,000,000	0.92691
Condenser Recovery	7.27	0.843	0	725.01	5,800,085	5,000,000	1.16002
Steam Turbine Recovery	6.72	0.787	0	671.83	5,374,632	5,000,000	1.07493
Gas Turbine Hot Gas Path Change	7.67	0.287	0.21879	634.17	5,073,331	8,000,000	0.63417
Condenser Cleaning	4.13	0.487	0	413.61	3,308,899	200,000	16.5445
Replacement of the HP tubes	1.65	0.194	0	165.12	1,320,988	2,000,000	0.66049
Cleaning the HP tubes	0.94	0.107	0	93.304	746,428	20,000	37.3214
Replacement of the LP tubes	0.8	0.087	0	78.522	628,173	1,500,000	0.41878
Cleaning the LP tubes	0.38	0.047	0	38.535	308,278	20,000	15.4139

Using the above procedure, each equipment degradation costs due to production losses, have been calculated. Here, instead of performance improvement parameters, degraded performance values have been used on above formulations. The results are displayed on Table 27.

Table 27: Costs of Equipment Degradation

Degraded Equipment	Hourly Cost ($)	Annual Cost ($)
Gas Turbine B	2,119	16,951,985
Gas Turbine A	2,003	16,023,986
Condenser	961	7,688,114
Steam Turbine	758	6,063,989
HRSG A HP Evaporator	155	1,239,989
HRSG B HP Evaporator	151	1,207,989
HRSG A LP Evaporator	123	983,989
HRSG A Preheater	116	927,989
HRSG B LP Evaporator	103	823,989
HRSG B Preheater	89	711,989
HRSG B LP Superheater	2	15,989
HRSG A LP Superheater	1	7,989

It has been found that the block have almost 60 MW degradation. It has been calculated that this deterioration translates to 48,919,982 $ annual production loss.

8. CONCLUSION

The Power Industry has been generating electricity for more than a century. As a result, both the industry professionals and the academia have great experience and knowledge about the power plant operation issues. The theoretical knowledge is matured and well understood. Although there are many studies concerning thermodynamic cycles and theoretical performance of various combined cycle power plant configurations, publications about actual combined cycle power plant hardware performance and their observed degradation data remain rather limited in the open literature. In this study, performance and degradation evaluation study of an actual combined cycle power plant has been presented. Typically, each power plant has unique features, specs and hardware combinations that are customized to site conditions. As a case study, the 1350 MW Ambarli Combined Cycle Power Plant, which has been in service more than 20 years has been completely modeled and analyzed. The calculated design performance is validated and calibrated by commissioning test values of the power plant. Using actual plant operation data current operating performance of the plant has been calculated. Degradation analysis of the power plant has been carried out by comparing the design and current performance results. A sensitivity analysis has been performed to illustrate the components' and subsystems' contributions to the overall power plant performance. Possible rehabilitation scenarios for the power plant have been analyzed. Finally, cost-benefit analyses of various rehabilitation options have been performed to find out the most effective rehabilitation combinations. The results of the analysis performed can be summarized as follows:

- In the Ambarlı Power Plant, the degradations in gas turbines have the biggest contribution to the overall plant performance degradation.
- The steam turbine generation has reduced almost 30 MW since the commissioning period. However, it has been calculated that actually steam turbine component degradation is individually around 7 MW. The remaining portion of the steam turbine performance degradation is due to other equipment deteriorations.

- The gas turbine rehabilitations provide the biggest benefit. However, gas turbine rehabilitations are very expensive. Therefore, among other plant subsystems they do not have the highest cost-benefit ratio. Yet, it is observed that the gas turbine rehabilitations have the potential to pay back their cost in almost one year.
- The condenser recovery would improve the overall efficiency much more than the other component rehabilitations.
- High power output improvements can be achieved mainly by gas turbine rehabilitations.
- The cost-benefit analysis revealed that cleaning the HRSG and Condenser tubes should be done very frequently as they have the highest cost-benefit ratio.
- The HRSG components have minor effect on performance than the other main equipments.
- The cost benefit analysis reveal that all of the evaluated rehabilitation projects have favorable cost-benefit ratios which means major power plant component rehabilitations are all generally feasible investments.
- The presented model is very useful tool to guide operators to plan the rehabilitations. If they have limited budget they may select the rehabilitations with high cost-benefit ratios. If they their main concern is the power output boost they can choose the gas turbine rehabilitations. If their concern is efficiency, they should consider to recover the condenser.
- The case study results indicate that the Ambarlı power plant has almost 60 MW degradation per group/block. With current average prices this degradation loss translates to 48,919,982 $ per year.
- The performance and degradation analyses indicate that the operators can get great benefits by rehabilitations. The calculated benefits also reflect the magnitude of the production loss of the equipment. It has been observed that the power plant have great production loss because of degradation.
- The results illustrate that degradations in gas turbines, steam turbines and condensers have the greatest impact on the costs. The results show the importance of rehabilitation projects.

In short, evaluating and modeling performance of a power plant is not easy. Performance concepts should be well understood. A comprehensive model of the power plant should be developed to better study the performance. This case study on actual hardware and plant systems revealed that degradation of an old power plant can cause high production losses. However, it is possible to recover the performance with appropriate rehabilitation projects. Although these projects seem to be very expensive, the study showed that their costs are easily justified.

As a last word, apart from costs and benefits the environmental effects should be also considered. It is clear that a degraded power plant will pollute the environment much more than efficient power plants. Although we can not avoid the polluting emissions, it is in our hands to minimize the impact. Therefore, the rehabilitations should be also considered for a clean environment.

9. REFERENCES

[1] Basaran, T., “Global Energy, Environment, Economic System and Hydrogen”, Hydrogen Energy Course Project Report, Sabancı University, Istanbul, (2009)

[2] Zhao, Y., “An Integrated Framework For Gas Turbine Based Power Plant Operational Modeling and Optimization”, Georgia Institute of Technology, PHD Thesis, Atlanta, USA, (2005)

[3] Nessler, H., Preiss R., Eisenkolb, P., “Developments in HRSG Technology”, The 7th Annual Industrial & Power Gas Turbine O & M Conference, Birmingham, UK, (2001).

[4] “Natural Gas Combined Cycle Gas Turbine Power Plants”, Northwest Power Planning Council, New Resource Characterization for the Fifth Power Plan, (2002).

[5] Chase, D.L., Kehoe, P.T., “GE Combined-Cycle Product Line and Performance”, GE Power Systems, New York, USA.

[6] Franco, A., Casarosa, C., “On Some Perspectives for Increasing the Efficiency of Combined Cycle Power Plants”, Applied Thermal Engineering, Volume 22, Pages 1501-1518, (2002).

[7] Ishikawa,M., Terauchi,M., Komori, T., Yasuraoka, J., “Development of High Efficiency Gas Turbine Combined Cycle Power Plant”, Mitsubishi Heavy Industries Technical Review, Volume 45, Number 1, (2008).

[8] Arnas, A.Ö., Boettner, D.D., Norberg, S.A., Tamm, G., Whipple, J.R., “On the Teaching of Performance Evaluation and Assessment of a Combined Cycle Cogeneration System”, Journal of Energy Resources Technology, Volume 131, New York, USA, (2009).

[9] “BP Statistical Review of World Energy”, London, UK, (2008).

[10] Watanabe, K., Arimura, H., Akagi, K., Sakuma, H., “Modernization and Upgrade Programs for Mitsubishi Heavy-Duty Gas Turbines”, International Gas Turbine Congress, Tokyo, Japan, (2003).

[11] Gülen, S.C., Griffin, P.R., "Real-Time On-Line Performance Diagnostics of Heavy-Duty Industrial Gas Turbines", International Gas Turbine & Aeroderivative Congress & Exhibition, Munich, Germany, (2000).

[12] Zwebek, A., Pilidis, P., "Degradation Effects on Combined Cycle Power Plant Performance-Part I: Gas Turbine Cycle Component Degradation Effects", Journal of Engineering for Gas Turbines and Power, Volume 125, UK, (2003).

[13] Kehlhofer, R., "Combined Cycle Gas & Steam Turbine Power Plants", Penwell Publishing Company, Oklahama, USA, (1997).

[14] Devki Energy Consultancy, "Best Practice Manual: Cogeneration", Vadodara, India, (2006).

[15] Polyzakis, A.L., Koroneas, C., Xydis, G., "Optimum Gas Turbine Cycle for Combined Cycle Power Plants", Energy Conversion and Management, Volume 49, (2008).

[16] Bassily, A.M., "Modeling, Numerical Optimization and Irreversibility Reduction of a Dual-Pressure Reheat Combined Cycle", Applied Energy, Volume 81, (2005).

[17] Karthikeyan R., Hussein, M.A., Reddy, B.V., Nag, P.K., "Performance Simulation of Heat Recovery Steam Generator", International Journal of Energy Research, Volume 22, (1998).

[18] Frangopoulos, C.A., "Synthesis and Operation of Thermal Systems by the Thermoeconomic Functional Approach", Journal of Engineering for Gas Turbines and Power, Volume 114, (1992).

[19] Tsatsaronis, G., "Application of Thermoeconomics to the Design and Synthesis of Energy Plants", Encyclopedia of Life Support Systems.

[20] Von Spakovsky, M.R., Evans, R.B., "The Design and Performance Optimization of Thermal System Components", Journal of Energy Resources Technology, Volume 111, (1989).

[21] Von Spakovsky. M.R., Pelster, S., Favrat, D., "The Thermoeconomic and Environomic Modeling and Optimization of the Synthesis, Design, and Operation of Combined Cycles With Advanced Options ", Journal of Engineering for Gas Turbines and Power, Volume 123, (2001).

[22] Abusoglu, A., Kanoglu M., “Exergoeconomic Analysis and Optimization of Combined Heat and Power Production: A Review”, Renewable and Sustainable Energy Reviews, Volume 13, (2009).

[23] Koch, C., Cziesla F., Tsatsaronis G., “Optimization of Combined Cycle Power Plants Using Evolutionary Algorithms”, Chemical Engineering and Processing, Volume 46, (2007).

[24] Khaliq, A., Kaushik, S.C., “Second Law Based Thermodynamic Analysis of Brayton/Rankine Combined Power Cycle With Reheat”, Applied Energy, Volume 78, (2004).

[25] Li, H., Marechal, F., Burer, M., Faurat, D., “Multi-Objective Optimization of an Advanced Combined Cycle Power Plant Including CO_2 Separation Options”, Energy, Volume 31, (2006).

[26] Franco, A., Russo, A., “Combined Cycle Plant Efficiency Increase Based on the Optimization of the Heat Recovery Steam Generator Operating Parameters”, International Journal of Thermal Sciences, Volume 41, (2002).

[27] Kotowicz S., Bartela, L., “The Influence of Economic Parameters on the Optimal Values of Design Variables of a Combined Cycle Plant”, Energy, Volume 35, (2010).

[28] Valdes, M., Rovira, A., Duran, M.D., “Influence of the Heat Recovery Steam Generator Design Parameters on the Thermoeconomic Performances of Combined Cycle Gas Turbine Power Plants”, International Journal of Energy Research, Volume 28, Pages 1243-1254, (2004).

[29] Mohagheghi, M., Shayegan, J., “Thermodynamic Optimization of Design Variables and Heat Exchangers Layout in HRSGs for CCGT, Using Genetic Algorithm”, Applied Thermal Engineering, Volume 29, Pages 290-299, (2009).

[30] Cihan A., Hacihafizogullari, O., Kahveci, K., “Energy-Exergy Analysis and Modernization Suggestions for a Combined Cycle Power Plant”, International Journal of Energy Resources, Volume 30, (2006).

[31] Deng, C.S., Chuang, C.C., “Engineering Design and Exergy Analyses For Combustion Gas Turbine Based Power Generation System”, Energy, Volume 29, (2004).

[32] Ameri, M., Ahmadi, P., Khanmohammadi, S., "Exergy Analysis of a 420 MW Combined Cycle Power Plant", International Journal of Energy Research, Volume 32, (2007).

[33] Attala, L., Facchini, B., Ferrara, G., "Thermoeconomic Optimization Method as Design Tool in Gas-Steam Combined Plant Realization", Energy Conversion and Management, Volume 42, Issue 18, (2001).

[34] Vieira, L.S., Matt, C.F., Guedes, V.G., Cruz, M.E., Castelloes, F.V., "Maximization of the Profit of a Complex Combined Cycle Cogeneration Plant Using a Professional Process Simulator", Journal of Engineering for Gas Turbines and Power, Volume 132, (2010).

[35] Arrieta, F.R.P., Lora, E.E.S., "Influence of Ambient Temperature on Combined Cycle Power Plant Performance", Applied Energy, Volume 80, (2005).

[36] Unver, U., Kilic, M., "Second Law Based Thermoeconomic Analysis of Combined Cycle Power Plants Considering the Effects of Environmental Temperature and Load Variations", International Journal of Energy Research, Volume 31, (2007).

[37] Wu, C., "Intelligent Computer Aided Sensitivity Analysis of a Multi-Stage Brayton/Rankine Combined Cycle", Energy Conversion and Management, Volume 40, (1999).

[38] Chuang, C.C., Sue, D.C., "Performance Effects of Combined Cycle Power Plant with Variable Condenser Pressure and Loading", Energy, Volume 30, (2005).

[39] Bhargava, R., Bianchi, M., Melino, F., Peretto, A., "Parametric Analysis of Combined Cycles Equippet with Inlet Fogging", Journal of Engineering for Gas Turbines and Power, Volume 128, (2006).

[40] Boonnasa, S., Namprakai, P., "Sensitivity Analysis for the Capacity Improvement of a Combined Cycle Power Plant (100-600 MW)", Applied Thermal Engineering, (2008).

[41] Zwebek, A., Pilidis, P., "Degradation Effects on Combined Cycle Power Plant Performance-Part II: Steam Turbine Cycle Component Degradation Effects", Journal of Engineering for Gas Turbines and Power, Volume 125, UK, (2003).

[42] Zwebek, A., Pilidis, P., "Degradation Effects on Combined Cycle Power Plant Performance-Part III: Gas and Steam Turbine Component Degradation Effects", Journal of Engineering for Gas Turbines and Power, Volume 126, UK, (2004).

[43] Kurz, R., Brun, K., Wollie, M., "Degradation Effects on Industrial Gas Turbines ", Journal of Engineering for Gas Turbines and Power, Volume 131, (2009).

[44] Valero, A., Verda, V., Serra, L., "Thermoeconomic Diagnosis: Zooming Strategy Applied to Highly Complex Energy Systems, Part 1: Detection and Localization of Anomalies", Journal of Energy Resources Technology, Volume 127, (2005).

[45] Valero, A., Verda, V., Serra, L., "Thermoeconomic Diagnosis: Zooming Strategy Applied to Highly Complex Energy Systems, Part 2: On the Choice of the Productive Structure", Journal of Energy Resources Technology, Volume 127, (2005).

[46] Valero, A., Correas L., Zaleta, A., Lazzaretto, A., Verda, V., Reini, M., Rangel, V., "On the Thermoeconomic Approach to the Diagnosis of Energy System Malfunctions, Part 1: TADEUS Problem", Energy, Volume 29, (2004).

[47] Valero, A., Correas L., Zaleta, A., Lazzaretto, A., Verda, V., Reini, M., Rangel, V., "On the Thermoeconomic Approach to the Diagnosis of Energy System Malfunctions, Part 2: Malfunction Definitions and Assessment", Energy, Volume 29, (2004).

[48] Lazzaretto, A., Toffolo, A., "A Critical Review of the Thermoeconomic Diagnosis Methodologies for the Location of Causes of Malfunctions in Energy Systems", Journal of Energy Resources Technology, Volume 128, (2006).

[49] Mathioudakis, K., Stamatis, A., Bonataki, E., "Allocating the Causes of Performance Deterioration in Combined Cycle Gas Turbine Plants", Journal of Engineering for Gas Turbines and Power, Volume 124, (2002).

[50] Valero, A., Serra, L., Zaleta, A., Lazzaretto, A., Verda, V., Reini, M., Rangel, V., Toffolo, A., Tacconi, R., Donatini, F., Trucato, E., "On the Thermoeconomic Approach to the Diagnosis of Energy System Malfunctions, Part 3: Approaches to the Diagnosis Problem", Proceedings of ECOS 2003 Copenhagen, Denmark (2003).

[51] Toffolo, A., Lazzaretto, A., “A New Thermoeconomic Method for the Location of Causes of Malfunctions in Energy Systems”, Journal of Energy Resources Technology, Volume 129, (2007).

[52] Verda, V., Serra, L., Valero, A., “The effects of the Control System on the Thermoeconomic Diagnosis of a Power Plant”, Energy, Volume 29, (2004).

[53] Correas, L., “On the Thermoeconomic Approach to the Diagnosis of Energy System Malfunctions Suitability to Real-Time Monitoring”, International Journal of Thermodynamics, Volume 7 (2004).

[54] Armor, A.F., Hesler, S.H., “Productivity Improvement for Fossil Steam Power Plants”, Electric Power Research Institute, California, USA, (2006).

[55] Heyena, G., Kalitventze, B., “A Comparison of Advanced Thermal Cycles Suitable for Upgrading Existing Power Plant”, Applied Thermal Engineering, Volume 19, Pages 227-237, (1999).

[56] “Improving Steam System Performance”, Office of Industrial Technologies, Energy Efficiency and Renewable Energy, US Department of Energy, Lawrence National Laboratory, Washington D.C., USA

[57] “Heat Rate Improvement Reference Manual”, Electric Power Research Institute, Charlotte, North Carolina, (1998)

[58] Maghon, H., Kreyenberg, O., Thun, O., “Proven Upgrade of Siemens SGT5-4000F (V94.3A) at Mainz-Wiesbaden”, Germany, (2005)

[59] Oakey, J.E., Simms, N.J., Allen, D.H., “Environmental Degradation Issues in Gas Turbines and Their Relevance to Plant Life Extension-R&D Initiatives”, OMMI, Volume 3, Issue 1, UK, (2004).

[60] Carlson, A., Friedman, J.R., “Performance Test Methods to Determine the Benefits of Various Gas Turbine Modernizations and Upgrades”, Proceedings of PWR:ASME Power, Atlanta, USA, (2006).

[61] Henkel, N., Schmid, E., Gobrecht, E., “Operational Flexibility Enhancements of Combined Cycle Power Plants”, Power-Gen Asia, (2008).

[62] Gay, R.R., Palmer, C.A., Erbes, M.R., “Power Plant Performance Monitoring”, R-Squared Publishing, California, USA.

[63] Tsou, J.L., “Demonstration of EPRI Heat Rate Guidelines at Southern California Edison Ormond Beach Unit 2”, Electric Power Research Institute Technical Report, California, USA, (1992).

[64] “DOE Fundamentals Handbook: Thermodynamics, Heat Transfer and Fluid Flow Volume 1”, U.S. Department of Energy, Washington D.C. (1992).

[65] Çengel, Y.A., Boles, M.A., “Thermodynamics: An Engineering Approach”, Mc-Graw Hill, (1999).

[66] Giampolo, T., “Gas Turbine Handbook: Principles and Practices: 3rd edition”, The Fairmont Press, Lilburn, USA (2006).

[67] Jonas, O., Machemer, L., “Steam Turbine Corrrosion and Deposits Problems and Solutions”.

[68] Ishikawa, M., Uchida, S., Okada, K., Hiramoto, K., XiuFu, X., "Mitsubishi F Series Gas Turbine Combined Cycle Operating Experience”, Technical Bulletin, Japan.

[69] Petek, J., Hamilton, P., “Performance Monitoring for Gas Turbines”, GE Energy Magazine, ORBIT, (2005).

[70] Yui, Y.V., Yee, Y.V., Singh, P., “Combined Cycle Power Plant Performance Degradation”, Power-Gen, (1998).

[71] Mamat, Z.A., “GT Compressor Water Washing Practices & Effectiveness for TNB’s Combined Cycle Plants”

[72] Loud. R.L., Slaterpryce, A.A., “Gas Turbine Inlet Air Treatment”, GE Power, New York, USA.

[73] “Technical Guidelines: Generator Efficiency Standards”, Australian Greenhouse Office in the Department of Environment and Heritage, Canberra, Australia, (2006)

[74] Schofield, P., “Steam Turbine Sustained Efficiency”, GE Power, New York, USA

[75] Albert, P., “Steam Turbine Thermal Evaluation and Assessment”, GE Power Systems, New York, USA.

[76] Grace, D., “Steam Turbine and Generator Designs for Combined Cycle Applications”, Electric Power Research Institute Technical Report, California, USA, (2003).

[77] Hesler, S., “Replacement Interstage Seals for Steam Turbines”, Electric Power Research Institute Technical Report, California, USA, (2005).

[78] Starr, F., “Background to the Design of HRSG Systems and Implications for CCGT Plant Cycling”, OMMI, Volume 2, Issue 1, UK, (2003).

[79] “Performance Degradation of HRSGs; Technical Bulletin”, Vogt Power International, (1999).

[80] Brummel, H., LeMieux, D., Voigt, M., Zombo, P., “Online Monitoring of Gas Turbine Power Plants”, Siemens, USA, (2005).