ELECTRIC POWER RESEARCH INSTITUTE

Impact of Cycling on the Operation and Maintenance Cost of Conventional and Combined-Cycle Power Plants

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Impact of Cycling on the Operation and Maintenance Cost of Conventional and Combined-Cycle Power Plants

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EPRI Project Manager
R. Roberts

ELECTRIC POWER RESEARCH INSTITUTE
3420 Hillview Avenue, Palo Alto, California 94304-1338 ▪ PO Box 10412, Palo Alto, California 94303-0813 ▪ USA
800.313.3774 ▪ 650.855.2121 ▪ askepri@epri.com ▪ www.epri.com

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European Technology Development
Fountain House, Cleeve Road, Leatherhead
Surrey KT22 7LX, UK

Principal Investigators
A. Shibli
F. Akther
S. Hampson

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ABSTRACT

The ongoing privatization of electricity generation across the world, competition and shareholder demand for higher profits, stricter regulations on environmental impacts, changes in fuel prices, and the increasing penetration of nondispatchable energy have resulted in an increasing need for larger energy generators to operate as non-baseload units. As a result, both conventional power plants and combined-cycle power plants are increasingly being subjected to load-following and cyclic operation. However, cyclic operation introduces new types and higher rates of damage and can result in reduced performance and increased operation and maintenance and repair costs.

This research project studied the effect of increased plant cycling on plant performance. The report provides estimates of the annual operation and maintenance costs and the basic causes of cost impacts through top-down statistical analysis so that utilities can better understand the economic implications of load-following and cyclic modes of operation. The report is intended to provide better information for decision making regarding dispatching of units and the economics of investing in them, with a view to reducing costs through improvements in operating, inspection, and maintenance procedures.

Keywords
Cyclic duty
Flexible operation
Fossil plants
Operation and maintenance costs
Plant performance

EXECUTIVE SUMMARY

The ongoing privatization of electricity generation across the world, competition and shareholder demand for higher profits, stricter regulations on environmental impacts, changes in fuel prices, and the increasing penetration of nondispatchable energy have resulted in an increasing need for larger energy generators to operate as non-baseload units. As a result, both conventional power plants and combined-cycle power plants are increasingly being subjected to load-following and cyclic operation. However, cyclic operation introduces new types and higher rates of damage and can result in reduced performance and increased operation and maintenance (O&M) and repair costs.

This research project studied the effect of increased plant cycling on plant performance. The report provides estimates of the annual O&M costs and the basic causes of cost impacts so that utilities can better understand the economic implications of load-following and cyclic modes of operation. The report is intended to provide better information for decision making regarding dispatching of units and the economics of investing in them, with a view to reducing costs through improvements in operating, inspection, and maintenance procedures.

Examination of the basic causes of these increased costs with the intent of reducing them through improvements in operating, inspection, and maintenance procedures is of vital importance. Relatively small differences in costs and reliability can make a large difference in the station ranking. Many of the early, less efficient plants have been downgraded to two-shifting operation. However, when a unit is subjected to cyclic operation, reliability can suffer. There is a danger of a vicious spiral in which more cyclic operation leads to more unreliability, which leads to more cyclic operation.

Recent investigations found that a change to operating under cycling conditions results in the following:

- Increased capital spending for component replacement
- Increased routine O&M costs due to increased levels of wear and tear
- Lower availability due to increased failure rate and outage time
- High fuel consumption during startup and shutdown due to inefficient heat transfer and non-optimum heat rate

The difficulty of a cost analysis process lies in establishing the degree to which these factors are applicable to a specific plant. The plant-specific factors that can be influential include the following:

- The overall design of the plant (conventional or combined-cycle plant)
- The individual design of major components (material, thickness, and so on)
- The way the plant is operated (ramp rates)
- The quality of the water chemistry
- The size and age of the plant (generally, size is the most important factor)
- The previous maintenance philosophy
- The control system for setup of the power plant parameters
- The quality and type of fuel

An improved model, designed by using top-down statistical analysis, is presented in this report. The model is based on simplicity and effectiveness, by determining the strength of association between variables, using statistical analysis to determine the dominant driving factors for trend building. The main driving factors and their statistical relationships were determined to assist plant operators in estimating their cost of cycling and forecasting the impact of increasing penetration of nondispatchable renewable energy.

The methodology used to assess startup costs consists of four phases, focusing exclusively on plant performance indicators (equivalent forced outage factor, equivalent planned outage factor, availability, and reliability) and O&M costs in terms of creep and fatigue-life consumption (online hours and starts) to determine the impact of cyclic operation on non-fuel O&M costs. The cost analysis in this study considered the reported expenditure for each unit over a number of years to determine an annualized non-fuel O&M cost. This non-fuel O&M cost did not include costs related to the long-term service agreement. Fuel costs were not included in the cost model because it is a market-specific factor and can be changed at any time. These four phases assist with the total characterization of cycling costs based on plant O&M expenditure. This simple yet robust approach is shown to be effective for creating reliable trends for any type of generation unit to understand the impact of cycling on plant O&M costs.

The report provides an analysis of the levels of equivalent forced and planned outage rates for the entire life cycle of a conventional and combined-cycle plant operating in the cycling regime (considered all types of cycling regimes). The findings reveal familiar bathtub trend lines with three distinct lifecycle phases. The report also provides a breakdown of failure typology for the various regions of the conventional and combined-cycle plant.

The report seeks to establish the annual non-fuel O&M cost for a model plant over its service life. The annualized non-fuel O&M cost is calculated by summing the costs of maintenance and repairs, chemistry, capital expenditures (such as plant modifications to enhance the unit's cycling capability), increased frequency of inspection, and other operating costs. The reported costs of database units are then corrected for capacity and related to two time-dependent variables—creep life (measured in online hours) and fatigue life (measured in starts).

A case study is provided based on the operating and cost information of a specific European system, which allowed a far more detailed analysis to be performed within that system. The case study demonstrates the strength of connection between lifetime online hours and the annual non-fuel O&M cost for baseload-designed units in this system, as well as the strength of correlation between annual non-fuel O&M cost and fatigue-life consumption. Finally, the case study demonstrates the model flexibility by estimating the annualized non-fuel O&M costs with the introduction of nondispatchable wind penetration and provides per-start costs for each year of operation.

The report describes the damage mechanisms associated with conventional and combined-cycle power plant cycling. It also provides recommendations for the improvement of plant management, operation, monitoring, design, staff levels, and training to optimize plant life cycle costs. These recommendations are based on the latest maintenance philosophies and techniques that are currently being applied in only a few plants. The plant database used in the analysis is described in Appendix A.

ABBREVIATIONS AND ACRONYMS

The following abbreviations and acronyms are used in this report:

BOP	balance of plant
CBM	condition-based maintenance
CCGT	combined-cycle gas turbine
CEGB	Central Electricity Generating Board
CLT	central limit theorem
CPP	conventional power plant
CSA	contractual service agreements
DPI	dye penetrant inspection
EAM	enterprise asset management
EFDH	equivalent forced derated hour
EFOF	equivalent forced outage factor
EHS	equivalent hot start
EPDH	equivalent planned derated hour
EPOF	equivalent planned outage factor
ETD	European Technology Development
FAC	flow-assisted corrosion
FD	forced derated state
FGD	flue gas desulphurization
FOH	forced outage hour
GT	gas turbine
HP	high pressure
HPEva	high-pressure evaporator
HPSH	high-pressure superheater
HRSG	heat recovery steam generator
IP	intermediate pressure

IRIS	internal rotary inspection system
LCPD	large combustion plant directive
LP	low pressure
LTRA	long-term repair agreements
LTSA	long-term service agreement
MBO	maintenance basis optimization
MPI	magnetic particle inspection
NDE	nondestructive examination
O&M	operation and maintenance
PaM	proactive maintenance
PD	planned derated state
PdM	predictive maintenance
PH	period hour
PM	preventive maintenance
PMBD	Preventive Maintenance Basis Database
POH	planned outage hour
PTFE	polytetrafluoroethylene
RBMI	risk-based maintenance and inspection
RCM	reliability-centered maintenance

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1 INTRODUCTION

The on-going privatization of electricity generation across the world, and the ensuing competition and demand by shareholders for higher profits has resulted in an increasing need to supply power on demand. As a result, Conventional and Combined Cycle Power Plants (CCGT) are increasingly being subjected to load-following and/or cyclic operation. However, cyclic operation can introduce new types of damage with higher rates and hence results in increased maintenance and repair costs. Analysis of the basic causes of these increased costs is of vital importance. Relatively small differences in costs and reliability can make a large difference in the station ranking. Many of the early, less efficient plants have been downgraded to two-shifting operation. In this study, the basic causes of these higher costs are examined, with a view to reduce them through improvements in operation, inspection and maintenance procedures.

Even new CCGT units may need to be operated in cyclic mode prematurely. Power from hydro, wind and solar energy sources cannot be scheduled in the same way as that from fossil fuel or nuclear plant. Most combined heat and power systems operate in base-load mode, leaving the peaks to be picked up by large centralized stations. Both trends imply further increase in cyclic operation and load-following in future.

The chief pieces of equipment in power plants include the gas turbine (usually referred to in the USA as the combustion turbine), the steam turbine, the alternator (more commonly known as "the generator or electrical generator") and the boiler/ HRSG (Heat Recovery Steam Generator) which utilizes the waste heat from the GT and BoP (Balance of Plant) which represents the auxiliary system required to operate the plant.

Although many new plants are now designed for cyclic operation, efforts to increase short-term reliability are still uppermost in the thinking of the designers. Furthermore, designs are being upgraded in large steps and this can lead to unforeseen problems. The newer units are also presenting a whole new set of materials and component problems. Thus, for example, new higher strength materials (such as P/T91, P/T23 and P/T24) are being utilized in boilers without the benefit of long operational trials and with uncertainty about the long-term weld performance. Differential thermal expansion results in increased stresses, leading to new problems. Furthermore, the read-across from aero gas turbine experience and its transfer to CCGTs is problematic because of scale effects and the completely different operating regime.

Thermal fatigue for both conventional and CCGT plants is at its most damaging when the component is operating in the creep range. The higher the temperature, the higher the creep rate and creep damage, of course. Fortunately high-temperature operation in generation units, in the sense of this being an error as the part of the operating staff is rare. Nevertheless, both the HP turbine stage and sections of the boilers/ HRSGs in modern plants operate well into the creep regime. It follows that creep-fatigue of the classical type, involving gross cracking, can be a major concern. One important issue is that the rates of heating up and cooling down of HRSGs are very high compared to furnace and superheater structures in conventional steam plant.

Indeed, creep-fatigue failures are now being experienced in the modern CCGT and conventional power plants.

Failure by simple differential expansion, without the presence of a high static load, will affect combustor cans, leading to cracking and distortion. The oxidation-resistant and thermal barrier coatings on combustor cans and turbine blades can also be degraded in this way, the former due to spallation of protective oxide films, and the latter due to delamination at the coating-substrate interface. Simple differential expansion may also be a problem in boilers/ HRSGs even when these are not operating in the creep range. This can be due to poor detailed design, poor operation, but it can also be due to condensate being trapped in some legs of the boiler/ HSRGs impeding steam flow during start-up or shut-down.

Waterside issues in plants grow in importance as temperatures and pressures increase. But as with conventional steam plants, cyclic operation can interfere with water treatment and can result in lower water quality. There could be increased problems with corrosion-fatigue of economizer tubing and stress corrosion in steam turbines and many CCGT units require good quality steam to control NOx. Dry air-cooled condensers are sometimes used, rather than wet systems, since there is less need for copious amounts of water, widening the possible range of plant locations.

Generator and switchgear can be susceptible to increased fatigue, wear, and other forms of degradation due to repeated stop-start operation and these aspects are covered in this project. In view of the likely increased damage rates when operating in cyclic mode, the monitoring and inspection of plant components for damage is obviously of great importance. Therefore, monitoring and diagnostics and advanced Non Destructive Examination (NDE) techniques are considered in the optimization section of this study, including vibration monitoring of power plant equipment and on-line component life utilization software systems.

It is clear that cyclic operation of power plant demands a closer understanding of the issues involved, better monitoring of the plant operation and behavior of critical components, a strategy of component inspection and replacement and proper assessment of the costs involved.

The principal aims of this study are to collate and review available information and experience on the operation of power plants under cycling conditions. Assessment of the impact of costs on the Operation & Maintenance (O&M) of conventional and CCGT power plants involved in cyclic operation is the most important aspect of this study. Major identifiable costs are envisaged to be due to component repair, replacement and plant overhauls. There will also be costs due to increased inspection, wear and tear, reduced unit efficiency due to component degradation and management complexity.

Recent investigations found that a change to operating under cycling conditions results in:

- Increased capital-spend for component replacement;
- Increased routine O&M cost for increased levels of wear and tear;
- Lower availability due to increase in failure rate and outage time;
- High fuel consumption during start-up and shut-down due to inefficient heat transfer and non-optimum heat rate.

The difficulty of a cost analysis process lies in establishing the degree to which each of these factors will be applicable to any specific plant. The plant specific factors that can be influential include:

- The overall design of the plant – conventional or combined cycle plant etc
- Individual design of major components – material, thickness, etc.
- The way the plant is operated – ramp rates
- The quality of the water chemistry
- The size and age of the plant – generally size is the most important factor
- Previous maintenance philosophy
- Control system for set-up of the power plant parameters
- Quality/type of the fuel
- Plants' power purchase agreement

Each of these factors will influence or even determine the degree to which cycling will impact on the overall O&M and capital costs. This way, even units that were developed for cyclic operation should be subjected to a thorough analysis of their cyclic operations and costs in order to optimize operations and determine the true costs, thus resulting in significant savings, higher operational flexibility, faster response and improved profitability. As stated above, the historic apportioning of costs has been poor, hence the difficulty in establishing a relationship between plant and probable cost. Nevertheless, the database has accumulated a considerable amount of cost, reliability and availability information allowing the creation of start-up costs model for non-fuel O&M costs which is more effective in aiding plant financial performance and management.

An improved cost module has been developed and is presented in this report. The philosophy of this module is based on its simplicity and effectiveness. Therefore, several correction factors were determined to assist plant operators to determine their true cost of cycling and forecast the impact of increasing penetration of non-dispatchable renewable energy.

2
EVOLUTION OF CONVENTIONAL AND COMBINED CYCLE POWER PLANTS AND IMPLICATION FOR CYCLIC OPERATION

The aim of this study is to identify the problems associated with power plant cycling operation and its impact on O&M costs and on plant performance. There are a large number of plants operating in the cycling regime across the world. Cycling plants are becoming the mainstay of power generation in many countries. Cyclic operation of power plant requires closer understanding of all the issues involved, improved monitoring of the plant operation and behavior of critical components and the strategy of component inspection, modification and replacement. Evolution of Conventional Power Plants (CPP) and Combined Cycle Gas Turbine (CCGT) plants regarding cycling and implication for cyclic operation are discussed in this Section.

2.1 Plant Cycling Experience

The question of two-shifting is becoming a global problem. The database has information on cycling experience of several plants from its worldwide collaboration with plant owners, operators, researchers and associated organizations in Europe, Asia, North America and Australia. This worldwide experience on cyclic operation for conventional and CCGT plants is summarized below.

2.1.1 Conventional Power Plant

A number of factors have led to an increase in fossil fuelled units needing to move to two-shifting. These include:

- Privatization, particularly when the end result is a fragmentation of the industry.
- The fact that nuclear, hydroelectric power and 'green' power plants have priority over fossil fuel units.
- The rise of low-cost gas-fired combined cycle plants which have displaced older more expensive fossil-fired units.
- The increased availability of low sulphur and chlorine coal that has allowed coal-fired units to continue to operate, without the necessity for flue gas clean-up.
- The introduction of windmill farms has also contributed to the fossil plant cycling and will continue to do so for the foreseeable future.

The operation of coal fired power plants in the European Union (EU) is significantly affected by their official report on the limitation of emissions of certain pollutants into the air from large combustion plant or the Large Combustion Plant Directive (LCPD), as it is often called [1]. Plants which do not comply with the emissions limits (NOx, SOx and particulates, for example) in this directive can only run for 20,000 hours between 1 January 2008 and 31 December 2015 after which they must be closed.

The LCPD and the planned closure of nuclear power stations in some European countries such as Belgium, Germany and the UK will result in changes to the operation of the fossil fuelled plant in the next few years. The recent (last two to three years) volatility in the UK gas prices due to the reduction in production from the North Sea and restrictions in gas imports from Europe has resulted in big swings in the operating regimes seen by gas and coal-fired stations in the UK.

Thermal cycling of fossil-fired plant has been practiced within the UK for many years. Since privatization in 1989/90, together with the competition from gas-fired CCGT's, most of the 500/660 MWe coal-fired units have had to adapt to two-shift operation on a regular basis. The need for two-shift operation was recognized prior to 1980 and early trials were carried out by the CEGB (Central Electricity Generating Board) in the UK to identify the potential operational problems. A number of reports were produced and seminars were held.

Figure 2-1
The ultimate development of the CEGB design – Drax 660 MW turbine – HP, IP and 3 double flow LP cylinders

The construction of the Isle of Grain power station in the UK was delayed by industrial disputes. This delay followed by the oil price rise resulted in the decision to never complete the fifth unit. Despite the errors made in deciding to build these oil-fired stations or the plant design selection they became very good at not only two-shifting but multiple starts and stops during a 24 hour period usually Monday to Friday and were often completely shut down at weekends.

In recent years the advent of environmentally friendly generation has put pressures on rest of the generation portfolio. In Denmark, which has one of the highest proportions of wind generation, the load factor of conventional plant has fallen and the number of starts / proportion of part load running have risen. The conventional plant has to be there ready in case the wind does not blow or it blows too hard either of which severely reduces the output of the wind turbine.

Changes in environmental quotas have recently been another reason for the running regime of plant to change suddenly. If a country or generating company only has a certain quota for an emission, for example, oxides of sulphur then it may have to increase the operation of plants which emit no or little sulphur such as gas fired or coal fired with flue gas desulphurization plants installed and to correspondingly decrease the operation of coal fired plant with no flue gas processing.

Changes to the grid system or possibly the availability or otherwise of imports from adjacent utility companies or countries due to political, environmental, economic or other factors can cause a company or country to rapidly reassess its operational situation. An example of this is the contribution of Norwegian hydro based electricity generation to the Scandinavian and Finnish areas. In years of plentiful rainfall the increased Norwegian hydro output pushes the Finnish coal fired stations into the flexible operational regime. In dryer years they move towards base load operation.

In Canada at the Ontario region, two-shift trials were carried out as early as the 1970's with particular emphasis on optimizing operational procedures. Work has been carried out to examine detailed effects of thermal transients on the plant including finite element analysis of major components.

A surplus of capacity led to several fossil power plants being moth-balled in the 1990's, while much of the in-service plant operation fluctuated with the seasonal, weekly and daily demands. Due to the move to a competitive market in Ontario, the ability to operate fossil plants in a flexible mode is seen to be an important part of the generation mix in a commercial environment. Some moth-balled plants may be brought back into use for two-shift operation.

Ignoring the relatively rapid changes which can occur due to the reasons explained above then generally it would be expected that as a plant ages it becomes a less efficient member of the fleet and gradually moves into the flexible operation regime. If the plant gradually moves into a flexible operating regime then the transition is a gentle one with cyclic operation tending to start with summer weekends and gradually spreading from there.

It is expected that a fully competitive market will evolve in the USA similar to those currently developing around the world and that the economics of two-shifting will necessitate more interest in this mode of operation.

Initially a plant may be uneconomical to run for relatively short periods – weekend nights in the summer, for example. In this case the decision may be taken to run through at minimum load. As these periods progressively become more frequent and longer including some weekday nights and longer weekend periods – perhaps all day in the summer the option to continue to operate at minimum load becomes less and less viable. The losses incurred by inflexible operation will mount. Eventually the situation is reached when the commercial position forces the plant into flexible operation or cycling.

Many prospective owners of power plants do not detail sufficiently in their specification what flexibility they require from the plant later in its life. In some cases they may not fully understand what flexibility they will require but they should specify some degree of flexibility and life in terms of number and type of starts (hot, warm and cold) that the plant will be required to carry out.

More modern power plants have more sophisticated computerized control systems which should be capable of correctly controlling the plant under all operating conditions. Once the commercial, environmental or other pressures become irresistible then the plant moves into flexible operation.

2.1.2 Combined Cycle Power Plant

It was not until the mid-1980s that combined cycle power plant or CCGT-units attracted serious attention as a competitor to coal-based steam or nuclear generating plants. CCGT units built up to that time offered net thermal efficiencies of around 40%, as both the gas turbine inlet and steam temperatures were modest. This level of efficiency was hardly better than that obtainable from the best coal-fired plants. Of course, on a gross basis the differences were even more marked because of the high hydrogen content of natural gas compared to coal. Output from individual GT units was also low, under 70 MW, so that where there was a need for high power output a number of gas turbines would be used in parallel with one steam turbine. Each GT would fire into a relatively small HRSG system, with the steam from this being fed into a steam turbine. A good example is Jersey Central in the USA which was built in 1974. Here there were four GT units, each producing 47.5 MW, with a steam turbine of 129 MW. The plant efficiency was about 38%.

Units of this type were relatively undemanding of gas turbine hot gas path components, heat recovery steam generators (HRSGs) and steam turbine equipment, even when subject to cyclic operation. The GT temperatures were low, and during start-up it would be possible to bring on individual units separately, so that steam turbines were not subject to shock temperature changes. The situation on the European continent was different to that in the USA, with some of the early units in Europe being subject to double-two-shifting, that is, units being started up and shut down twice a day. One of the early units, in Belgium, accumulated over 2,500 starts in its first decade of operation.

GT units on these early plants used un-cooled blading. Eventually cooled blading of a relatively unsophisticated type was introduced, where the cooling air was led through internal passages, which exited at the blade tips. More recently, blade cooling has followed the pattern used in aircraft gas turbines where the cooling air is led through holes at critical points on the blade surfaces. Over the years there have also been some changes in HRSG design. Some designers have opted for once-through rather than drum-type systems in the high pressure section of the

HRSG train. There has also been increasing interest in vertical HRSG boilers rather than horizontal designs. The former seem to be preferred in Europe because of the reduced foot print areas, saving on ground space.

From the mid-1980s onwards, because of the improvements in blade materials and cooling techniques, there has been a rapid evolution of GT temperatures, resulting in efficiencies reaching close to 60% at the present time. GT power outputs have also increased dramatically. Hence, a modern CCGT plant arrangement is one GT unit combined with one steam turbine, with the GT providing more than two-thirds of the total output. Despite the more advanced plant, construction times are still one to two years. Furthermore, the advantages of plant location, low manpower and capital requirements have also been maintained, which result in lower generating costs. However, it was the low price of gas throughout the 1990s, plus the dismantling of state-run conglomerates that were previously committed to coal or nuclear power, which have been the main drivers to CCGT construction.

The increase in the price of gas in recent years, together with the development of power trading, has seen a shift in the merit order of the first generation CCGT units, such that only the modern machines, of the highest efficiency, are guaranteed base load. Consequently, many of the original "base load" designed units are now load-following, two-shifting or even double-two-shifting depending on the economics and flexibility. The situation has varied from region to region. Thus, for example, at St Francis in Missouri, USA, the CCGT has run in a cycling mode since inception because of high gas prices. In the UK the workings of the so-called pool system encouraged the overbuilding of CCGT plants. As a result, there is now an excess of capacity in the UK, while coal prices have been dropping.

CCGT plant flexibility does appear to be much poorer than conventional steam plant, which can be run down to 40% of rated output. CCGT units have difficulty in getting below 60%. This is partly due to problems with NOx suppression at low plant outputs which can, in these circumstances, go up to 300 ppm. There is also a risk of the GT tripping out, particularly if the system frequency is dropping at this time. A further problem is the length of time it takes for the HRSG plant to achieve full output. Hence, although a CCGT plant may be able to produce power relatively quickly, it is not really suitable for load-following until sometime after start-up. These issues are reviewed in a New Zealand report on system frequency control. In view of these concerns the UK Government requested Merz and McLellan to review the situation, as it is anticipated that by 2020 up to 90% of the power in the UK will come from systems utilizing CCGT technology. It is understood that apprehension about this issue influenced the (New Electricity Trading Agreements (NETA), which impose stringent requirements on plant availability and output.

Despite these constraints some CCGT plants, intended for cycling, have been built for Independent Power Producers (IPP) or "Merchant Power" organizations. These units were intended, right from the start, to meet gaps in the market, and have been operated in a two-shift or base load mode throughout their history, although they may not be run in a load-following mode. CCGT plants of this type are particularly interesting because of the specific demands on equipment and personnel. As competitiveness is very high, most IPP organizations have cut plant personnel to a minimum but, on the positive side, the quality and know-how of the remaining staff is high. However, it can be difficult for operating personnel to keep abreast of new developments, resulting in a strong need for a report such as this.

Units which are built with cycling in mind as a prime requirement are often equipped with stack dampers which reduce heat losses from the HRSG during the shutdown period. Conversely, many of the plants intended for base load duty dispense with dampers, as flue gas leakage up the bypass stack can reduce HRSG output by around 0.25%, according to Jarvis and Raddings. Figure 2-2 shows a CCGT plant at Peterborough, England, in which the main and bypass stacks can clearly be seen.

Figure 2-2
CCGT plant at Peterborough, England, showing the main and bypass stacks

2.2 Changes to Plant Layout

The move to flexible operation requires plants to make various changes to plant layout and adjust the design criteria. This is applicable both to conventional and CCGT plants and discussed in this Section.

2.2.1 Conventional Power Plant

If it is considered economic to purchase a power plant then it is almost certain that it will be the best in the fleet when commissioned and will run at high load for the first part of its life. It will be the best for a number of reasons such as:

- Use of the current fuel of choice.
- Location of demand on the grid system.
- Meeting all the current environmental emissions limits.
- Higher efficiency than existing plants.

The move to flexible operation carries with it risks to the plant due to increased wear and tear from the cycling conditions within the plant. The stopping and starting of equipment and increased operation of items such as electrical circuit breakers all add up. The start up and shutdown processes utilize heat which does not immediately produce an electrical output. These have to be carefully weighed up against the savings which come from avoiding generating out of merit.

There is now extensive experience of cyclic operation worldwide, and whilst there are potential risks and added wear and tear associated with cycling, with due care and application of sound engineering and operational practice, economic cyclic operation can be achieved with confidence.

Avoiding thermal transients, both in the form of quenching and high rates of rise of temperature can be minimized by careful management of the unit off load and by the addition of engineering features to alleviate the potential problems. In the UK a number of natural circulation drum boilers have been fitted with off load circulating systems with the objective of pumping water slowly around the evaporative section to balance out temperature variations. The aim is to eliminate flow stagnation and tube-to-tube temperature differences. In addition, inter-stage drains or vents have also been fitted to promote flow through the superheater stages and avoid quenching problems arising when ‘cold’ condensate from platen elements is otherwise pushed into the relatively hot final superheater stages.

The problems associated with a plant will be influenced by its own conditions and requirements and need to be assessed individually. The principal considerations will depend to some extent on the type of plant, that is:

Drum natural/forced circulation or once through:

- Throttle or sliding pressure control
- Sub-critical or supercritical
- Fuel type
- Local circumstances

The main constraint on operation is matching of steam and turbine metal temperatures. Sliding pressure offers advantages over throttle control during a start up by establishing a flow to the turbine earlier in the sequence with lower overall heat input. It also offers advantages of retaining high temperatures on shut down. *Hence operators of plants that are required to regularly cycling, adopt partial sliding pressure control system.*

The downside to variable pressure operation is:

- Increased oxygen attack from condenser air in-leakage
- Release of steam bubbles in economizer and primary evaporative sections on pressure reduction leading to localized erosion-corrosion especially in horizontal sections (e.g. floors and roof sections)
- DNB (Departure from Nucleate Boiling) in lower tube sections resulting in increased concentrations of solids, corrosion and local overheating
- Local overheating and thermal fatigue arising from disruptions in flow at low loads.

Local constraints that need to be taken into account, when cycling, include the capacity and availability of the water treatment plant. Cyclic operation will increase drainage losses and hence increase demand. It may be necessary to install additional capacity for both storage and production. The light up system for coal stations may also be undersized for regular cycling, especially where more than one unit is being operated in this mode. Other aspects of local conditions include environmental considerations such as noise and flue gas treatment.

2.2.2 Combined Cycle Power Plant

Older CCGT units tended to be of the so-called 2-2-1 arrangement (i.e. 2 GTs, 2 HRSGs and 1 steam turbine). Increased GT inlet temperatures have resulted in some increase in exhaust gas temperatures, and HRSG steam temperatures and pressures have also risen. GTs also increased in size leading to larger HRSGs and a move to 1-1-1 combinations, which are preferred as it further reduces costs by allowing the use of single shaft machines, in which the GT and the steam turbine share a common drive shaft to a single generator, and benefits from a more compact and simple design.

The evolution of CCGT plants shows that there are opposing forces at work, which influence the type and scale of operating and maintenance problems. For example, the particular mix of units will have a substantial impact on the ability to load follow. The older 2-2-1 CCGT will be very flexible in terms of output compared to a more modern 1-1-1 arrangement with the same power output. Compared to a modern CCGT, the steam turbine of an older 2-2-1 system will see fewer shock changes in temperature. Furthermore, a good operator will have installed cross-connections between the two HRSG units so that thermal shocks are reduced and faster heating-up times are possible. On the other hand, many older HRSG units were built with inadequate provision for condensate drainage, so that some units suffered very badly from condensate quenching damage in the superheaters.

The 1-1-1 arrangement removes one complication, in that there is no need to match steam temperatures at a common steam outlet header where there are two or more HRSGs. This can be a problem in cyclic operation since the two sets of GT and HRSG are unlikely to start-up and come on stream at exactly the same time. However, on balance a 2-2-1 design is better, providing the operators make use of the facility of having two independent steam systems, to keep lines hot and run the steam turbine at a very low output.

Another trend is the wider introduction of vertical gas-path HRSG systems. It is noteworthy that vertical designs are relatively common in Europe, since the footprint or ground area of these units is somewhat smaller compared to horizontal designs. In addition, there is some evidence that the vertical type has advantages in that flue gas and water distributions are better during start-up. Hence these units are better for cycling duties.

2.3 Plant Limitations in Respect of Cycling

There are certain limitations for the plants (conventional and CCGT) operating in the cyclic regime and these are discussed in this Section.

2.3.1 Conventional Power Plant

From about the mid 1970's, most turbines for conventional plant were designed on the basis of a 200,000-hour operating life with up to 5,000 hot starts, 1,000 warm starts and a few hundred cold starts. Evidence to date would suggest that in general most turbine plant is on course to achieve this objective. The forced outage rate attributed to turbines is historically quite low with values of less than 0.5. The general perception is that turbines do not suffer significantly from operation in a cyclic regime, provided of course that due care is taken as set out below.

Most large turbine currently in use conform to a set of standard modules, usually comprising HP, IP and LP turbines based on a manufacturer's 'standard' configurations. HP cylinders are typically a single flow type with double shell construction, while IP and LP turbines are usually double flow single shell construction. The majority of rotors are monoblock type with two journal bearings located outboard at each end of the cylinder. The thrust bearing is usually located between the HP and IP turbines. Blading is usually a disc and diaphragm construction.

Operation in a cyclic regime has two main effects:

- Thermal fatigue and associated creep fatigue
- Mechanical fatigue due to load and speed variations

Creep fatigue associated with thick walled components, including governor and stop valves and HP and IP turbine inlet belts. Modern analysis methods, utilizing finite element methods are now widely available at reasonably low cost to permit modeling of components perceived to be at risk. Application of this type of modeling, whilst it may not be able to accurately predict the life of components, does provide a valuable understanding of the stress profiles within the component and identifies potential weaknesses and vulnerable areas. Armed with this knowledge, operational procedures can be optimized to minimize the effects of thermal fatigue and inspection procedures focused on selected locations at appropriate operating intervals. The addition of temperature and temperature differential instrumentation will enable the operator to minimize the intensity and duration of the adverse conditions which can then be incorporated into auto start sequences. The scope for modification to existing turbine plant is limited unless new rotors or casings are being fitted. Possible modifications to reduce thermal stresses include improvements to thermal insulation, pre warming (especially of half joint flanges) and slotting of flanges to increase flexibility. EDF are known to favor "skin peeling" of high temperature rotors where fatigue damage is of concern. This is basically skimming off a millimeter or so of material from the surface of potentially critical regions of the rotor e.g. radii etc. This effectively removes fatigue related damage what is essentially an "as new" surface. This process is typically carried out at midlife.

Clearly, operating in a cycling mode will require increased operation of turbine governor valves and stop valves. Inevitably there will be additional wear and tear on the valve seats and valve stems, especially under throttling conditions when flow induced vibration can lead to mechanical fatigue and wear. This can, in the main, be contained by redesign of the valve head, modification to the steam flow path and the use of stellite or similar hard facing materials on wear surfaces.

Concern has grown about generator ring integrity, despite the introduction of the Fe-18Mn-18Cr end ring alloys, leading to the holding of EPRI sponsored workshop on the subject in 1997. At this ABB recommended that before cycling commences, end rings should be ultrasonically tested and are able to supply an automated technique. If the end rings (Figure 2-3) are of the older Fe-18Mn-5Cr type alloy, the recommendation is that these be inspected, if there has been any suspicion of moisture contamination, because of the threat of stress corrosion (see later Sections).

Figure 2-3
End rings on an alternator rotor

The stress variations which result from cyclic operation are more important, although variations in the electrical output can also induce fatigue, even though there is no change in the rotor speed. This is due to eddy current induced heating in the end ring.

Older designs of end ring are likely to be more susceptible to damage, since they are prone to high cycle fatigue. One of the older variants uses retaining rings that are shrunk onto the end disc, which itself is shrunk onto the shaft. Here a relative movement between ring end and the end windings occurs with each revolution. The other design can induce cracking in the shaft, in the rotor tooth region and in the end ring itself. Here the ring is shrunk on to the end windings and the shaft. The most modern design, which is less susceptible, to high cycle fatigue, is of the so-called “cantilever type”. Here the end ring simply grips the alternator rotor, and does not rely on support from the shaft. Another shortcoming of older designs was that the fixing of the end ring to the rotor was of a relatively simple form. Modified designs are available to reduce stress concentrations in this area.

2.3.2 Combined Cycle Power Plant

As has been pointed out by Starr in papers dealing with both conventional and advanced power generating systems, a major consideration that should influence potential operators is the ability of any new-build plant to adapt to cycling as some of these limitations are essentially inherent in the design. To take an extreme example, nuclear plants are not suitable for two-shifting. On the other hand, some energy conversion systems may be able to two-shift with ease, providing certain compromises are made over capital cost, operating temperatures and efficiencies. The issue of the need to two-shift and to load-follow will grow as the proportion of plants increases in the system. This is possibly more of a problem in European and other countries which are gas-rich and coal-poor, e.g. the UK. Within a few years up to 90% of the generation may be of the CCGT type in these countries. How do the conventional CCGT plants then compare in terms of cycling? These issues have been addressed in a recent paper reviewing the situation in New Zealand and benefit of that review has been taken in this Section.

Much of the inflexibility results from the boiler/ HRSG system which, as emphasized in this study, is basically a different "animal" to the steam boiler in a coal- or oil-fired station. In the CCGT plant, the heat-train thermodynamics and heat transfer characteristics dictate that steam is raised at two, if not three, separate pressure levels. Furthermore, the exhaust steam from the HP turbine is mixed with the steam from the LP section of the HRSG, which are then piped into the LP turbine (Figure 2-4). This is mounted on the same shaft as the HP turbine. For a variety of reasons this heat-train and steam turbine combination results in greatly reduced flexibility.

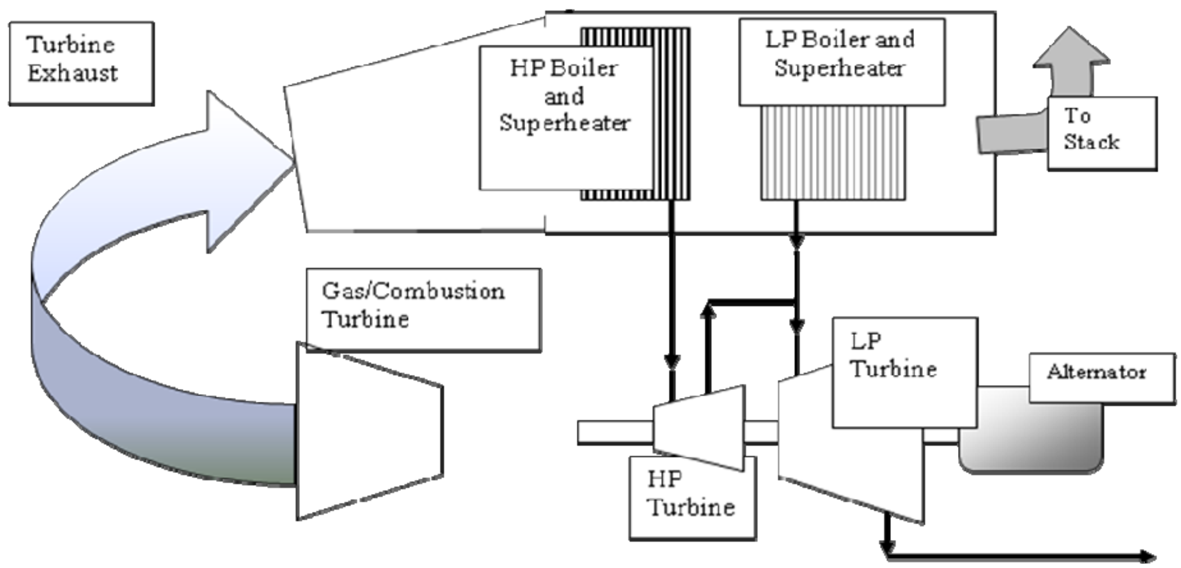


Figure 2-4
Schematic of dual-pressure HRSG and steam turbine system

There are a variety of reasons for this poor flexibility, which tends to affect load-following and frequency adjustment rather than start-up capabilities. Load-following tends to be poorer than with steam plant since:

Down to about 80% of load, there is only a relatively small change in gas turbine exhaust temperatures. Coupled with the small difference between the flue gas and steam-side temperatures, this implies that the HRSG responds only slowly to changes in plant output.

HRSG systems run in a sliding throttle mode, so that even with steam drum systems, there is no reserve of pressure to increase steam turbine power instantaneously.

Because the HP steam merges with the LP steam after leaving the HP turbine, there are potential problems in throttling back in that the exhaust flow from the HP turbine may be choked or opened up too much. Hence dropping the load quickly is not an easy option. There are other drawbacks to CCGT operation, in terms of frequency control, which apparently become more critical in some countries where frequency margins are much wider than in Europe. Nominal 50 Hz units are expected to run down to 49.5 Hz in Europe, whereas in New Zealand, for example, there is a requirement to go down to 45 Hz for very short periods.

Once again, the basic reason for these difficulties stems from the design of the HRSG. There is no feed-heating in the HRSG, and hence all of the steam passes into the LP turbine, which needs to be disproportionately large to accept all of the steam. In particular, the rotor size is large in terms of power output and the LP blades are longer. Long rotor blades can give problems with resonance, if the turbines are run at low system frequencies. Table 2-1 gives the following limiting times for a 50 Hz machine operating at and below normal speeds. Although low-speed off-design operation will not be of concern to most operators, it is worth keeping in mind the issue of blade resonance and the need to get through critical periods when bringing a machine up to speed.

Table 2-1
Lifetime limits imposed by blade resonance

Frequency Range	Lifetime Limit
Above 47 Hz	No time limit
46.5-47 Hz	90 minutes
46 –46.5 Hz	12 minutes
45-46 Hz	1 minute

Clearly susceptibility to the problems of resonance is machine-dependent. If concern about off-frequency-operation is thought to be an issue, it is worth asking about blade resonance characteristics and the amount of damping which has been provided.

The mass of the rotor can also give problems, particularly in the larger 1-1-1 units where the gas turbine and steam turbine are coupled together. One positive advantage is that the mass of the rotor combination provides a good deal of stored energy that can help stabilize the system. Conversely, when the grid is attempting to come back to normal frequency, the speeding-up of the rotor mass will impose some additional load. Furthermore, in low frequency situations, the gas turbine will be producing significantly less power, as pressure ratios will be down due to the lower speed. Load problems of this type will occur in hot conditions where there is a high domestic load for building-cooling. The gas turbine suffers, under these conditions, because air density is low and outlet compressor temperatures are high. The net result is that less fuel can be burnt. Even in temperate climates, such as in the UK, there is a noticeable difference between winter and summertime outputs. Less fuel can be burnt under warmer conditions, so power output suffers. System frequency is then likely to drop compounding the problems.

3
DATA COLLECTION AND STATISTICAL ASSESSMENT METHODOLOGY

This Section of the report describes the statistical model used to organize and analyze the data for the performance and cost assessment purposes.

3.1 Historical Data Base

One of the most important factors in a cyclic cost analysis is the amount of data used to forecast maintenance and cost evolution. Based on past experience, it is clear that a minimum of 50 plants is normally required to obtain an acceptable amount of data to provide a reliable result.

The time frame over which the data is available is also of much importance. Thus for a reliable cycling analysis it would be necessary to acquire a minimum of 7 years of operation & maintenance (O&M) data, since in some cases, especially for the newer plant, the impact of cycling does not appear to be immediate.

The cost analysis presented in this study was performed based on a total of 30 conventional and 65 CCGT plants from the ETD original database. This database includes O&M data from a number of worldwide plants (Europe, USA, Asia etc.) collected by individual plant surveys and from earlier studies.

Data were collected from both the conventional and CCGT plants operated in the base load mode and cycling regime and plant net generation is between 50 to 500MW capacity. During the analysis, the data were separated for base load & cyclic operation and based on these data the performance analysis was performed for both operating condition to determine the impact of cycling on O&M performance and costs. The plant age varies from 1 to 35 years of operation for conventional and from 1 to 30 years of operation for CCGT plants.

As mentioned before in this Subsection the amount of O&M data is critical for an accurate analysis, therefore plants with a minimum of information between 1 to 10 years old were selected for this analysis.

3.2 Statistical Assessment Methodology

The statistical data analysis was performed based on the *Central Limit Theorem (CLT)*. This theorem states that the distribution of a sum of many independent random variables (identically distributed) tends towards the normal density of normal distribution. This is illustrated Figure 3-1.

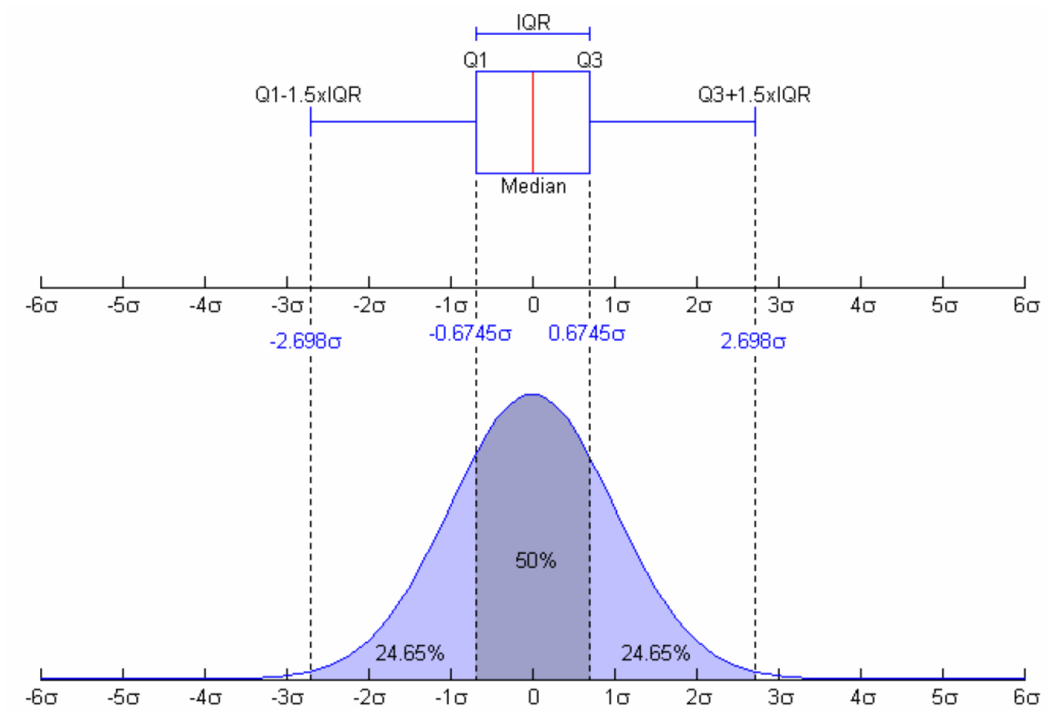


Figure 3-1
Illustration of central limit theorem

The mean value is equal to the median value in a normal distribution and about 68% of the values are within 1 standard deviation of the mean (mathematically, $\mu \pm \sigma$, where μ is the arithmetic mean and σ is the standard deviation), about 95% of the values are within two standard deviation ($\mu \pm 2\sigma$), and about 99.7% lie within 3 standard deviation ($\mu \pm 3\sigma$). This is known as the 68-95-99.7 rule or the empirical rule.

It is common to define the following quartiles:

First quartile (designated Q1) = lower quartile = cuts off lowest 25% of data = 25th percentile (μ - 0.6745σ)

Second quartile (designated Q2) = median = cuts data set in half = 50th percentile

Third quartile (designated Q3) = upper quartile = cuts off highest 25% of data, or lowest 75% = 75th percentile (μ + 0.6745σ)

The CLT is integrated with the normal distribution for the real case of cost analysis. Since this study integrate a large amount of random data, the probability distribution for this large random data will tend towards a normal distribution.

Based on this approach, the random data was included in several categories such as age, number of equivalent hot starts (EHS) etc. Each category is divided by several groups and then the population mean value was determined for each individual group.

If x are the random samples of size n, the population mean (μ) is defined as:

$$\mu = \frac{1}{n}\sum_{i=1}^{n} x_i$$

The standard deviation is determined for each individual group. The standard deviation (σ) shows how much variation exists from the mean value. A low standard deviation indicates that the data points tend to be very close to the mean value, whereas high standard deviation values indicates that the data points are spread out over a large range of values.

The standard deviation (σ) is defined as:

$$\sigma = \sqrt{\frac{1}{n}\sum\left(x_i - \mu_i\right)}$$

The next step is the calculation of the normal (or Gaussian) distribution which is a continuous probability distribution that has a bell-shaped probability density function, known as the Gaussian function or the bell curve. The normal distribution depends on the mean value (μ) and the standard deviation (σ). The normal distribution function is defined as:

$$f_{\mu,\sigma}(x) = \frac{1}{\sigma\sqrt{2\pi}} e^{-\frac{1}{2}\left(\frac{x-\mu}{\sigma}\right)^2}$$

The upper and lower limits were then determined by considering the standard deviation, confidence level and size of the sample. In this study, the ***upper and lower limits*** shown in some of the figures were determined by considering 95% confidence interval. In statistics, a confidence interval (CI) is a type of interval estimate of a population parameter and is used to indicate the reliability of an estimate. In other words, it is a range of values so defined that there is a specified probability that the value of a parameter lies within it. The desired confidence level is determined by the analyst and can be shown at several confidence levels, for example 50%, 95% and 99%. However, in practice, confidence intervals are typically stated at the 95% confidence level [12].

The CLT tells, quite generally, what happens when there is a sum of a large number of independent random variables, each of which contributes a small amount to the total. The normal distribution is considered the most prominent probability distribution in statistics. It is the first function that arises from the CLT, which as described above states that under normal conditions (identically distributed), the mean of a large number of random variables independently drawn from the same distribution is distributed approximately normally, irrespective of the original distribution. This gives the exceptionally wide application in sampling analysis. In addition, the normal distribution is very stable analytically i.e. a large number of results involving this distribution can be derived in explicit form. For these reasons the normal distribution was selected to perform data analysis in this study.

4 CONSIDERATIONS AND METHODOLOGY FOR CYCLING COST ASSESSMENT

Cycling operation has become the most popular operation regime for old conventional plant and for new and old combined cycle plants. To analyze the impact of cycling operation on O&M costs, a methodology that allows collecting data from various plant operators and inputting the data in a statistical model to generate O&M performance trends was developed. This helps with the understanding of the impact of cyclic operation on O&M costs and also on plant reliability and availability.

This Section describes the operation modes (various cyclic regimes) and start-up frequency that the plants are normally exposed/ requested to perform under cyclic mode. Furthermore, this Section shows the flow chart for the cost model developed based on the non-fuel Operation & Maintenance (O&M) costs to determine the start-up costs for cyclic operation and to understand the impact of cycling on the O&M costs.

4.1 Types of Cycling Regimes

It is important to understand various cycling operation regimes that have been considered for the cost analysis.

Generally it should be recognized that "cyclic operation" or "cycling", is a wide-ranging term that covers the following or any combination of:

- **Two-shifting** in which the plant is started up and shut down once a day.
- **Double two-shifting** in which the plant is started up and shut down twice a day.
- **Weekend shutdown** in which the plant shuts down at weekends. This is often combined with load-following and two-shifting.
- **Sporadic operation** in which the plant operates for periods of less than two weeks followed by shutdown for more than several days.
- **Load-following** in which the plant is on for more than 48 hours at a time, but varies output as demand changes.
- **On-load cycling** in which, for example, the plant operates at base load during the day and then ramps down to minimum stable generation overnight.

"Two-shifting" is probably the most common form of cycling, indicating that a power plant is shut-down at night so that it only works over the period of the morning and afternoon shifts. As noted above, two-shifting also normally implies that the plant will be shut-down from Friday night to Monday morning. "Double Two-shifting" is considered to be the most damaging form of cycling.

4.2 Equivalent Hot Start

Equivalent Hot Starts (EHS) has been calculated for conventional and CCGT plants to determine the impact of cycling on the plant performance and O&M costs. Starts are categorized as being hot, warm or cold, depending on the length of time that the unit has been offline and cooling before the start up process begins. The amount of service life consumed by a stop/start cycle is a function of the range of temperature change, so the longer that a unit has been cooling from its operating temperature (for example, 540°C for a steam turbine), the more fatigue life is consumed when the next start up cycle begins. The general approach adopted throughout the industry is that a hot start represents an overnight shutdown, or less than 8 hours offline (turbine metal temperatures >400°C); a warm start reflects a weekend shutdown of more than 8 hours and up to 48 hours (>200°C); and a cold start anything greater than 48 hours offline (<200°C).

In terms of service life consumed by each type of start and its consequent cost, starting from a low temperature after a prolonged period offline is more damaging and therefore costly than starting after a few hours, when the metal components are still hot. The industry metric used to measure the performance and cost impact of each type of start is the Equivalent Hot Start (EHS), which ascribes a higher relative cost to warm and cold starts to reflect the additional damage caused. Different studies have reported varying cost impacts for each start category; however, the widely used ratio of 1:3:5 [Refs. 13, 14, 15] for hot, warm and cold starts respectfully was adopted for this report. Depending on how much care is taken over the various starts then the severity of the cold and warm starts can vary enormously so the ratios could in reality be very different. Similarly there can be hot warm starts and cool warm starts which can cause different problems to different areas of the plant. Hot starts can be very damaging if care is not taken to minimize chilling of the boiler headers and turbine inlet sections - so they are not necessarily low damage unless they are executed correctly. One has to work with averages so these ratios are considered a reasonable estimate of the relative damage from the various types of starts. The relationship used for this study to calculate accumulated EHS is therefore shown below.

$$HS = 1 \times EHS, \quad \mathrm{WS} = 3 \times EHS, \quad \mathrm{CS} = 5 \times EHS$$

$$Total\ EHS = 1 \times HS + 3 \times WS + 5 \times CS$$

Where

EHS = Equivalent hot start

HS = Hot start

WS = Warm start

CS = Cold start

The regions for cycling regime, described in *Section 4.1 ‘Types of Cycling Regimes’*, can easily be identified from the equivalent hot start (EHS) performed per year by the cycling plant, shown in Figure 5-5. Based on this, it is considered that a plant is operating in the ‘***sporadic operation***’ if it performs more than ~ 50 and up to ~ 150 EHS/year, a plant is operating in the ‘***weekend shutdown***’ regime if it performs more than ~ 150 and up to ~ 250 EHS/year, a plant is operating in the ‘***two-shifting***’ regime if it performs more than ~ 250 and up to ~ 350 EHS/year and a plant is operating in the ‘***double two-shifting***’ regime if it performs more than ~ 350 EHS/year.

EHS have been calculated for all units in the database to determine the relationship between Equivalent Forced Outage Factor (EFOF), Equivalent Planned Outage Factor (EPOF) and EHS to understand the cycling impact on plant performance. It has been proved in several projects [Refs. 7, 8, 11] that the increase in start-ups is directly related to an increase in failure rate. ***Based on this previous experience, it is expected that the EFOF and EPOF increase with the increase in number of start-up which is reflected in the figures in Section 5 for conventional and CCGT plants. This approach is interesting and clearly shows the relationship between downtime due to failure and the cumulative number of equivalent hot starts. The analysis also demonstrates the impact of cycling on the O&M costs due to failure. Therefore, it appears to be beneficial to perform the cost analysis study based on the number of starts to fully understand the impact of cycling on the O&M costs.***

Note: Data for accumulated EHS were collected for the last 3 to 10 years of operation for each plant in the ETD database. Therefore, annual EHS represents the 'data collection period' (last 3 to 10 years of operation) only and the types of various cycling regimes (except load following) were identified based on this annual EHS according to the above definition of each cycling regime. The information for the load following cycling regime was provided by the relevant plants.

4.3 Cost Assessment Methodology

The methodology to assess start-up costs consists of four phases described below and focuses exclusively on plant O&M performance including creep & fatigue life consumption and associated costs to determine the impact of cyclic operation on non-fuel O&M costs. The costs data considered in this study are not the cost of cycling but are the annual non-fuel operation and maintenance (O&M) cost (i.e., all the variable annual non-fuel spend on operations). During the survey costs data were provided in different ways (some are more and some are less detailed) by different utilities; only the general cost variables for the O&M costs was considered for this study. Therefore, the annualized non-fuel O&M cost is calculated by summing the costs of maintenance and repairs, chemistry, modifications/capital expenditure (for example, in plant modifications to enhance the unit's cycling capability), increased frequency of inspection and other operating costs. This non-fuel O&M costs does not include LTSA (Long Term Service Agreement) costs. Fuel costs are not included in the cost model because this is a market specific factor and can change at any time. Lost generation cost due to forced and planned outages is also not included in the model as this depends on the current electricity price which is also a market specific factor. However, energy consumption for various sizes of cycling plants due to cycling is discussed in a separate Section (Section 7).

Phase one of this work is the data analysis from ETD original database. During this analysis, the data was separated as plant performance and cost data for base load & cyclic operation.

Phase two is the statistical analysis and normalization. To perform this task it is necessary to have a database with a considerable amount of plants and a statistical model which helps to perform data analysis and normalization. The statistical model, Central Limit Theorem, is used in association with the normal distribution. This theorem helps to determine the best practices based on normal distribution of data. In this phase, the plant performance (i.e. EFOF, EPOF etc.) analysis is performed where Reliability and Availability are determined on a life cycle basis. The plant performance analysis was performed for both base load and cyclic operation to determine the impact of cycling on O&M performance.

Phase three is involved with the detailed analysis of plant condition to separate the associated costs for various relevant operating and maintenance events (i.e. modification, damage, chemistry, management, inspection etc.). The statistical analysis also has been performed in this phase for normalization of the cost data (examining the correlation of the data) to determine the best fit data to develop cost model for cycling O&M costs.

Phase four is the last step of this process and is related to the determination of the start-up costs for cyclic operation. In this phase, creep life consumption (base load operation) and fatigue life consumption (cyclic operation) are measured to determine the dominant mechanism of damage or plant life degradation which supports the cycling costs assessment methodology. The start-up cost is determined for each type of start-up (hot, warm, cold etc.).

These four phases will assist with the total characterization of cycling costs based on plant O&M performance. This approach has proved to be effective for creating reliable start-up costs for any type of generation unit and for the understanding of the impact of cycling on plant O&M costs. The flow chart shown in Figure 4-1 summarizes the phases described in this Subsection.

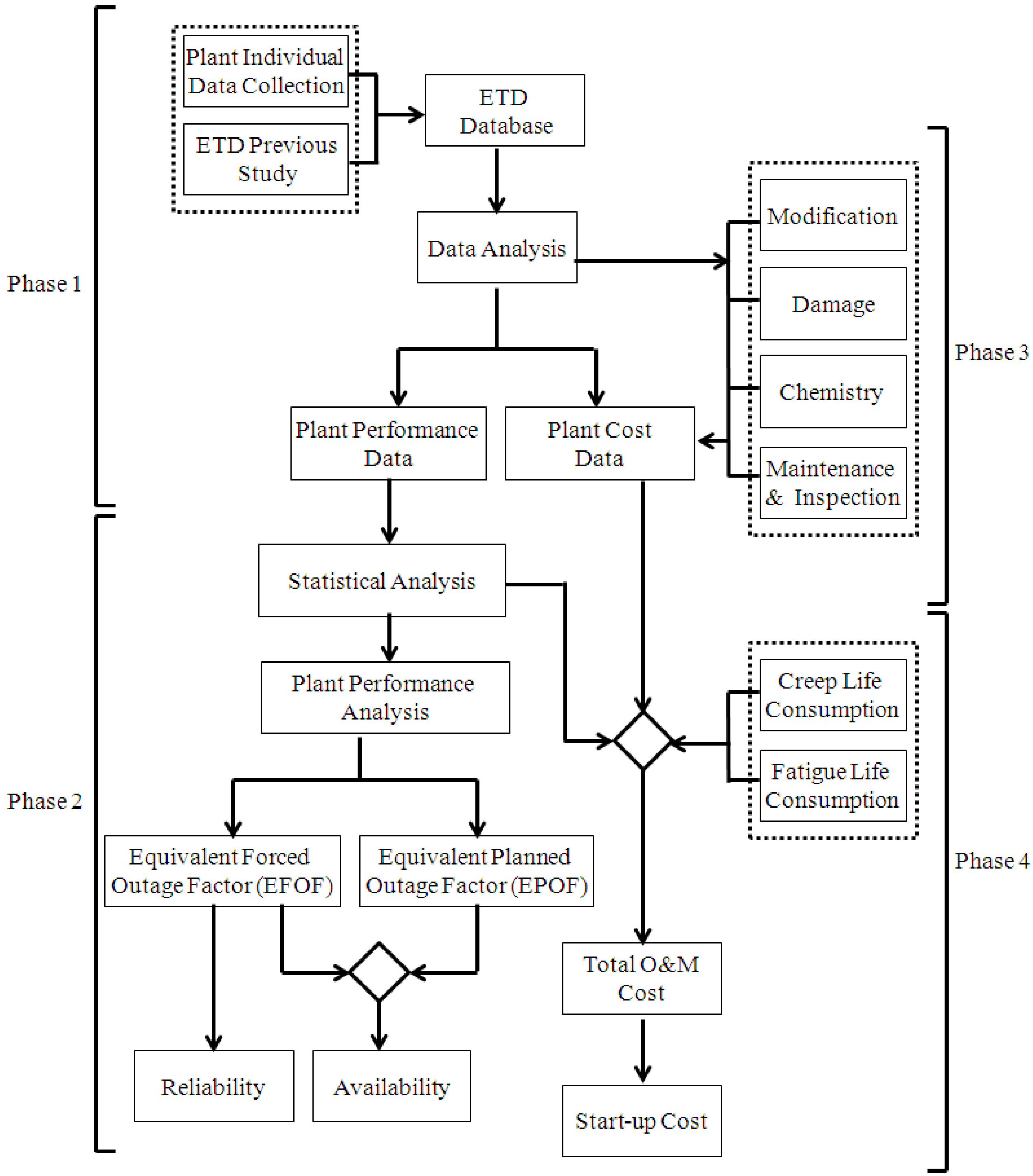


Figure 4-1
Flow chart exemplifying methodology for plant performance and cycling costs assessment

5 PLANT O&M PERFORMANCE ASSESSMENT

Plant condition is the most significant aspect of any power plant. The most important issue in the power industry business is high 'Availability and Reliability'; in some cases these two aspects are even more significant than higher efficiency. Plant 'Availability and Reliability' aspect is necessary to schedule a plant Operation & Maintenance (O&M) performance assessment.

There are two major factors that determine the O&M performance. These are planned and forced outages described as Planned Outage Factor (POF) and Forced Outage Factor (FOF). Power plants sometimes operate in the reduced load condition due to a failure or other problems; these terms are therefore designated as Equivalent Planned Outage Factor (EPOF) and Equivalent Forced Outage Factor (EFOF) as discussed later.

This Section discusses the performance assessment of conventional and CCGT plants operating in the cycling regime by demonstrating the evolution of EFOF and EPOF during the plant life cycle i.e. starting from first commissioning until the end of the plant life. The evolution of EFOF and EPOF is also made for the plants operating in the base load mode and different types of cycling regimes to understand the impact of cycling on the O&M performance i.e. availability, reliability etc.

Polynomial graphs are plotted (below) for the evolution of EFOF & EPOF and the strength of variables for each relationship is shown by the coefficient of determination. Coefficient of determination (R2) (or correlation factor, R) is an overall measure of the accuracy of a statistical model. R2 ranges the values between 0 and 1 and the higher the R2 values, the better the strength of variables of the model. Table 5-1 provides the details of consideration made in this study for the strength of variables of data based on the R2 values. Tables 5-2 and 5-3 provides the polynomial equations and the details of strength of variables of data for the each relationship of EFOF & EPOF for conventional and CCGT plants, respectively.

Table 5-1
Consideration for the correlation of data

Range of R^2	Correlation of Data
$0.9 > R^2 \leq 1.0$	Strong positive correlation
$0.8 > R^2 \leq 0.9$	Moderately strong positive correlation
$0.6 > R^2 \leq 0.8$	Weak positive correlation
$0.4 > R^2 \leq 0.6$	Weak correlation
$0.0 > R^2 \leq 0.4$	Poor or No correlation

Table 5-2
Polynomial equations and strength of variables for conventional plants

Figure No.	Description	Polynomial Equation	Strength of Variables			
			R^2	R	Correlation	Remarks
Relationship for Equivalent Forced Outage Factor (EFOF)						
5-1	Average EFOF v Age (operating in the cycling regime)	$y = 0.0001x^2 - 0.0036x + 0.0544$	0.63	0.79	Weak positive	Correlation of data is slightly weak
5-2	Average EFOF v Age (operating in the base load mode)	$y = 7E\text{-}05x^2 - 0.0022x + 0.0383$	0.62	0.79	Weak positive	Correlation of data is slightly weak
5-5	Average EFOF v annual EHS (operating in the cycling regime)	$y = -2E\text{-}07x^2 + 0.0004x + 0.0232$	0.81	0.90	Moderately strong positive	Correlation of data is less strong
5-6	Average EFOF v lifetime EHS (operating in the cycling regime)	$y = 6E\text{-}09x^2 - 3E\text{-}06x + 0.0383$	0.82	0.91	Moderately strong positive	Correlation of data is less strong
5-7	Average EFOF v lifetime Starts for hot, warm and cold (operating in the cycling regime)	$y = 5E\text{-}08x^2 - 9E\text{-}06x + 0.0312$	0.82	0.91	Moderately strong positive	Correlation of data is less strong
5-8	Average EFOF v Age for various cycling regime	Sporadic, $y = 9E\text{-}05x^2 - 0.0024x + 0.048$	0.68	0.83	Weak positive	Correlation of data is slightly weak
		Weekend shutdown, $y = 0.0007x^2 - 0.0359x + 0.5405$	0.83	0.91	Moderately strong positive	Correlation of data is less strong
		-	-	-	-	No available data
		-	-	-	-	No available data
		Base load, $y = 7E\text{-}05x^2 - 0.0022x + 0.0354$	0.62	0.79	Weak positive	Less available data
		Load follow, $y = 1E\text{-}04x^2 - 0.0029x + 0.0451$	0.62	0.79	Weak positive	Less available data

Table 5-2 (continued)
Polynomial equations and strength of variables for conventional plants

<table>
<tr><th rowspan="2">Figure No.</th><th rowspan="2">Description</th><th rowspan="2">Polynomial Equation</th><th colspan="4">Strength of Variables</th></tr>
<tr><th>R^2</th><th>R</th><th>Correlation</th><th>Remarks</th></tr>
<tr><th colspan="7">Relationship for Equivalent Planned Outage Factor (EPOF)</th></tr>
<tr><td>5-13</td><td>Average EPOF v Age (operating in the cycling regime)</td><td>$y = 0.0002x^2 - 0.0045x + 0.0884$</td><td>0.72</td><td>0.85</td><td>Weak positive</td><td>Correlation of data is slightly weak</td></tr>
<tr><td>5-14</td><td>Average EPOF v Age (operating in the base load mode)</td><td>$y = 0.0002x^2 - 0.0048x + 0.0695$</td><td>0.76</td><td>0.87</td><td>Weak positive</td><td>Correlation of data is slightly weak</td></tr>
<tr><td>5-17</td><td>Average EPOF v annual EHS (operating in the cycling regime)</td><td>$y = 9E\text{-}07x^2 + 0.0002x + 0.0679$</td><td>0.51</td><td>0.71</td><td>Weak</td><td>Correlation of data is weak</td></tr>
<tr><td>5-18</td><td>Average EPOF v lifetime EHS (operating in the cycling regime)</td><td>$y = 4E\text{-}09x^2 + 1E\text{-}05x + 0.0331$</td><td>0.76</td><td>0.87</td><td>Weak positive</td><td>Correlation of data is slightly weak</td></tr>
<tr><td>5-19</td><td>Average EPOF v lifetime Starts for hot, warm and cold (operating in the cycling regime)</td><td>$y = 3E\text{-}07x^2 + 0.0001x + 0.0331$</td><td>0.76</td><td>0.87</td><td>Weak positive</td><td>Correlation of data is slightly weak</td></tr>
<tr><td rowspan="6">5-20</td><td rowspan="6">EPOF v Age for various cycling regime</td><td>Sporadic,
$y = 0.0002x^2 - 0.005x + 0.0909$</td><td>0.71</td><td>0.84</td><td>Weak positive</td><td>Correlation of data is slightly weak</td></tr>
<tr><td>Weekend shutdown,
$y = 0.0005x^2 - 0.0234x + 0.3539$</td><td>0.64</td><td>0.80</td><td>Weak positive</td><td>Less available data</td></tr>
<tr><td>-</td><td>-</td><td>-</td><td>-</td><td>No available data</td></tr>
<tr><td>-</td><td>-</td><td>-</td><td>-</td><td>No available data</td></tr>
<tr><td>Base load,
$y = 0.0002x^2 - 0.0048x + 0.0695$</td><td>0.75</td><td>0.87</td><td>Weak positive</td><td>Less available data</td></tr>
<tr><td>Load follow,
$y = 0.0003x^2 - 0.0099x + 0.1281$</td><td>0.63</td><td>0.79</td><td>Weak positive</td><td>Less available data</td></tr>
</table>

Table 5-2 (continued)
Polynomial equations and strength of variables for conventional plants

Figure No.	Description	Polynomial Equation	Strength of Variables			
			R^2	R	Correlation	Remarks
Relationship for Availability and Reliability						
5-25	Life cycle Availability and Reliability v Age for base load and cycling plants	Base load availability, $y = -0.0002x^2 + 0.0073x + 0.8933$	0.83	0.91	Moderately strong positive	Correlation of data is less strong
		Base load reliability, $y = -7E\text{-}05x^2 + 0.0024x + 0.9637$	0.69	0.83	Weak positive	Correlation of data is slightly weak
		Cycling availability, $y = -0.0003x^2 + 0.0083x + 0.863$	0.75	0.87	Weak positive	Correlation of data is slightly weak
		Cycling reliability, $y = -0.0002x^2 + 0.0044x + 0.9409$	0.70	0.84	Weak positive	Correlation of data is slightly weak

Table 5-3
Polynomial equations and strength of variables for CCGT plants

Figure No.	Description	Polynomial Equation	Strength of Variables			
			R^2	R	Correlation	Remarks
Relationship for Equivalent Forced Outage Factor (EFOF)						
5-3	Average EFOF v Age (operating in the cycling regime)	$y = 0.0003x^2 - 0.008x + 0.0731$	0.95	0.97	Strong positive	Correlation of data is highly strong
5-4	Average EFOF v Age (operating in the base load mode)	$y = 0.0002x^2 - 0.004x + 0.0344$	0.91	0.95	Strong positive	Correlation of data is highly strong
5-9	Average EFOF v annual EHS (operating in the cycling regime)	$y = 2E\text{-}07x^2 + 7E\text{-}05x + 0.0115$	0.84	0.92	Moderately strong positive	Correlation of data is less strong
5-10	Average EFOF v lifetime EHS (operating in the cycling regime)	$y = 3E\text{-}09x^2 + 2E\text{-}05x - 0.004$	0.82	0.91	Moderately strong positive	Correlation of data is less strong

Table 5-3 (continued)
Polynomial equations and strength of variables for CCGT plants

Figure No.	Description	Polynomial Equation	Strength of Variables			
			R^2	R	Correlation	Remarks
5-11	Average EFOF v lifetime Starts for hot, warm and cold (operating in the cycling regime)	$y = 9E\text{-}09x^2 + 4E\text{-}05x + 0.0048$	0.82	0.91	Moderately strong positive	Correlation of data is less strong
5-12	EFOF v Age for various cycling regime	Sporadic, $y = 0.0007x^2 - 0.016x + 0.1333$	0.82	0.91	Moderately strong positive	Correlation of data is less strong
		Weekend shutdown, $y = 0.0003x^2 - 0.007x + 0.0663$	0.74	0.86	Weak positive	Correlation of data is slightly weak
		Two shifting, $y = 0.0003x^2 - 0.0075x + 0.0788$	0.78	0.88	Weak positive	Correlation of data is slightly weak
		Double two shifting, $y = 0.0007x^2 - 0.016x + 0.1333$	0.85	0.92	Moderately strong positive	Correlation of data is less strong
		Base load, $y = 0.0002x^2 - 0.0039x + 0.0337$	0.88	0.94	Moderately strong positive	Correlation of data is less strong
		Load follow, $y = 0.0002x^2 - 0.0058x + 0.0494$	0.64	0.80	Weak positive	Correlation of data is slightly weak

Table 5-3 (continued)
Polynomial equations and strength of variables for CCGT plants

Figure No.	Description	Polynomial Equation	Strength of Variables			
			R^2	R	Correlation	Remarks
Relationship for Equivalent Planned Outage Factor (EPOF)						
5-15	EPOF v Age (operating in the cycling regime)	$y = 0.0003x^2 - 0.0062x + 0.0806$	0.89	0.94	Moderately strong positive	Correlation of data is less strong
5-16	Average EPOF v Age (operating in the base load mode)	$y = 0.0002x^2 - 0.0057x + 0.0644$	0.92	0.96	Strong positive	Correlation of data is highly strong
5-21	Average EPOF v annual EHS (operating in the cycling regime)	$y = 5E\text{-}07x^2 - 0.0002x + 0.0453$	0.75	0.87	Weak positive	Correlation of data is slightly weak
5-22	Average EPOF v lifetime EHS (operating in the cycling regime)	$y = 4E\text{-}09x^2 + 1E\text{-}05x + 0.0331$	0.76	0.87	Weak positive	Correlation of data is slightly weak
5-23	Average EPOF v Starts/year for hot, warm and cold (operating in the cycling regime)	$y = 3E\text{-}07x^2 + 0.0001x + 0.0331$	0.76	0.87	Weak positive	Correlation of data is slightly weak
5-24	EPOF v Age for various cycling regime	Sporadic, $y = 0.0003x^2 - 0.0057x + 0.0654$	0.84	0.92	Moderately strong positive	Correlation of data is less strong
		Weekend shutdown, $y = 0.0003x^2 - 0.0069x + 0.0801$	0.79	0.89	Weak positive	Correlation of data is slightly weak
		Two shifting, $y = 0.0003x^2 - 0.0065x + 0.0896$	0.87	0.93	Moderately strong positive	Less available data
		Double two shifting, $y = 0.001x^2 - 0.0196x + 0.1524$	0.75	0.87	Weak positive	Less available data
		Base load, $y = 0.0002x^2 - 0.0057x + 0.0565$	0.92	0.96	Strong positive	Correlation of data is highly strong
		Load follow, $y= 0.0003x^2 - 0.0064x + 0.0633$	0.62	0.79	Weak positive	Less available data

Table 5-3 (continued)
Polynomial equations and strength of variables for CCGT plants

Figure No.	Description	Polynomial Equation	Strength of Variables			
			R^2	R	Correlation	Remarks
Relationship for Availability and Reliability						
5-26	Life cycle Availability and Reliability v Age for base load and cycling plants	Base load availability, $y = -0.0004x^2 + 0.0096x + 0.9097$	0.95	0.97	Strong positive	Correlation of data is highly strong
		Base load reliability, $y = -0.0002x^2 + 0.004x + 0.9656$	0.91	0.95	Strong positive	Correlation of data is highly strong
		Cycling availability, $y = -0.0007x^2 + 0.0165x + 0.8369$	0.89	0.94	Moderately strong positive	Correlation of data is less strong
		Cycling reliability, $y = -0.0003x^2 + 0.008x + 0.9269$	0.94	0.97	Strong positive	Correlation of data is highly strong

Note: Data for average EFOF and EPOF were collected for the last 3 to 10 years of operation for each plant. Therefore, average EFOF and EPOF represents the 'data collection period' (last 3 to 10 years of operation) only.

5.1 Equivalent Forced Outage Factor

EFOF is the fraction of a given operation period in which a plant or a train *is not available due to forced outages*. This particular parameter is very useful in measuring forced outages in cycling power plants since it takes into account the derating hours. Furthermore EFOF is also an important factor used during performance assessment, as it allows conducting a direct comparison with other similar generation units.

Note that according to US NERC (North American Electric Reliability Corporation) GADS (Generating Availability Data System) outage definitions.

Forced Outage (FO) is the period when the unit is forced to go offline for repairs that cannot be postponed until beyond the end of the next weekend.

Maintenance Outage (MO) is an outage that can be deferred beyond the end of the next weekend but requires that the unit be removed from service, another outage state, or Reserve Shutdown state before the next Planned Outage (PO).

ETD database contains plants' data from its worldwide (i.e., North America, Europe & Asia) collaboration. During the survey, North American plants provided the maintenance outage data together with forced outage data as the total forced outage hours for a specific year. Therefore, separate maintenance outage data were not available for these plants. European and Asian plants consider all type of unplanned outage (outage which is not planned at least one year before) as forced outage events. So that unplanned outage data similar to maintenance outage are also included in the forced outage data for these plants. ***Therefore, it can be stated that the forced outage hours data considered in this study also includes the maintenance outage hours data for all plants.***

The equivalent forced outage factor can be calculated using the following formula:

$$EFOF = \left(\frac{FOH + EFDH}{PH} \right) \times 100\%$$

Where:

EFOF = Equivalent forced outage factor [%]

FOH = ‘Forced outage hour’ is the sum of all hours experienced during forced outage [hrs] According to IEEE 762-2006 this (FOH) represents the number of hours a unit is in:

1. Start-up failure (trip)
2. Condition requiring immediate outage
3. Extension of forced outages due to unplanned work

PH = Period hours, which is the number of hours a unit is in the active state = 8760 hrs/year

EFDH = ‘Equivalent forced derated hours’ is a conversion of the load reduction time due to a failure in equivalent hours of forced outage and can be determined by the following relationship:

$$EFDH = \frac{\sum_{i=1}^{n} FD_i \times T_i}{MC}$$

Where:

FD = Forced derated states

T = Number of hours accumulated during the period the unit was operating with reduced load due to damage/ failure of any component causing a generation restriction [hrs]

MC = Maximum capacity [MW]

Note: To measure the forced outage performance, another metric used in industry which is similar to EFOF is the EFOR (Equivalent Forced Outage Rate), and is defined by ‘A measure of the probability that a generating unit will not be available due to forced outages when there is demand on the unit to generate’. ***EFOR has not been used in this particular study.***

EFOR can be calculated by using following equation:

$$\boxed{EFOR = \frac{FOH + EFDH}{FOH + SH} \times 100}$$

Where:

FOH = Forced outage hours, which is the sum of all hours experienced during forced outage.

According to IEEE 762-2006 this represents the number of hours a unit is in:

1. Start-up failure (trip)
2. Condition requiring immediate outage
3. Extension of planned outages due to unplanned work

SH = Service Hours, total service hours since commissioning

The determination of forced outage level is an important step to determine the reliability performance since an unexpected event due to cycling such as a failure will consequently affect the reliability of a power plant. A detailed analysis of the results (see the following figures) shows that the cycling operation generates a considerable impact in the plant performance level (i.e. availability, reliability etc). From the analysis, three distinct areas are identified for the entire life cycle of a power plant operating either in base load or cycling regime. These three areas characterize the different forced outage levels, for the entire life cycle of the plant as follows.

Commissioning period: This is identified to be approximately between 1st to 6th years of operation. During this period the plant shows somewhat higher EFOF value due to increased number of outages resulting from the installation problems.

Useful life period: This is considered to be the most important period for a plant since this is the period where the lower failure rate results lower outages and where most of the generation profit is made. For a plant, this period is estimated to be located approximately between ~ 6 to 20 years of operational life. During this period the EFOF value starts to decrease and is at its lowest level between 10 to 20 years (for conventional plants) and 14 to 20 years (for CCGT plants) resulting from reduced number of installation problems and other related factors such as improvement of knowledge of operation, maturity of the technology, improvement of maintenance activities etc.

Major component wear out period: This period represents near the end of plant life i.e. the period of time when the failure rate increases due to component life exhaustion. This period is identified to be approximately after 20 years of operation and during this period EFOF value starts to increase (assuming that the major components that are near or end of life have not been replaced). The EFOF can be reduced by implementing improved maintenance techniques characterized by high preventive maintenance (PM), predictive maintenance (PdM) and even high proactive maintenance (PaM) contributing to a reduced failure rate and consequently reduced downtime (discussed later).

Figure 5-1 represents the average levels of forced outage for the entire life cycle of a ***conventional plant*** operating in the cycling regime (considered all types of cycling regime). The EFOF polynomial (second order) curve for conventional cycling plants shows a weak positive correlation of the data, with a coefficient of determination (R^2) 0.63. According to this figure, during the 'commissioning period', a plant operating in the cycling regime shows an EFOF value of ~ 3.5% to 5.5%. For a cycling plant, during the 'useful life period' the EFOF value is observed to be between ~ 2% to 3.5% and the EFOF value achieves its minimum between 10 to 20 years. During the 'major component wear out period' which is near the end of life, EFOF value for cycling conventional plants increases to more than ~ 9%.

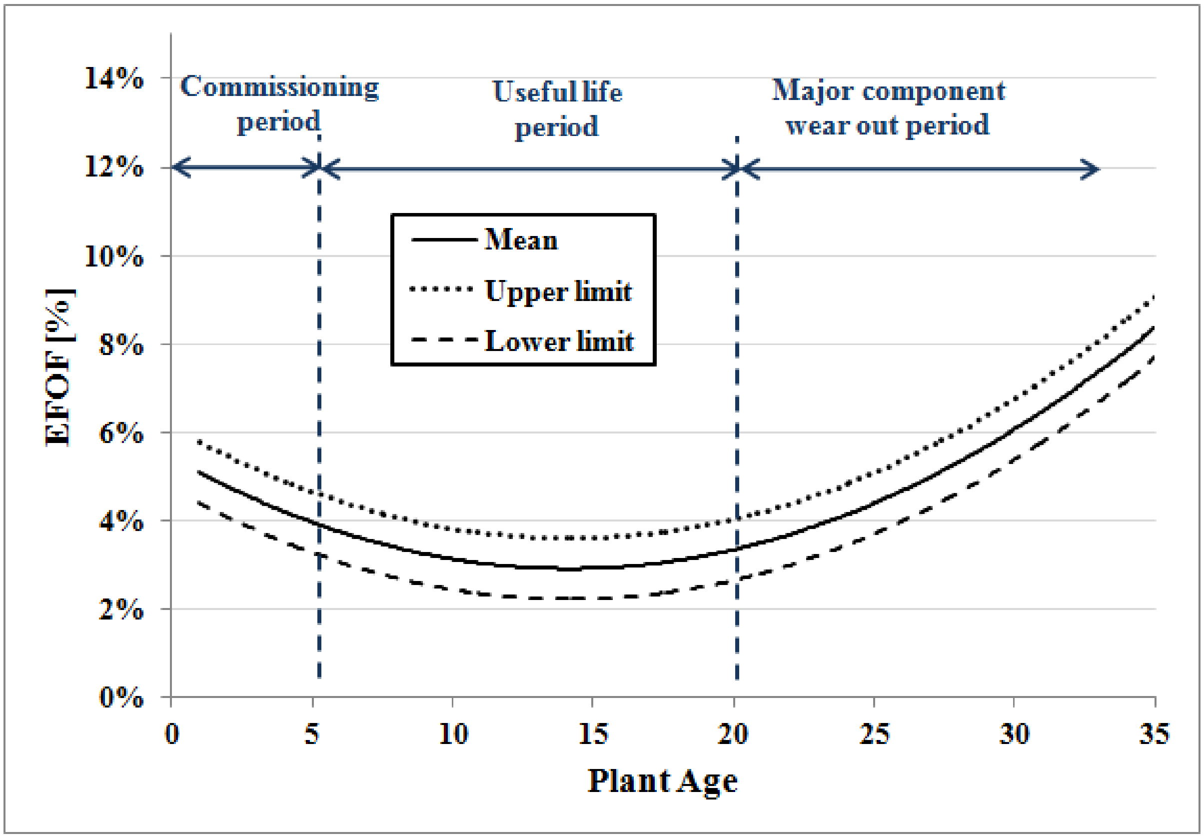


Figure 5-1
Life cycle EFOF v. Age for conventional plants operating in the cycling regime

Figure 5-2 shows comparison of the average EFOF levels between cycling and base load plants. The grey lines (upper, lower and mean value) in the figure represent the expected forced outage levels for a conventional plant operating in the base load mode. This has been included in the same figure with cycling curve to provide a comparison between cycling and base load plants forced outages which provide a clear understanding of the reliability impact on the plant due to cycling. The polynomial trend line for conventional base load plants shows a weak positive correlation of the data, with a coefficient of determination (R^2) 0.62. According to Figure 5-2, the average EFOF value for conventional plants operating in the cycling regime is ~ 1.5% higher than the plants operating in the base load mode in the first 6 years of operation, and ~ 1.0% higher between 6 to 20 years of operation. EFOF for the cycling plants increases much more abruptly between 20 to 35 years of operation compared with the base load plants. This increase has been estimated to be as much as ~ 5% which means that a cycling conventional power plant life should be limited to about 25 to 35 years and that it may be uneconomical to run a cycling plant beyond this period (assuming major components at or near end of life are not replaced). This finding will be further explored and discussed in the subsequent Sections of this report.

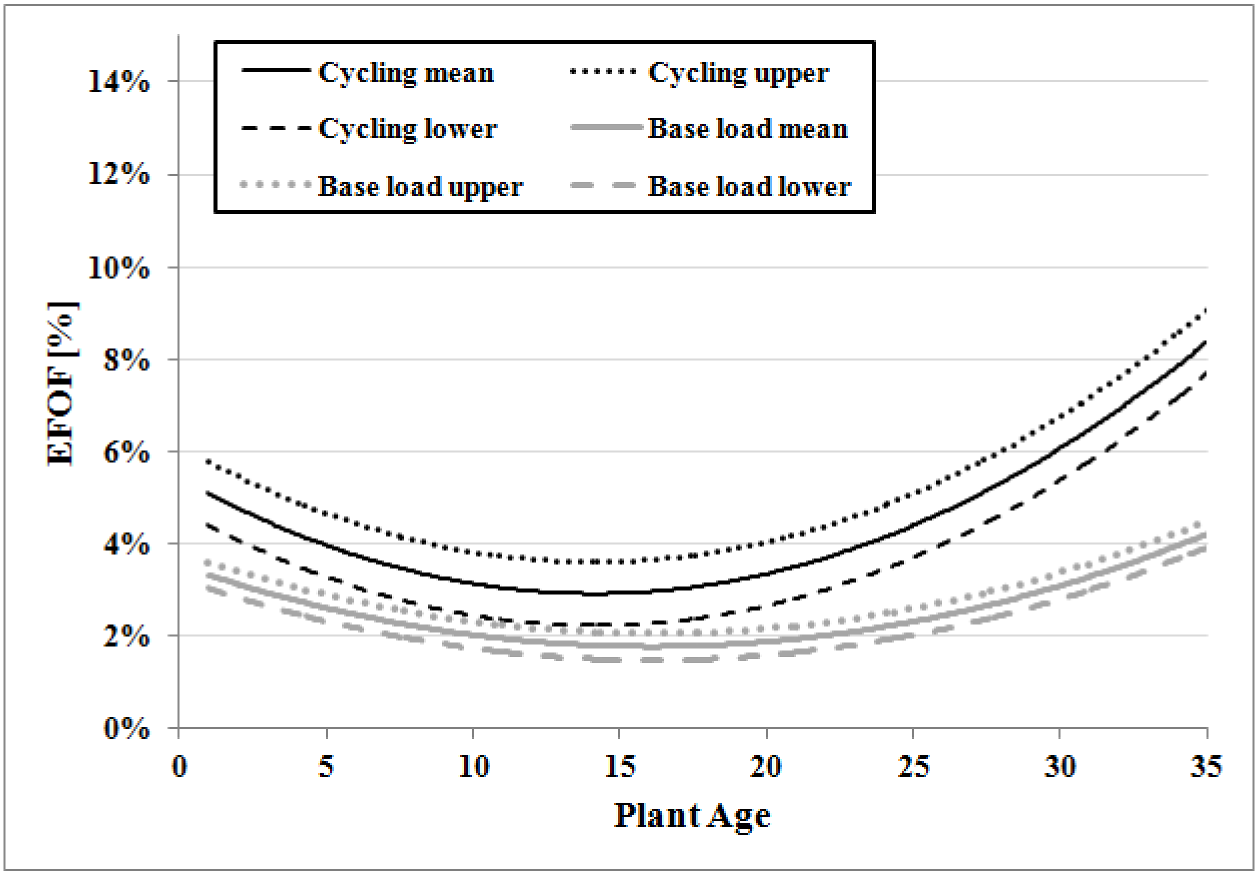


Figure 5-2
Life cycle EFOF v. Age for conventional plants operating in the base load mode and cycling regime

Figure 5-3 represents the average levels of forced outage for the entire life cycle of a ***CCGT plant*** operating in the cycling regime (considered all types of cycling regimes). The polynomial curve for CCGT cycling plants shows a strong positive correlation of the data, with a coefficient of determination (R^2) 0.94. According to this figure, during the 'commissioning period', a plant operating in the cycling regime shows an EFOF value of ~ 4 % to 6%. For a cycling plant, during the 'useful life period' the EFOF value is observed to be between ~ 1.8% to 3.8% and the EFOF value achieves its minimum value between 10 to 20 years. During the 'major component wear out period' which is near the end of life, EFOF value for cycling CCGT plants increases within the range of 12 ~ 14%.

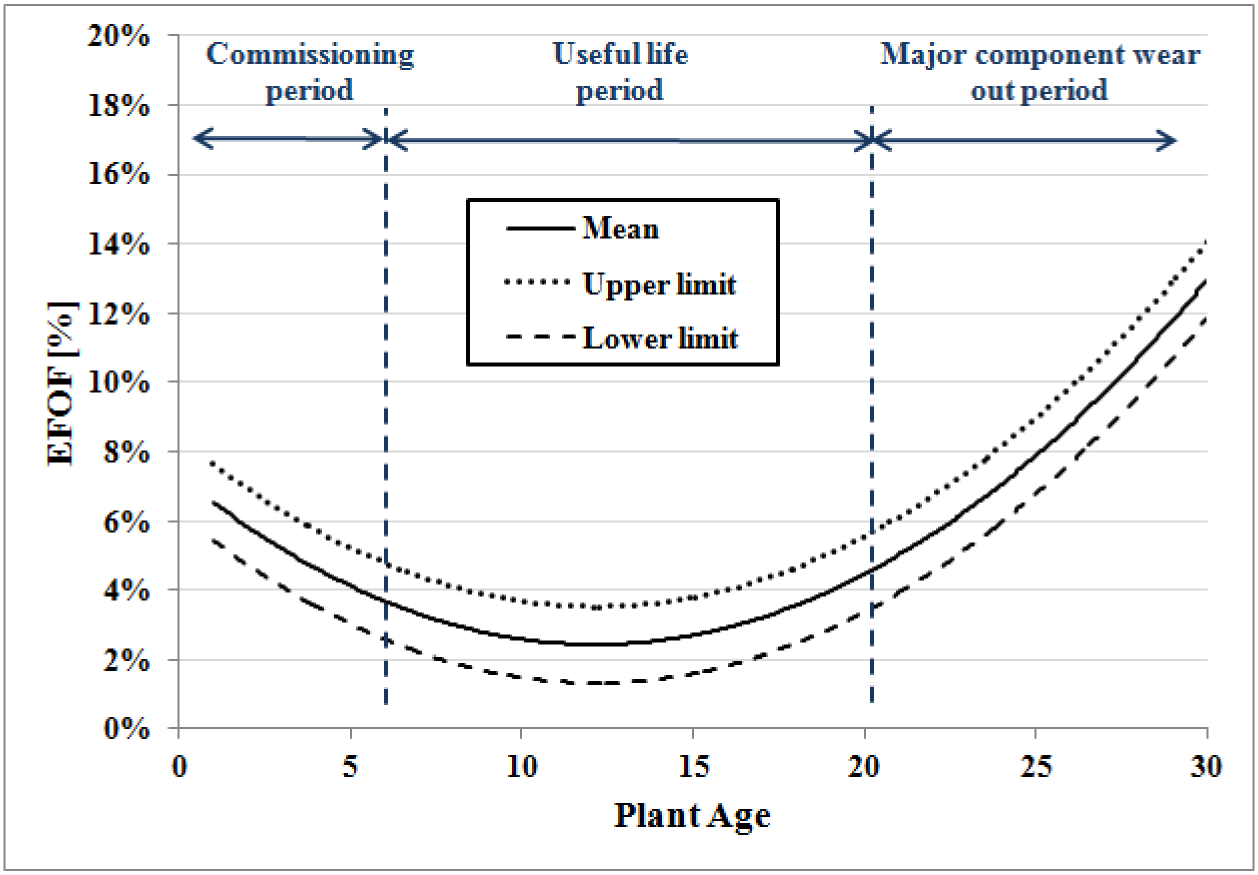


Figure 5-3
Life cycle EFOF v. Age for CCGT plants operating in the cycling regime

Figure 5-4 shows comparison of the average EFOF levels between cycling and base load plants. The grey lines (upper, lower and mean value) in the figure represent the expected forced outage levels for a CCGT plant operating in the base load mode. This has been included in the same figure with cycling curve to provide a comparison between cycling and base load plants forced outages which provide a clear understanding of the reliability impact on the plant due to cycling. The polynomial curves obtained from the statistical analysis for base load plant data show a strong positive correlation, with a coefficient of determination (R^2) in this case of 0.91. According to Figure 5-4, the average EFOF value for CCGT plants operating in the cycling regime is ~ 3% higher than the plants operating in the base load mode in the first 6 years of operation and ~ 1.5% higher between 6 to 20 years of operation. EFOF for the cycling plants increases much more abruptly between 20 to 30 years of operation compared with the base load plants. This increase has been estimated to be as much as ~ 6.5% which means that a cycling CCGT plant life should be limited to about 20 to 25 years and that it may be uneconomical to run a cycling plant beyond this period (assuming major components at or near end of life are not replaced).

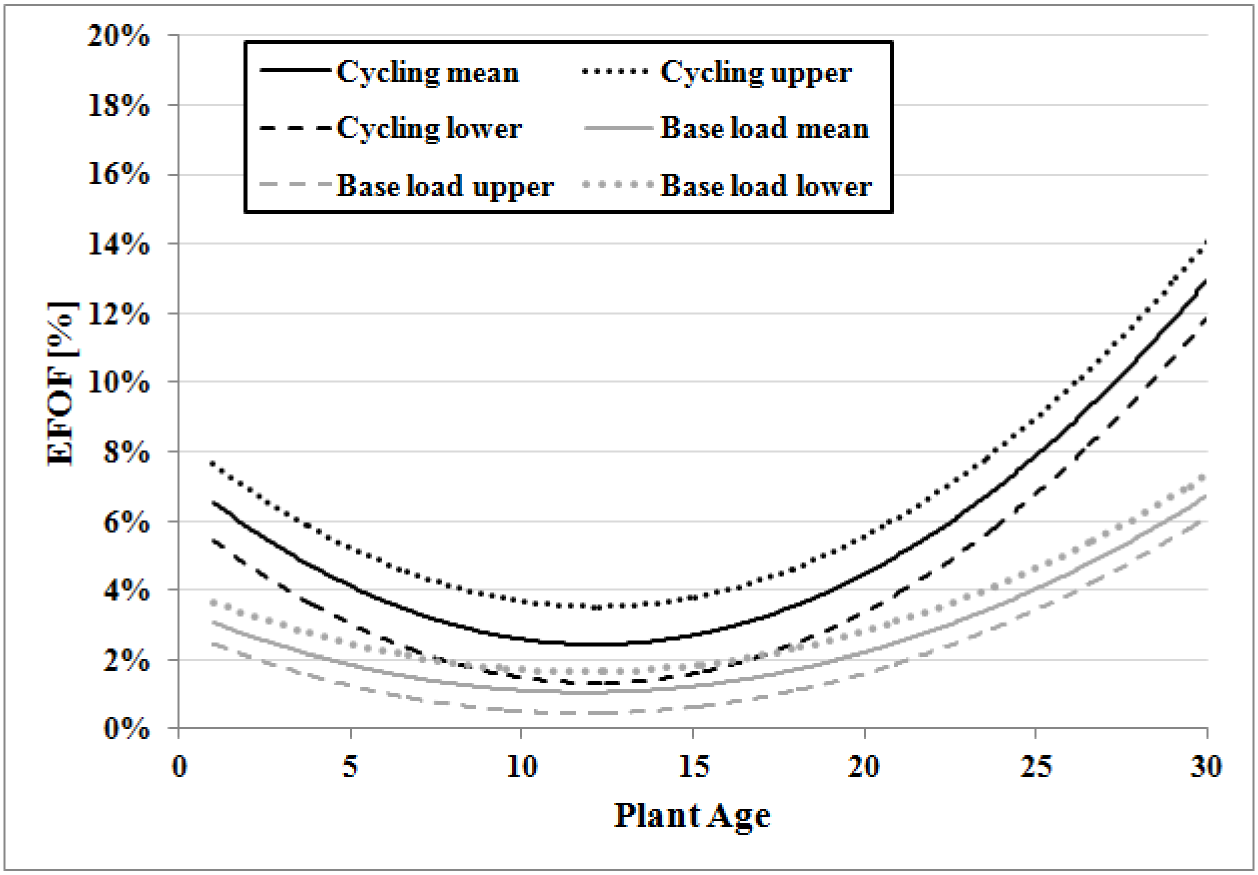


Figure 5-4
Life cycle EFOF v. Age for CCGT plants operating in the base load mode and cycling regime

5.2 Equivalent Forced Outage Factor for Various Cycling Regimes

This Subsection describes the EFOF levels for different types of cycling regimes (e.g. sporadic, weekend shutdown, two shifting, double two shifting etc.) for the entire life cycle of a plant.

***Note** that the relationship of average EFOF & EPOF and lifetime online hours shows poor correlation of data. Therefore, no assessment has been made for these relationships. The polynomial and the correlation of data for online hours were presented in **Appendix A**, Figures A-7 to A-10.* It is already mentioned in the previous Section that the cost analysis performed in this study is mainly based on the number of starts. Therefore, the performance analysis is also performed based on the similar parameter.

Annual EHS was calculated from the lifetime EHS for all units in the database to determine the relationship of average Equivalent Forced Outage Factor (EFOF) levels for various types of cycling regimes. ***Note that types of various cycling regimes were identified from the annual EHS based on the definition provided in Section 4.*** Load following plants for the performance analysis were identified from the information provided by the client. There were no available cost data for load following cycling regime. In **Appendix A**, lifetime EHS was plotted against lifetime online hours where color code is used to identify the plants operating under different cycling regimes. Figure 5-5 shows the average EFOF levels versus annual EHS for a cycling (considering all types of cycling modes) ***conventional plant*** and the polynomial curve obtained from the statistical analysis shows a moderately strong positive correlation, with a coefficient of determination (R^2) in this case of 0.81. According to this figure, a conventional plant operating in the 'sporadic cycling regime' shows an average EFOF value of ~ 2.0% to 4.0%, if the plant operates in the 'weekend shutdown regime' the average value for EFOF is ~ 6.5.0% to 8.0%,. The maximum annual EHS calculated for all conventional units in the database is 245. No data were available for the plants operating under two shifting and double two shifting cycling regimes. Average EFOF versus lifetime EHS was also plotted to present a better view to understand the impact of cycling or number starts on the EFOF performance, with a coefficient of determination (R^2) 0.82, shown in Figure 5-6. The figure clearly demonstrates that the failure rate increases with the increase in start-up number, also shown in Figure 5-7.

Figure 5-7 represents the EFOF levels versus lifetime number of starts which was calculated for all conventional units in terms of hot, warm and cold starts. The polynomial curve for all three types of starts shows a moderately strong positive correlation, with a coefficient of determination (R^2) 0.82. Overall, all three trend lines demonstrated the fact that the average EFOF levels increase with increase in the number of start-ups. The figure also indicated that the cold start is more damaging.

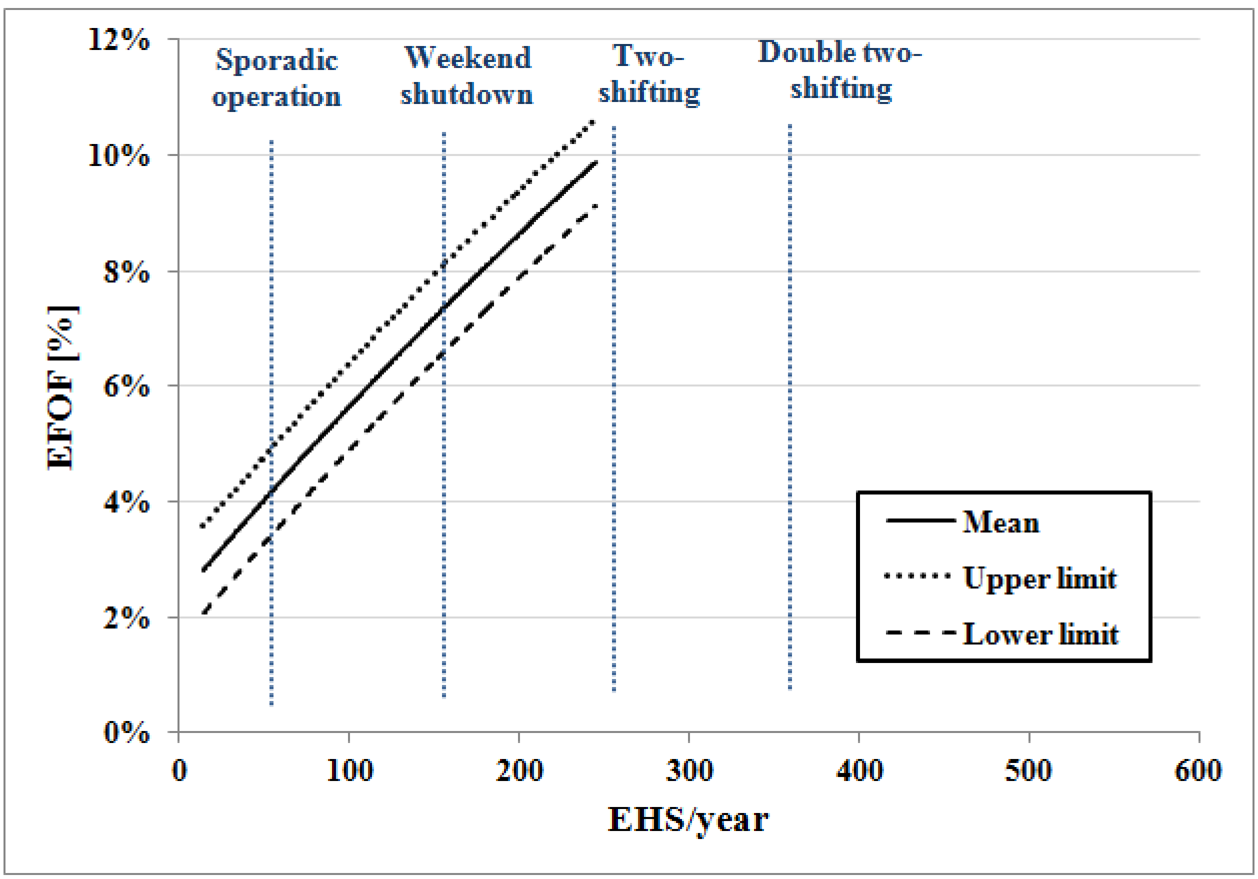


Figure 5-5
Average EFOF v. annual EHS for conventional plants operating in the cycling regimes

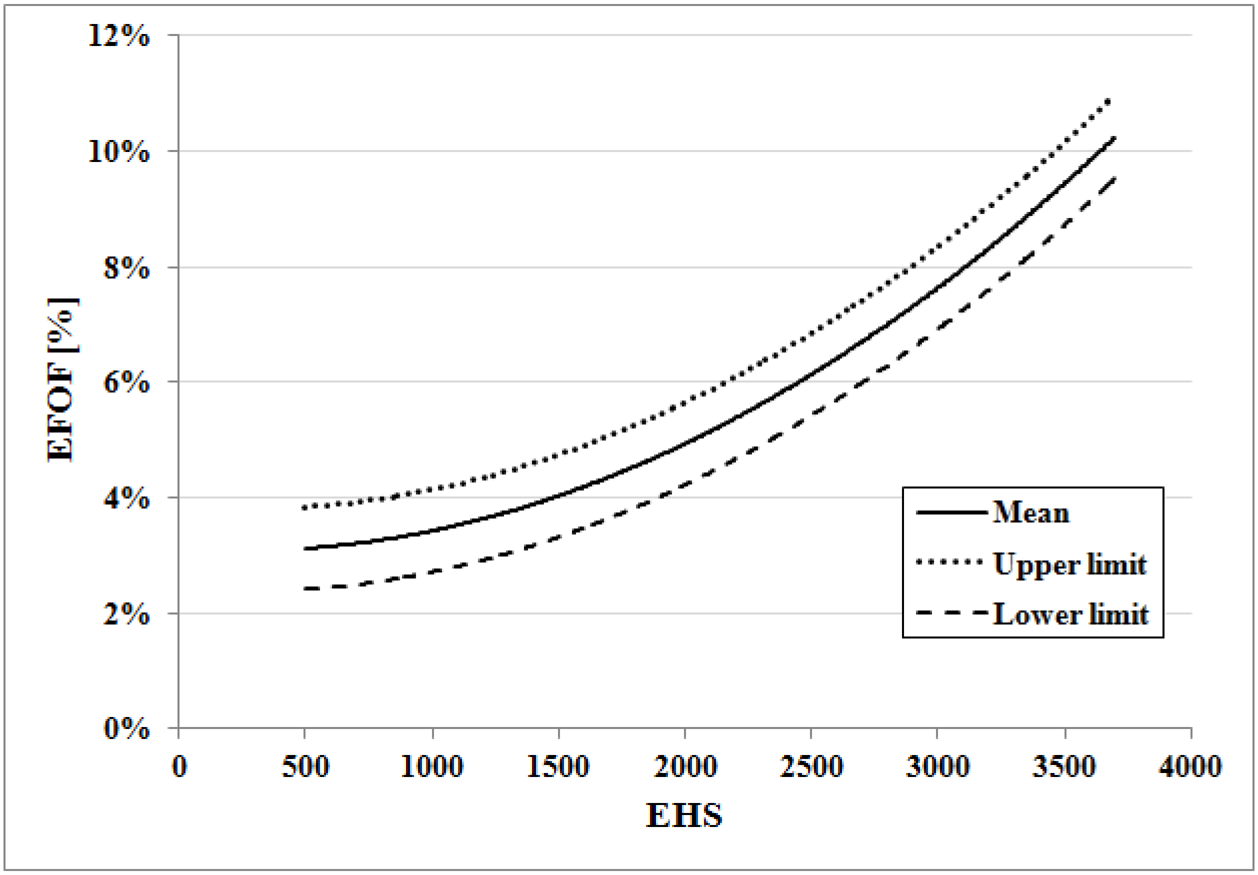


Figure 5-6
Average EFOF v. lifetime EHS for conventional plants operating in the cycling regimes

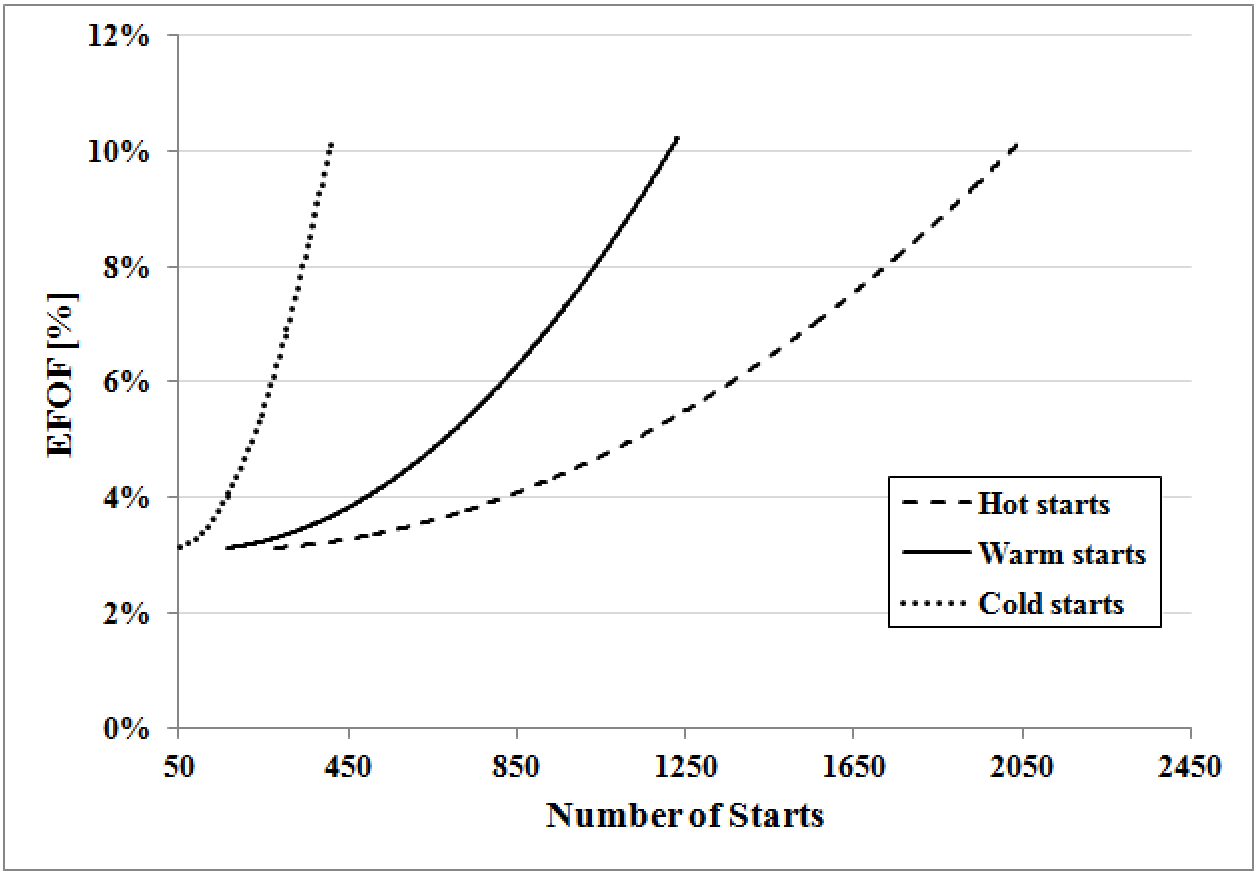


Figure 5-7
Average EFOF v. lifetime hot, warm and cold starts for conventional plants operating in the cycling regimes

The data were separated according to different types of cycling regimes and the relationships were plotted for the average EFOF levels for conventional plants, shown in Figure 5-8. As mentioned above, no data were available for two shifting and double two shifting cycling regimes for conventional units. The data for sporadic operation covers the age range between 3 to 31 years of operation and shows a weak positive correlation, with a coefficient of determination (R^2) 0.68, whereas the data for weekend shutdown operation covers the age range only from 30 to 35 years of operation and shows a moderately strong positive correlation, with a coefficient of determination (R^2) 0.83. The data for load following cycling operation covers the age range from 17 to 31 years of operation and shows a weak positive correlation, with a coefficient of determination (R^2) 0.62. The polynomial trend line for sporadic operation represents that the average value for EFOF is ~ 4% for the first 6 years, ~ 3.5% for 7 to 20 years and the value increases after 20 years. The polynomial trend line for weekend shutdown operation shows that the average value for EFOF increases at the 'major component wear out period' where the data only covers the age range of 30 to 35 years. The polynomial trend line for load following operation represents that the average value for EFOF increases after 20 years of operation. The figure also includes the trend line for base load operation to demonstrate the impact of cycling operation.

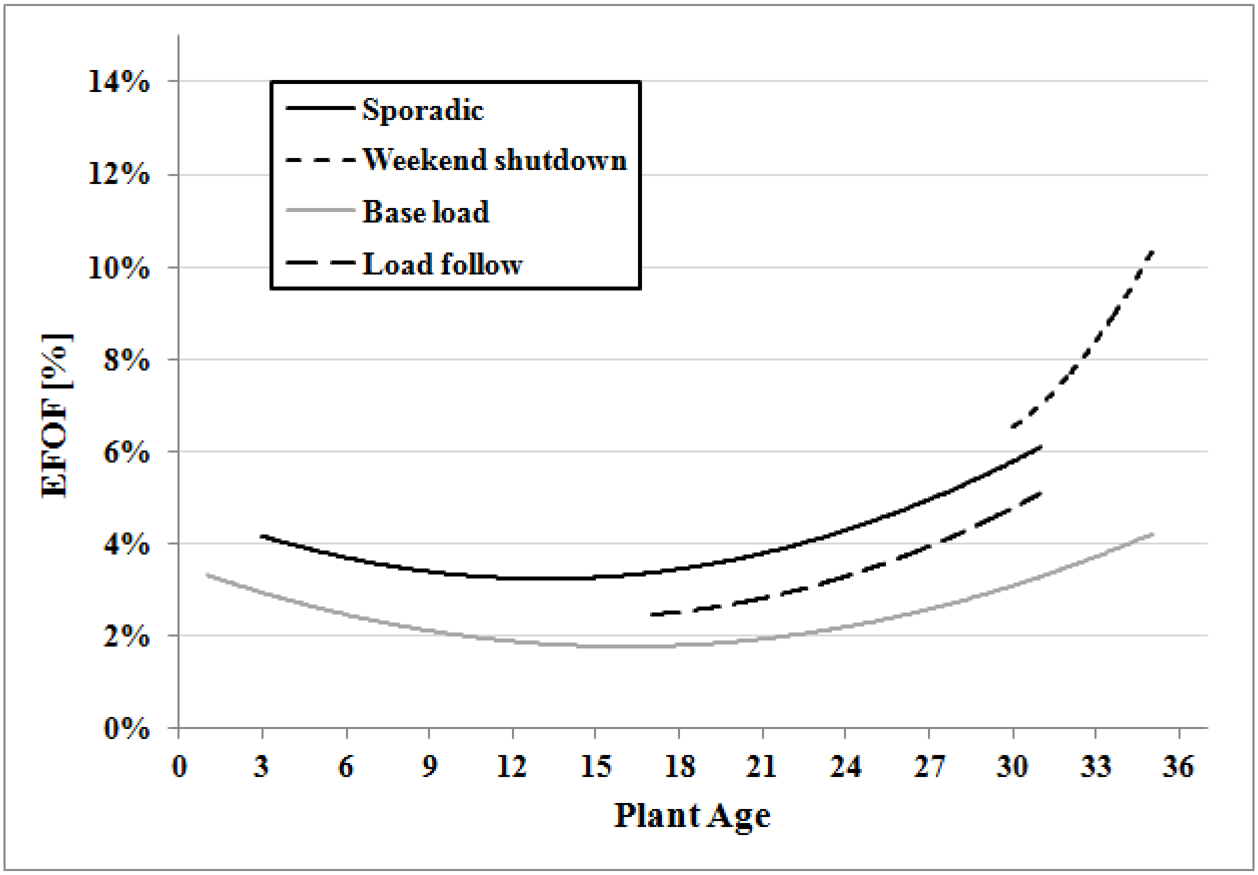


Figure 5-8
Average EFOF v. Age for conventional plants operating in the various cycling regimes

Figure 5-9 shows the average EFOF levels versus annual EHS for a cycling (considering all types of cycling modes) CCGT plant and the polynomial curve obtained from the statistical analysis shows a moderately strong positive correlation, with a coefficient of determination (R^2) in this case of 0.84. According to this figure, a CCGT plant operating in the 'sporadic cycling regime' shows an average EFOF value of ~ 0.5% to 3.0%, if the plant operates in the 'weekend shutdown regime' the average value for EFOF is ~ 2.0% to 4.0%, if the plant operates in the 'two shifting regime' the average value for EFOF is ~ 3.5% to 5.0% and if the plant operates in the 'double two shifting regime' the average value for EFOF increases to ~ 7.5% and above. Figure 5-9 also shows that the more damaging type of cycling is the 'double two-shifting'. Average EFOF versus lifetime EHS was also plotted to present a better view to understand the impact of cycling or number starts on the EFOF performance, with a coefficient of determination (R^2) 0.82, shown in Figure 5-10. The figure clearly demonstrates that the failure rate increases with the increase in start-up number, also shown in Figure 5-11.

Figure 5-11 represents the EFOF levels versus lifetime number of starts which was calculated for all conventional units in terms of hot, warm and cold starts. The polynomial curve for all three types of starts shows a moderately strong positive correlation, with a coefficient of determination (R^2) 0.82. Overall, all three trend lines demonstrated the fact that the average EFOF levels increase with increase in the number of start-ups. The figure also indicated that the cold start is more damaging.

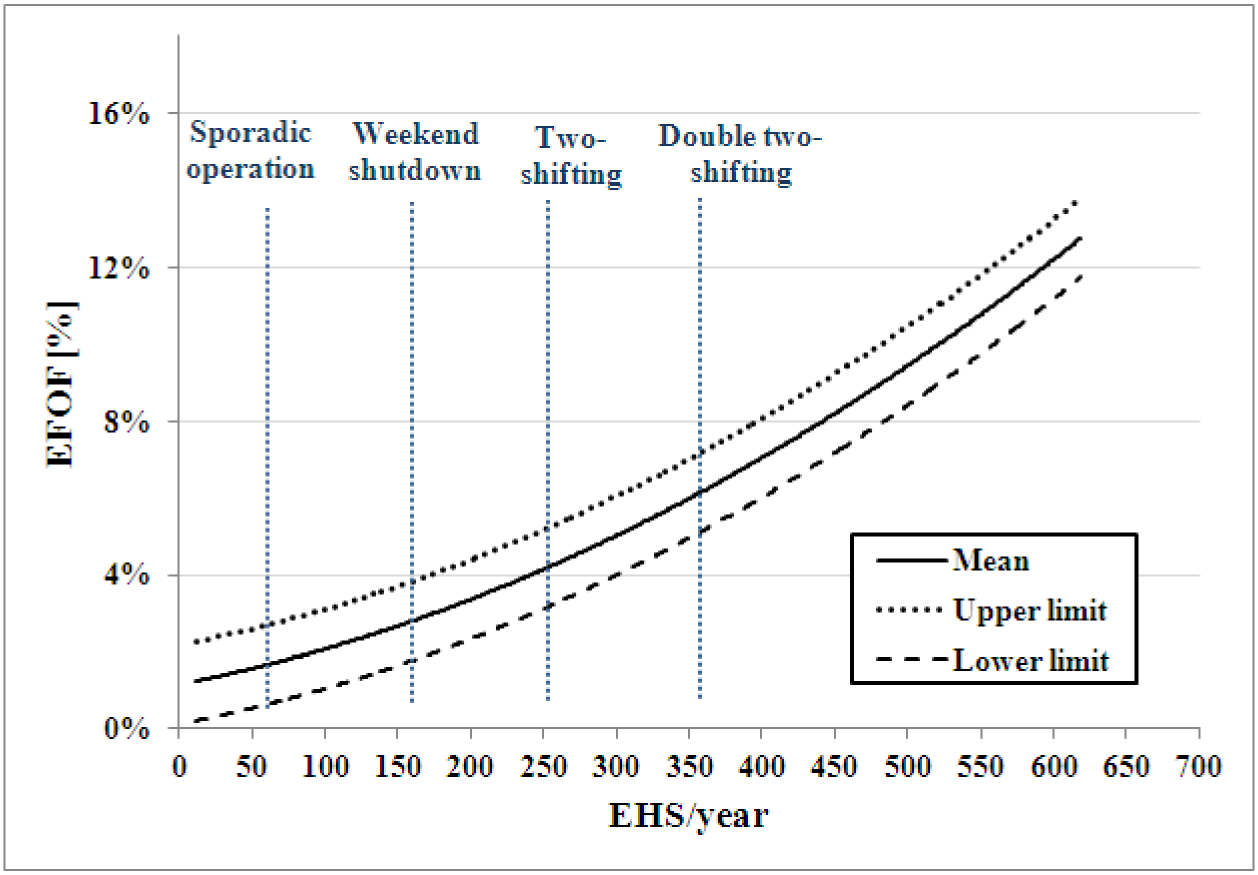


Figure 5-9
Average EFOF v. annual EHS for CCGT plants operating in the cycling regimes

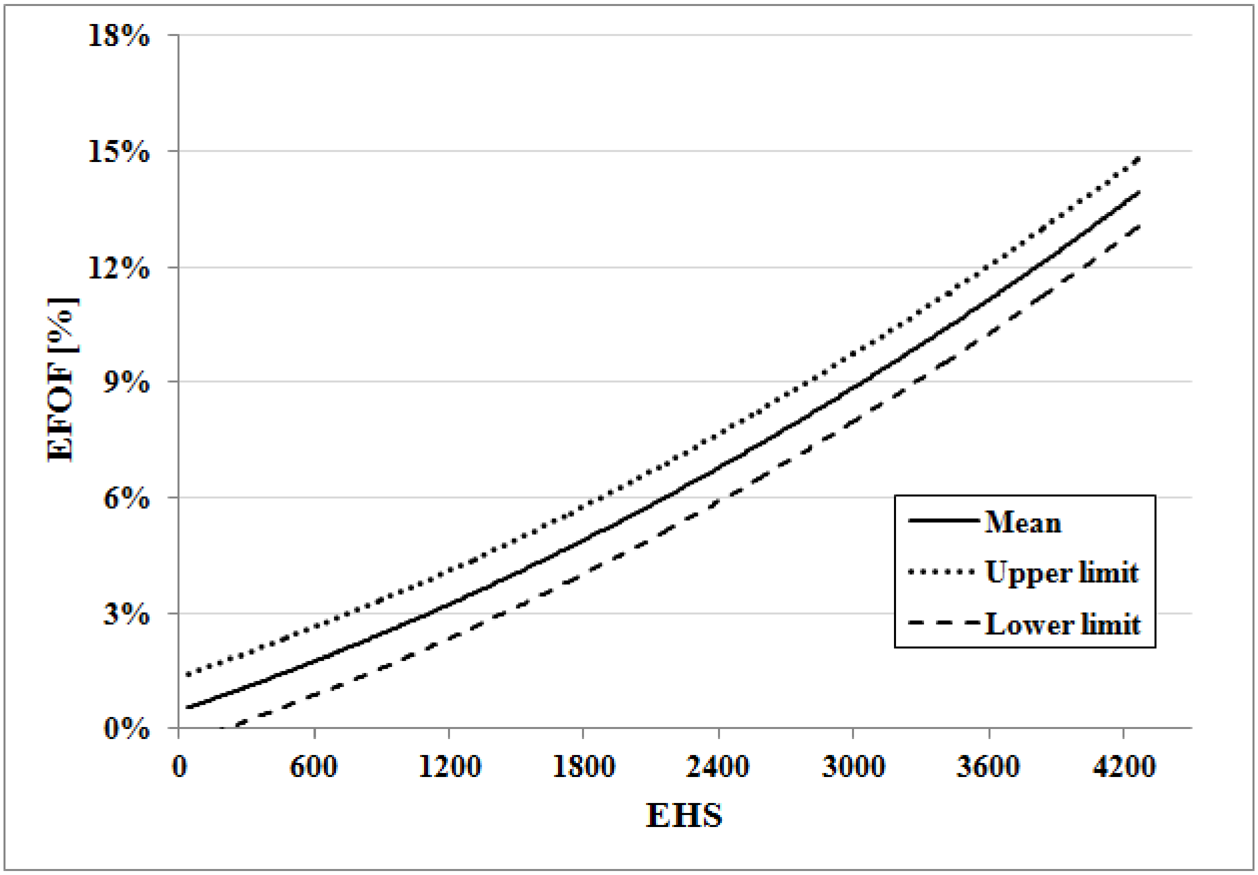


Figure 5-10
Average EFOF v. lifetime EHS for CCGT plants operating in the cycling regimes

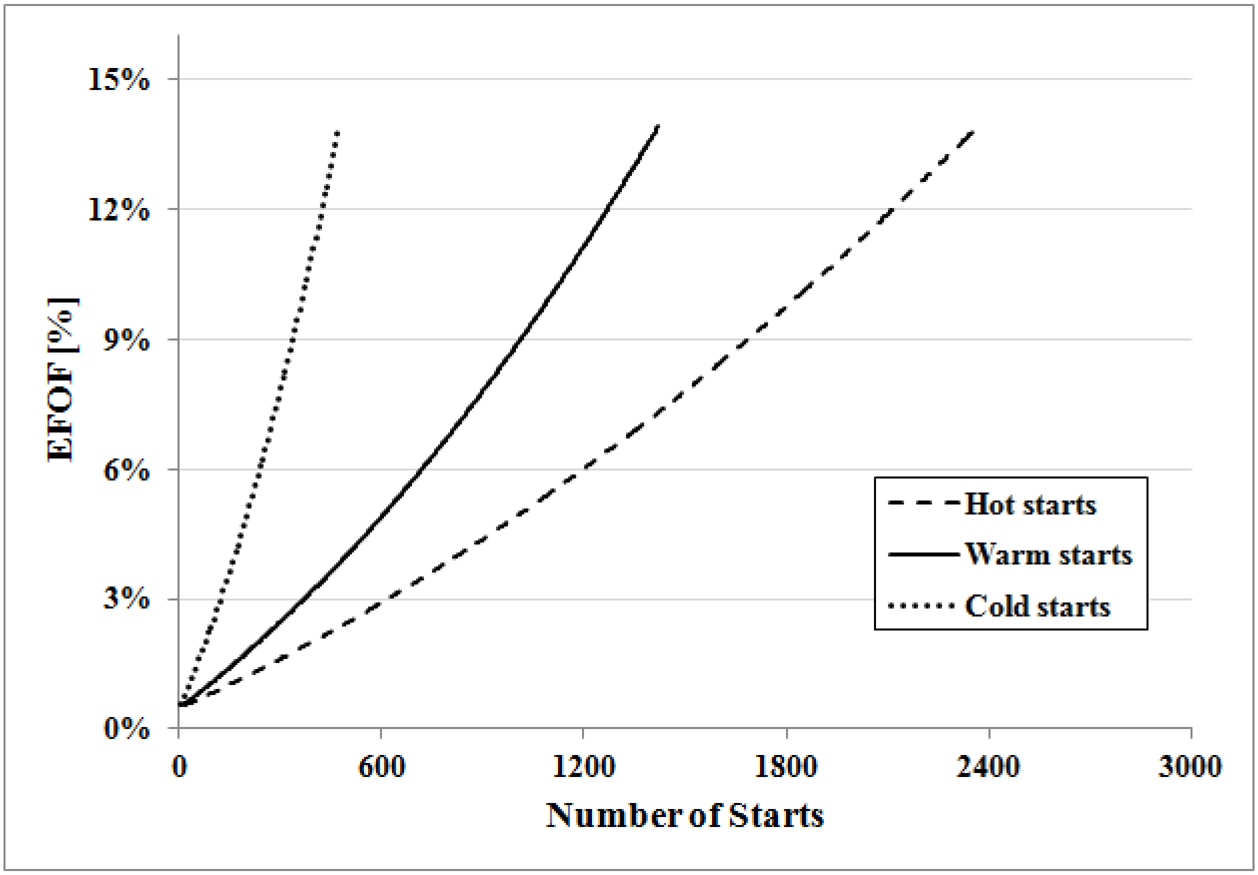


Figure 5-11
Average EFOF v. lifetime hot, warm and cold starts for CCGT plants operating in the cycling regimes

The data were separated for different types of cycling regimes and the relationships were plotted for the annual average EFOF levels for CCGT plants, shown in Figure 5-12. The data for *sporadic operation* covers the age range between 1 to 29 years of operation and shows a moderately strong positive correlation, with a coefficient of determination (R^2) 0.82. The data for *weekend shutdown* and *double two shifting operation* covers the age range only from 1 to 15 years of operation and shows a weak positive correlation, with a coefficient of determination (R^2) 0.74 and moderately strong positive correlation, with a coefficient of determination (R^2) 0.85, respectively. The data for *two shifting operation* covers the age range from 1 to 7 years and 23 to 30 years of operation and shows a weak correlation, with a coefficient of determination (R^2) 0.40. The data for *load following cycling operation* covers the age range from 1 to 19 years of operation and shows a weak positive correlation, with a coefficient of determination (R^2) 0.64. It is observed from the figure that for all types of cycling regimes the general trend for EFOF level is that it is slightly higher during the' commissioning period', then gradually reduces to a certain level during the 'useful life period' and again starts to increase after 20 years of operation during the 'major component wear out period'. For *double two shifting* plants it is not possible to confirm whether the plants only last 16 years or more as there are no available information beyond this age limit for this type of plants.

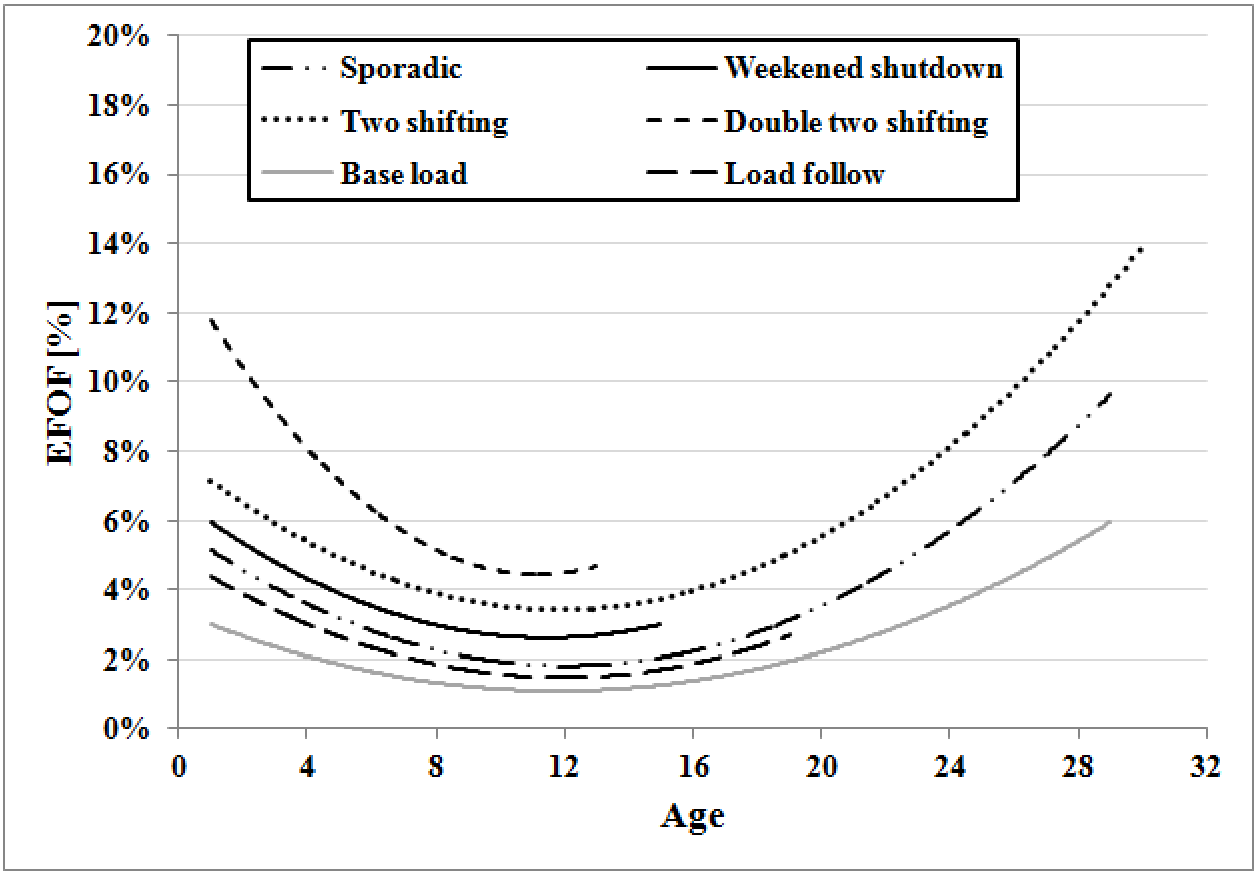


Figure 5-12
Average EFOF v. Age for CCGT plants operating in the various cycling regimes

5.3 Equivalent Planned Outage Factor

Planned outages normally refer to the removal of a unit from service to perform work on specific components that is scheduled well in advance and has a predetermined duration such as annual overhaul, inspection, component testing etc. In general, increased routine maintenance is required due to increased levels of wear and tear when a plant moves from base load operation to cyclic mode. Therefore, Equivalent Planned Outage Factor (EPOF) is useful to measure the planned outage performance which allows understanding of the impact of cycling on the plant availability.

For an individual generation unit the EPOF can be calculated using the formula shown below.

$$EPOF = \left(\frac{POH + EPDH}{PH} \right) \times 100\%$$

Where:

POH = Planned Outage Hours which is the sum of all hours experienced during planned/ scheduled outage.

PH = Period Hours (8760 hrs/year) (as previously defined)

EPDH = Equivalent planned derated hours which takes into account the maintenance performed during plant reduced load operation. The EPDH is calculated using the following equation:

$$EPDH = \frac{\sum_{i=1}^{n} PD_i \times T_i}{MC}$$

Where:

PD = Planned derated states

T = Number of hours accumulated during the period the unit was operating with reduced load due to planned maintenance [hrs]

MC = Maximum capacity [MW]

Figure 5-13 represents the average levels of planned outage for the entire life cycle of a ***conventional plant*** operating in the cycling regime (considering all types of cycling regimes). The EPOF polynomial curve obtained from the statistical analysis for cycling conventional plants data show a weak positive correlation, with a coefficient of determination (R^2) in this case of 0.72. According to this figure, the planned outage levels for cycling conventional plants are within ~ 7.5 to 9.5% during the first 6 years of operation, within ~ 5.0 to 7.0% for the next 14 years of operation and the EPOF achieves its minimum levels (~ 5.0%) between 12 to 15 years. This Figure also shows that after 25 years of operation the EPOF value increases significantly (~ 14-18%) that affects the generation level resulting in reduced availability and increased generation loss.

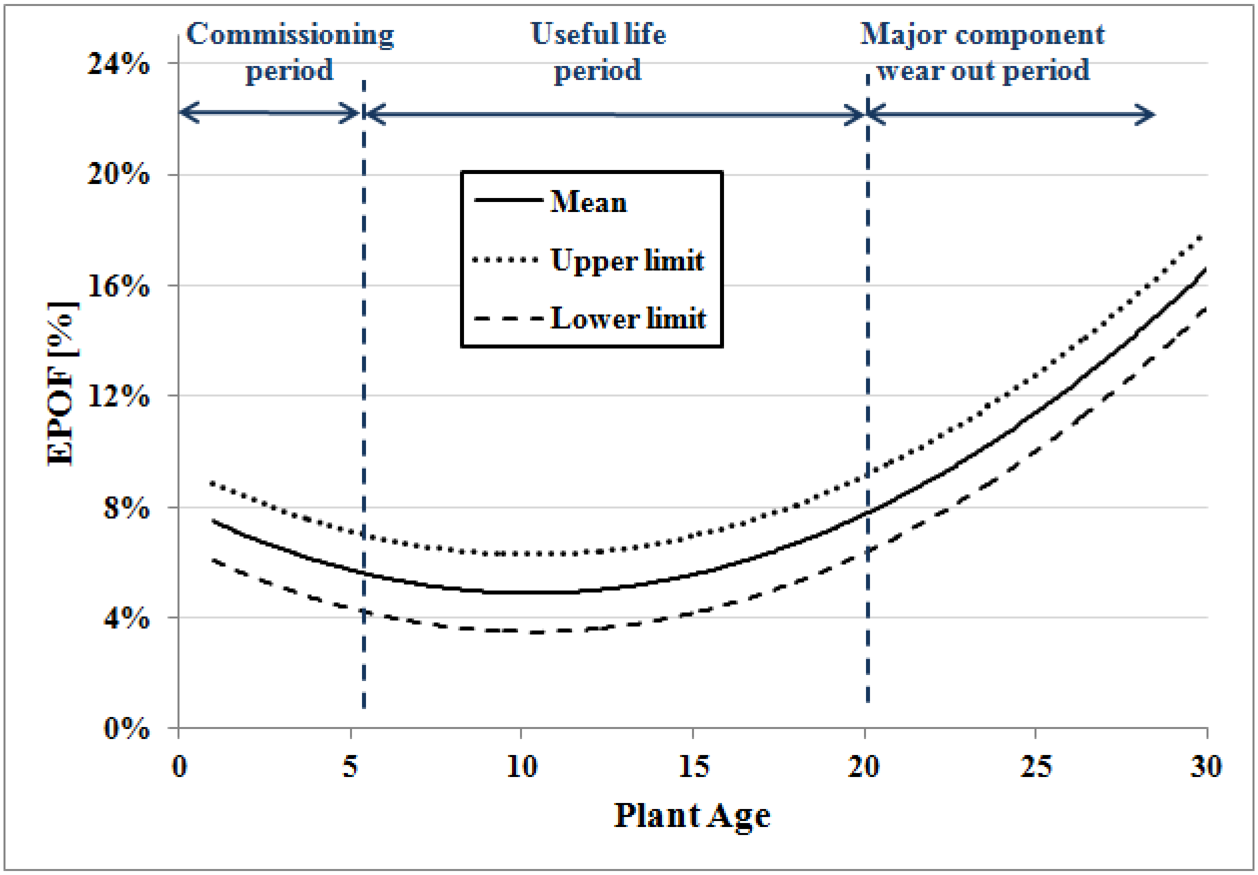


Figure 5-13
Life cycle EPOF v. Age for conventional plants operating in the cycling regime

Figure 5-14 shows comparison of the average EPOF levels between cycling and base load plants. The grey lines (upper, lower and mean value) in this figure represent the planned outage levels for a conventional plant operating in the base load mode. This has been included in the same figure with cycling curve to provide a comparison between the conventional plants operating in the cycling and base load mode. The EPOF polynomial trend line for conventional base load plants shows a weak positive correlation of the data, with a coefficient of determination (R^2) 0.76. According to Figure 5-14, the average EPOF value for conventional plants operating in the cycling regime is ~ 2.0% higher than the plants operating in the base load mode in the first 6 years of operation, and ~ 2.5% higher between 6 to 20 years of operation. EPOF for the cycling plant increases after 20 to 25 years of operation compared to the base load plant which is estimated to be as much as 6%. It is noted that the EPOF level increases at or near the end of life of the plant due to the increase of both frequency (number of outages) and duration (longer hours of outages) of planned outages.

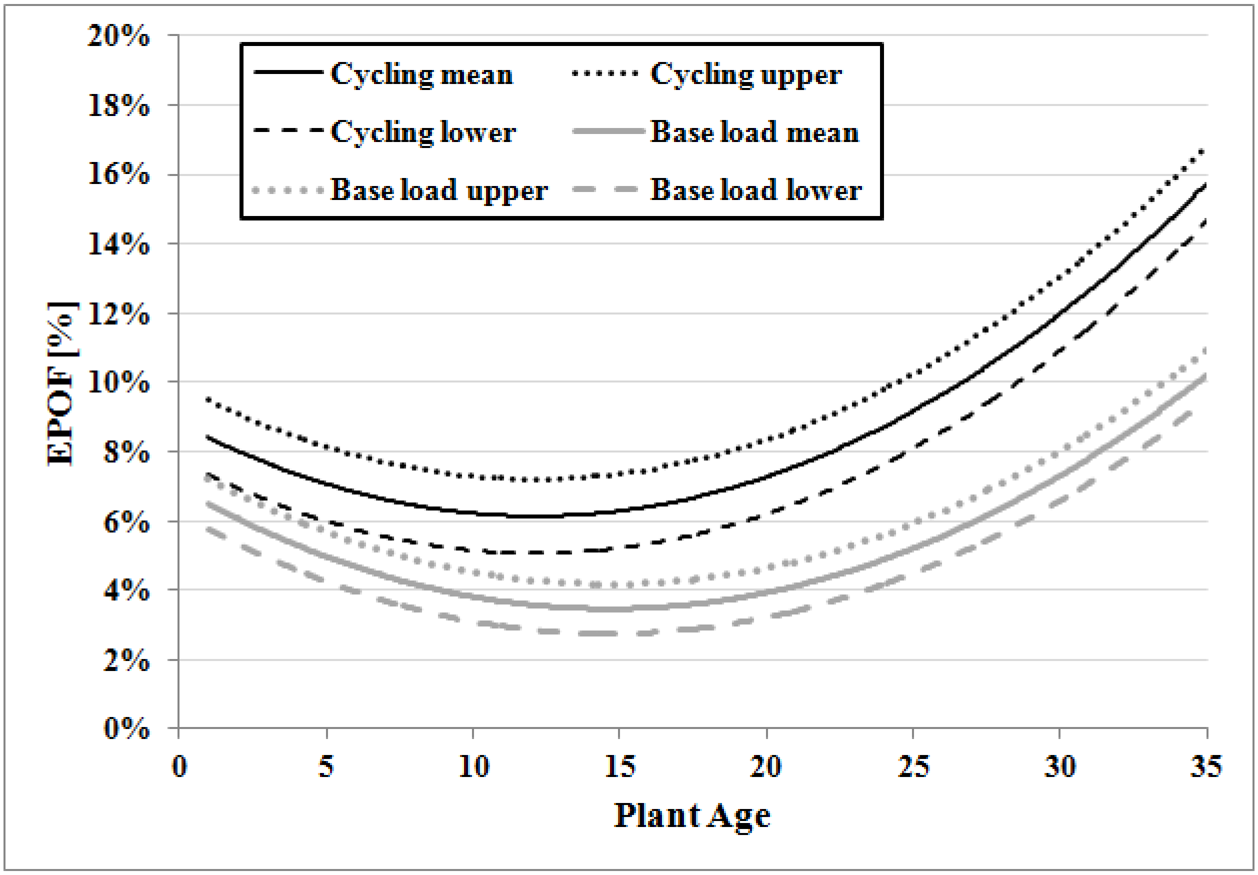


Figure 5-14
Life cycle EPOF v. Age for conventional plants operating in the base load mode and cycling regime

Figure 5-15 represents the average levels of planned outage for the entire life cycle of a ***CCGT plant*** operating in the cycling regime (considering all types of cycling regimes). The EPOF polynomial trend line for CCGT cycling plants shows a strong positive correlation of the data, with a coefficient of determination (R^2) 0.89. According to this figure, the planned outage levels for cycling CCGT plants are within ~ 6 to 9% during the first 6 years of operation and within ~ 4 to 6% for the next 14 years of operation and the EPOF achieves its minimum level between 10 to 14 years. During the 'major component wear out period' which is near the end of life (assuming major components at or near end of life have not been replaced), EPOF value for cycling CCGT plants increases to ~ 15-18%.

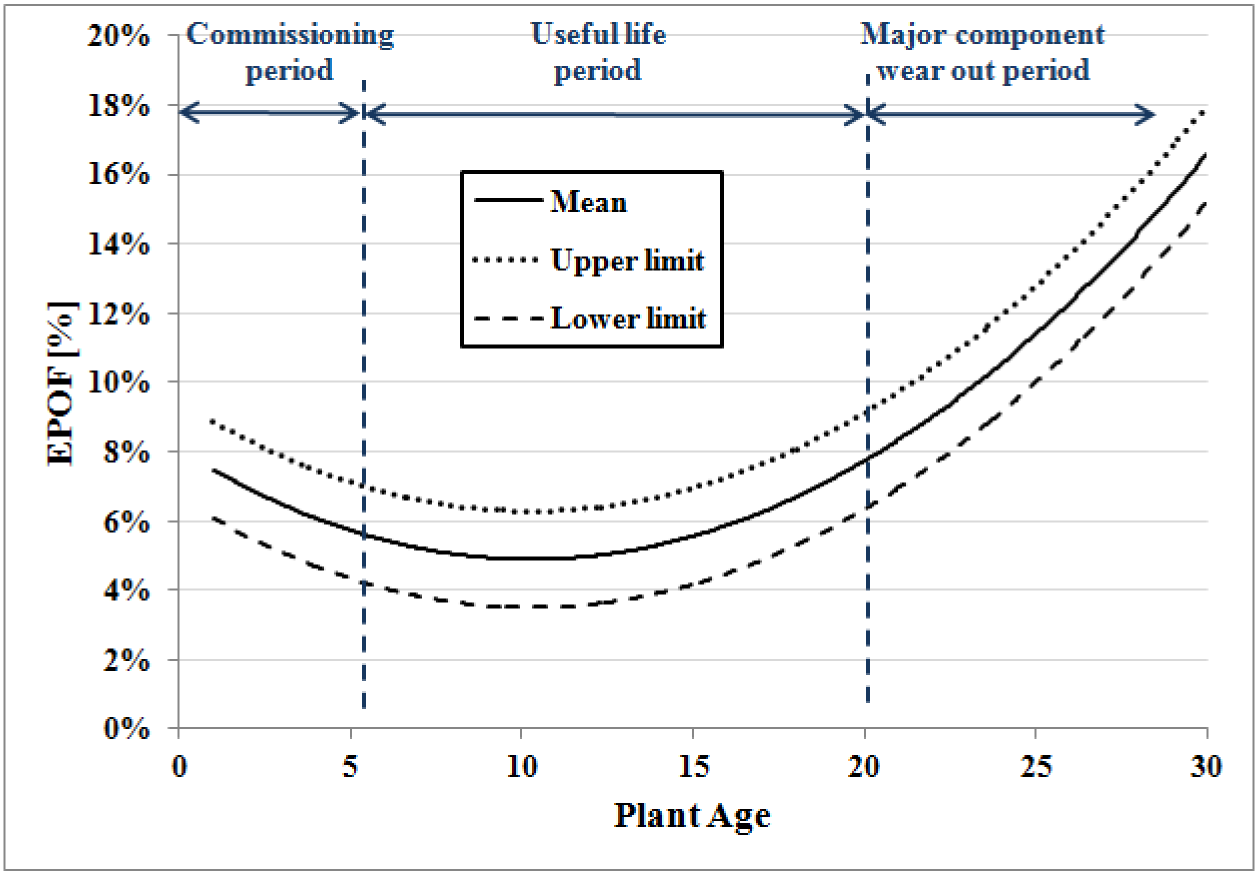


Figure 5-15
Life cycle EPOF v. Age for CCGT plants operating in the cycling regime

Figure 5-16 shows comparison of the average EPOF levels between cycling and base load plants. The grey lines (upper, lower and mean value) in this figure represent the planned outage levels for a CCGT plant operating in the base load mode. This has been included in the same figure with cycling curve to provide a comparison between cycling and base load plants planned outages which provide a clear understanding of the availability impact on the plant due to cycling. The EPOF trend line for CCGT base load plants shows a strong positive correlation of the data, with a coefficient of determination (R^2) 0.92. According to Figure 5-16, the average EPOF value for CCGT plants operating in the cycling regime is ~ 2.5% higher than the plants operating in the base load mode in the first 6 years of operation and ~ 3.5% higher between 6 to 20 years of operation. EPOF for the cycling plants increases much more abruptly after 20 to 30 years of operation compared to the base load plants which is estimated to be as much as 7%.

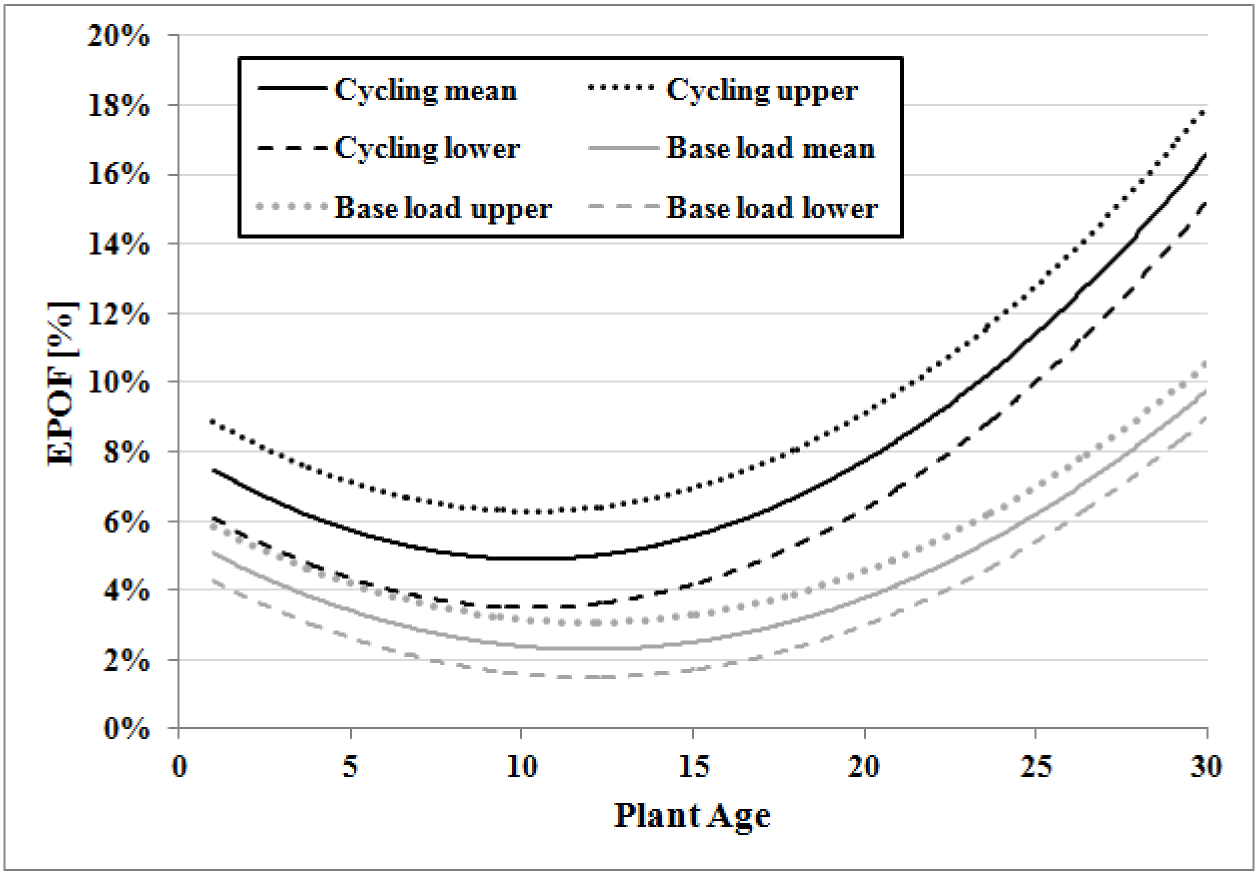


Figure 5-16
Life cycle EPOF v Age for CCGT plants operating in the base load mode and cycling regime

It is observed from the above analysis that planned outages also have a great impact on the performance of cycling plants. It is possible to reduce downtime due to planned outages by following an effective maintenance procedure. The improvement of maintainability and maintenance effectiveness for cycling conventional and CCGT plants by implementing a Reliability Centered Maintenance and Condition Based Maintenance to reduce downtime due to planned maintenance is discussed in detail in *Section 8*.

5.4 Equivalent Planned Outage Factor for Various Cycling Regimes

This Subsection describes the average EPOF levels for different types of cycling regimes (e.g. sporadic, weekend shutdown, two shifting, double two shifting etc.) for the entire life cycle of a plant. The regions for cycling regime, can easily be identified from the equivalent hot start (EHS) performed per year by the cycling plant, showed in Subsection 5.2 for EFOF.

Annual EHS have been calculated for all units in the database to determine the relationship of average Equivalent Planned Outage Factor (EPOF) levels for various types of cycling regimes. Figure 5-17 shows the average EPOF levels versus annual EHS for a cycling (considering all types of cycling modes) ***conventional plant*** and the polynomial curve obtained from the statistical analysis shows a weak correlation, with a coefficient of determination (R^2) in this case of 0.51. According to this figure, a conventional plant operating in the 'sporadic cycling regime' shows an annual average EPOF value of ~ 10% to 12%, if the plant operates in the 'weekend shutdown regime' the average value for EPOF is ~ 13% to 15%. It has already been stated that no data were available for the plants operating under two shifting and double two shifting cycling regimes. Average EPOF versus lifetime EHS was also plotted to present a better view to understand the impact of cycling or number starts on the EPOF performance, with a coefficient of determination (R^2) 0.66, shown in Figure 5-18. The figure clearly demonstrates that the planned maintenance rate increases with the increase in start-up number, also shown in Figure 5-19.

Figure 5-19 represents the average EPOF levels versus lifetime number of starts which was calculated for all conventional units in terms of hot, warm and cold starts. The polynomial curve for all three types of starts shows a moderately strong positive correlation, with a coefficient of determination (R^2) 0.66. Overall, all three trend lines demonstrated the fact that the average EPOF levels increase with increase in the number of start-ups. The figure also indicated that the cold start is more damaging.

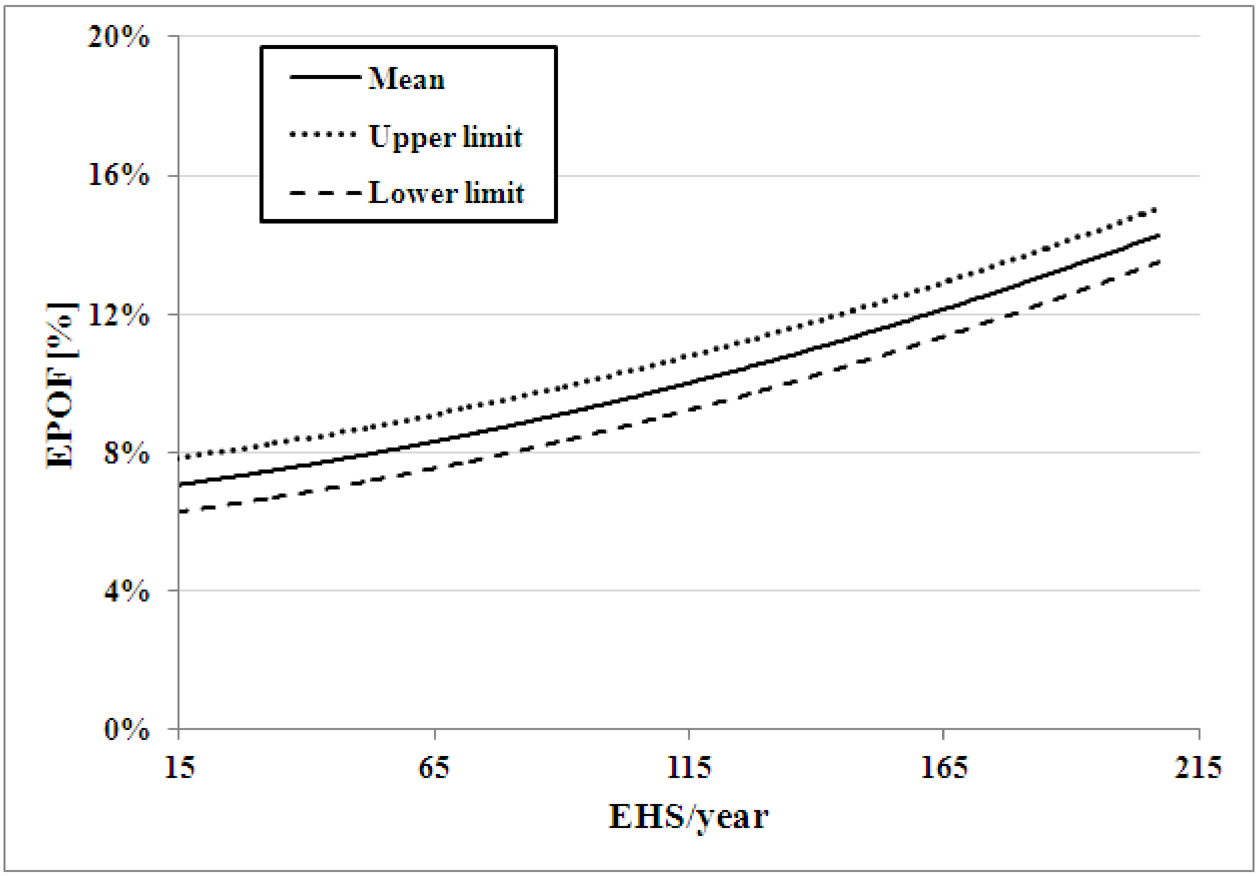


Figure 5-17
Annual average Equivalent planned outage factor (EPOF) v. Equivalent hot starts (EHS) for conventional plants operating in the cycling regimes

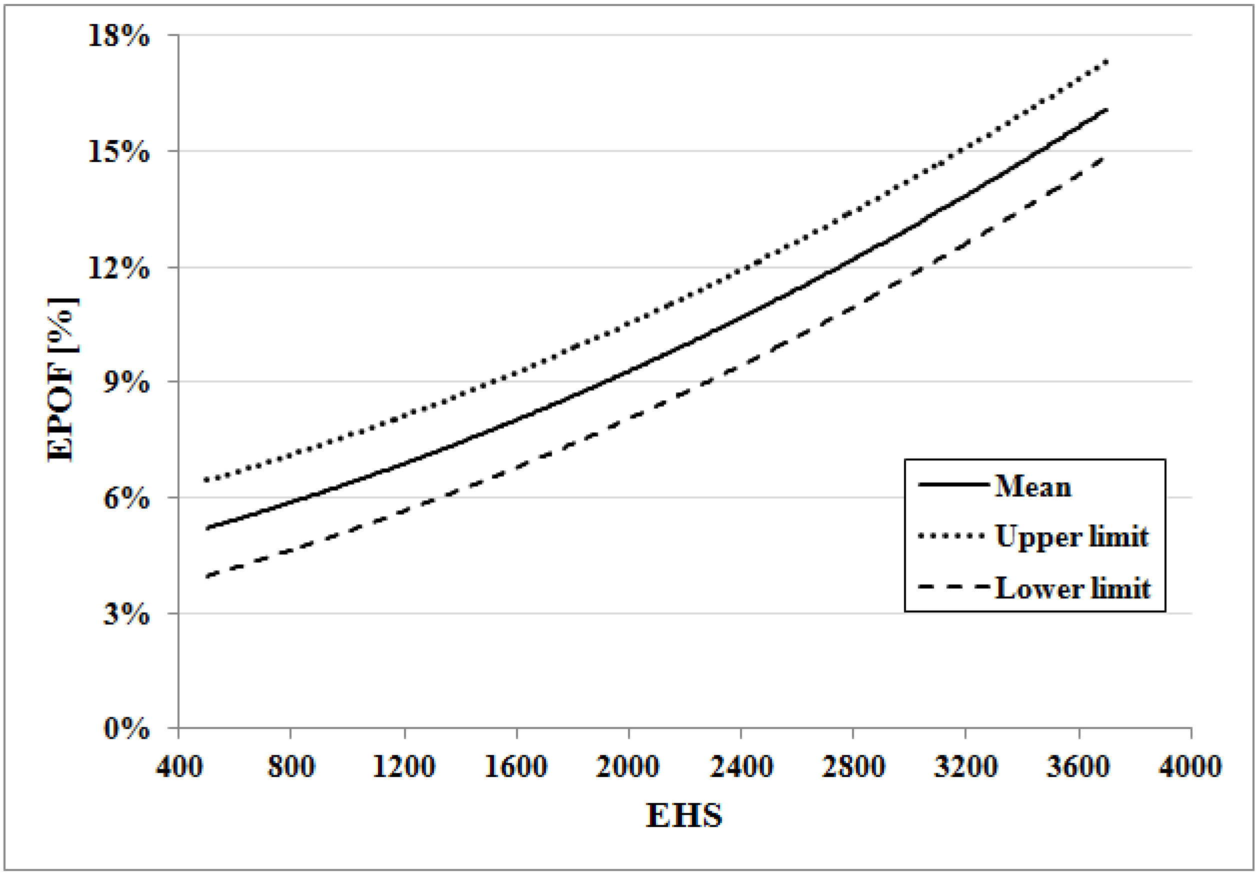


Figure 5-18
Average EPOF v. lifetime EHS for conventional plants operating in the cycling regimes

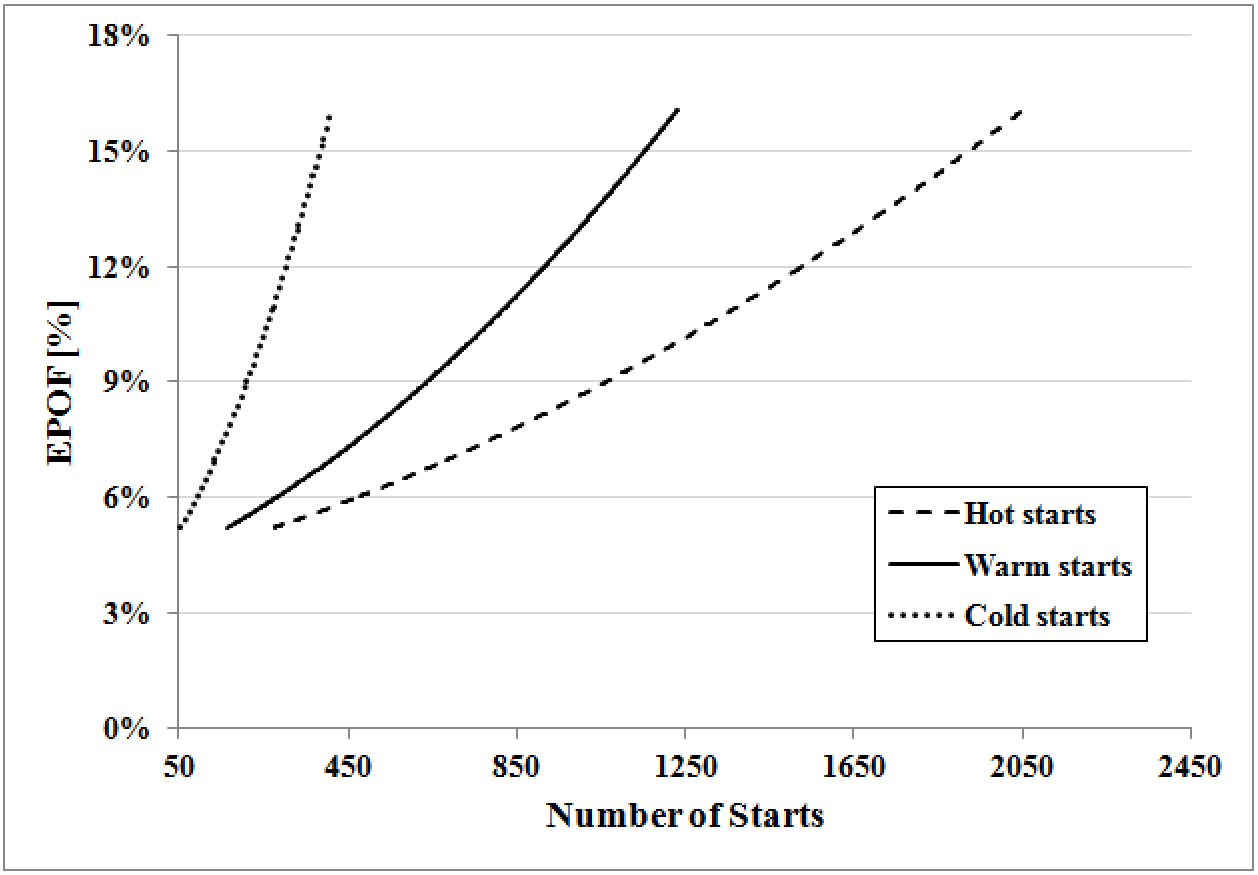


Figure 5-19
Average EPOF v. lifetime hot, warm and cold starts for conventional plants operating in the cycling regimes

The data were separated according to different types of cycling regimes and the relationships were plotted for the EPOF levels for conventional plants, shown in Figure 5-20. As mentioned above, no data were available for two shifting and double two shifting cycling regimes for conventional units. The data for sporadic operation covers the age range between 3 to 31 years of operation and shows a weak positive correlation, with a coefficient of determination (R^2) 0.71, whereas the data for weekend shutdown operation covers the age range only from 29 to 35 years of operation and shows a weak correlation, with a coefficient of determination (R^2) 0.45. The data for load following cycling operation covers the age range from 18 to 31 years of operation and shows a weak positive correlation, with a coefficient of determination (R^2) 0.63. The polynomial trend line for sporadic operation shows that the average value for EPOF is ~ 7.0% for the first 6 years, ~ 6.0% for 7 to 20 years and the value increases after 20 years of operation. The polynomial trend line for weekend shutdown operation shows that the average value for EPOF increases at the ‘major component wear out period’ where the data only covers the age range of 29 to 35 years of operation. The polynomial trend line for load following operation represents that the average value for EPOF increases after 20 years of operation. The figure also includes the trend line for base load operation to demonstrate the impact of cycling operation.

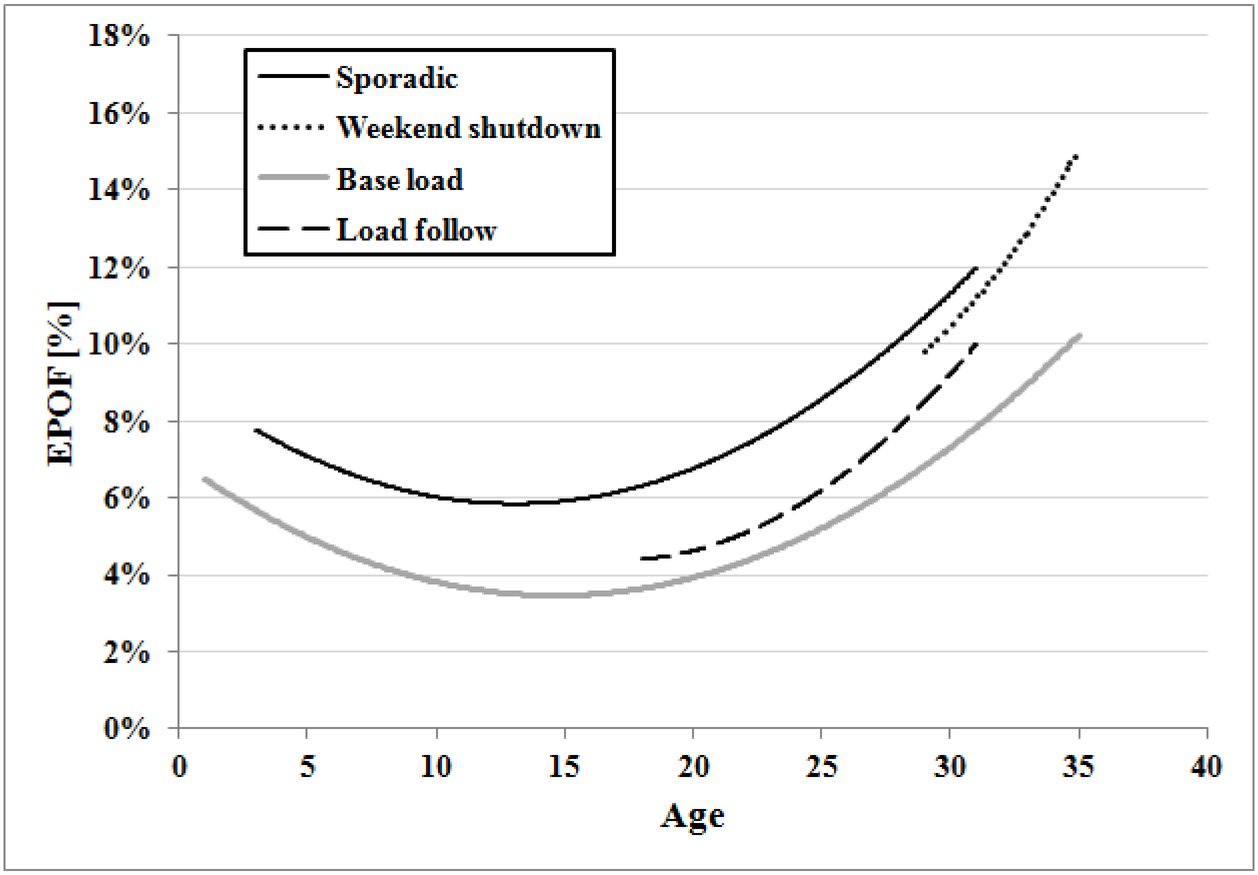


Figure 5-20
Average EPOF v. Age for conventional plants operating in the various cycling regimes

Figure 5-21 shows the average EPOF levels versus annual EHS for a cycling (considering all types of cycling modes) ***CCGT plant*** and the polynomial curve obtained from the statistical analysis shows a weak positive correlation, with a coefficient of determination (R^2) in this case of 0.75. According to this figure, a CCGT plant operating in the 'sporadic cycling regime' shows an average EPOF value of ~ 2.5% to 4.0%, if the plant operates in the 'weekend shutdown regime' the average value for EPOF is ~ 3.0% to 4.5%, if the plant operates in the 'two shifting regime' the average value for EPOF is ~ 4.0% to 6.5% and if the plant operates in the 'double two shifting regime' the average value for EPOF increases to ~ 6.5% and above. Figure 5-21 also shows that the more damaging type of cycling is the 'double two-shifting'. Average EPOF versus lifetime EHS was also plotted to present a better view to understand the impact of cycling or number starts on the EPOF performance, with a coefficient of determination (R^2) 0.76, shown in Figure 5-22. The figure clearly demonstrates that the planned maintenance increases with the increase in start-up number, also shown in Figure 5-23.

Figure 5-23 represents the average EPOF levels versus lifetime number of starts which was calculated for all conventional units in terms of hot, warm and cold starts. The polynomial curve for all three types of starts shows a moderately strong positive correlation, with a coefficient of determination (R^2) 0.76. Overall, all three trend lines demonstrated the fact that the average EFOF levels increase with increase in the number of start-ups. The figure also indicated that the cold start is more damaging.

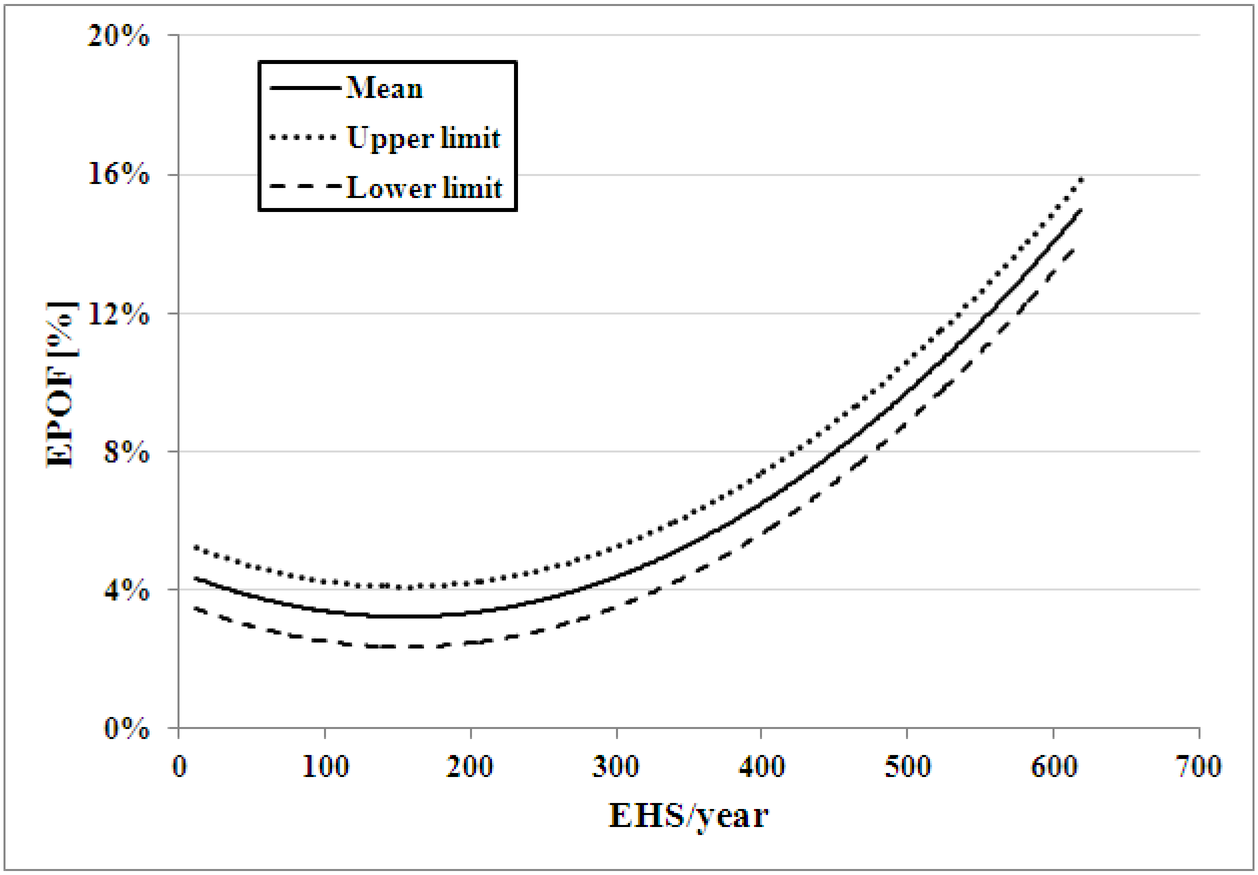


Figure 5-21
Average EPOF v. annual EHS for CCGT plants operating in the cycling regimes

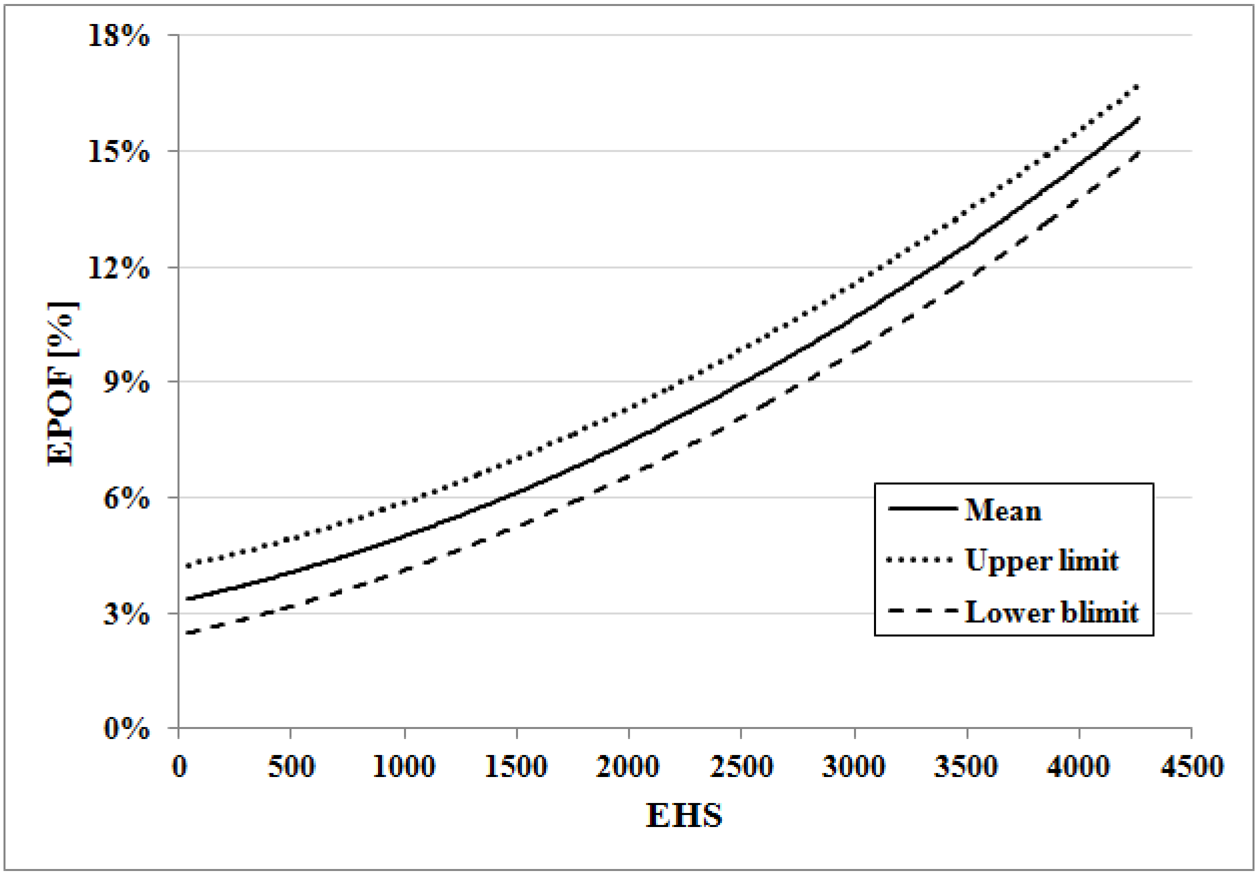


Figure 5-22
Average EPOF v. lifetime EHS for CCGT plants operating in the cycling regimes

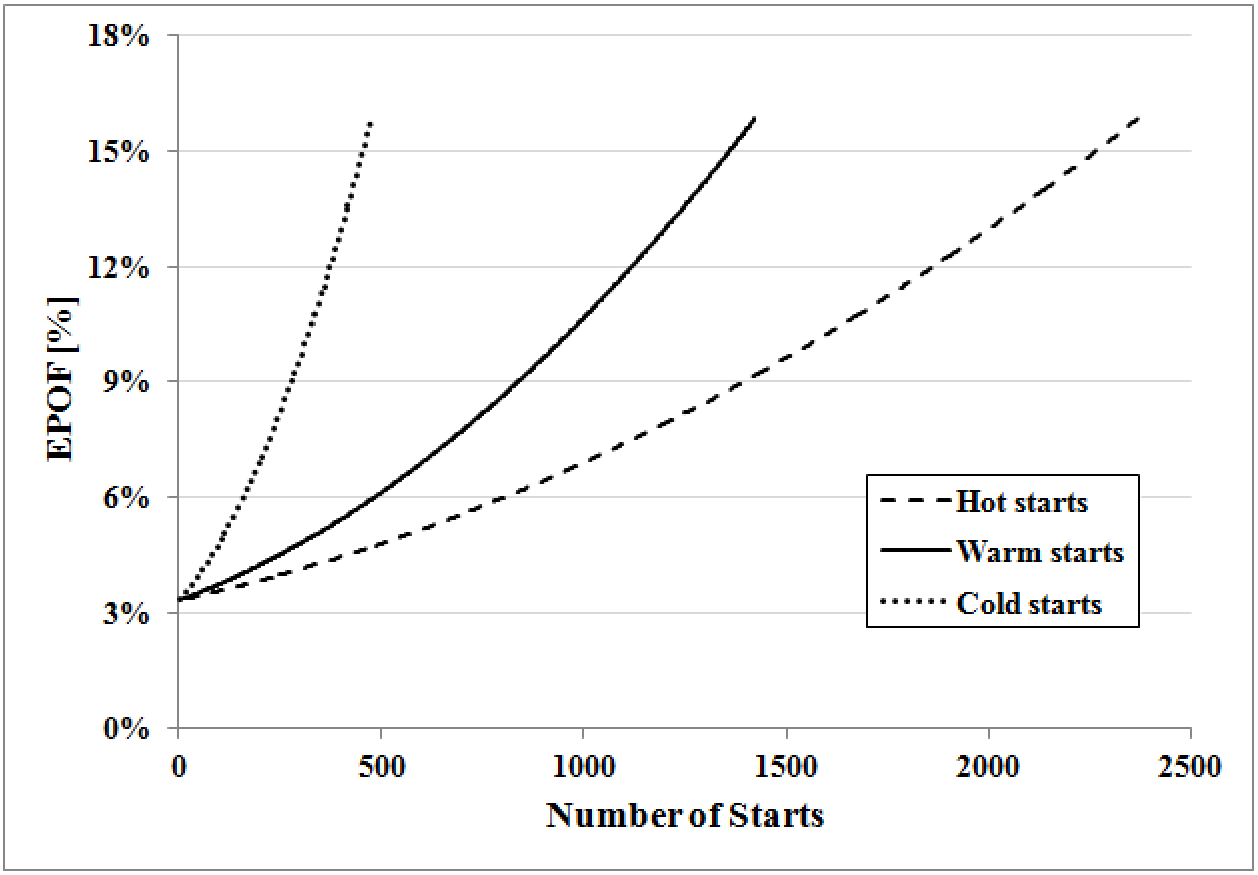


Figure 5-23
Average EPOF v. lifetime hot, warm and cold starts for CCGT plants operating in the cycling regimes

The data were separated for different types of cycling regimes and the relationships were plotted for the EPOF levels for CCGT plants, shown in Figure 5-24. The data for sporadic operation covers the age range between 1 to 25 years of operation and shows a moderately strong positive correlation, with a coefficient of determination (R^2) 0.84. The data for weekend shutdown and double two shifting operation covers the age range only from 1 to 15 years of operation and shows a weak positive correlation, with a coefficient of determination (R^2) 0.79 and 0.75, respectively. The data for two shifting operation covers the age range from 1 to 7 years and 23 to 30 years of operation and shows a moderately strong positive correlation, with a coefficient of determination (R^2) 0.86. The data for load following cycling operation covers the age range from 1 to 17 years of operation and shows a weak positive correlation, with a coefficient of determination (R^2) 0.62. It is observed from the figure that for all types of cycling regimes the general trend for the EPOF level is that it is slightly higher during the' commissioning period', then gradually reduces to a certain level during the 'useful life period' and again starts to increase after 20 years of operation during the 'major component wear out period'.

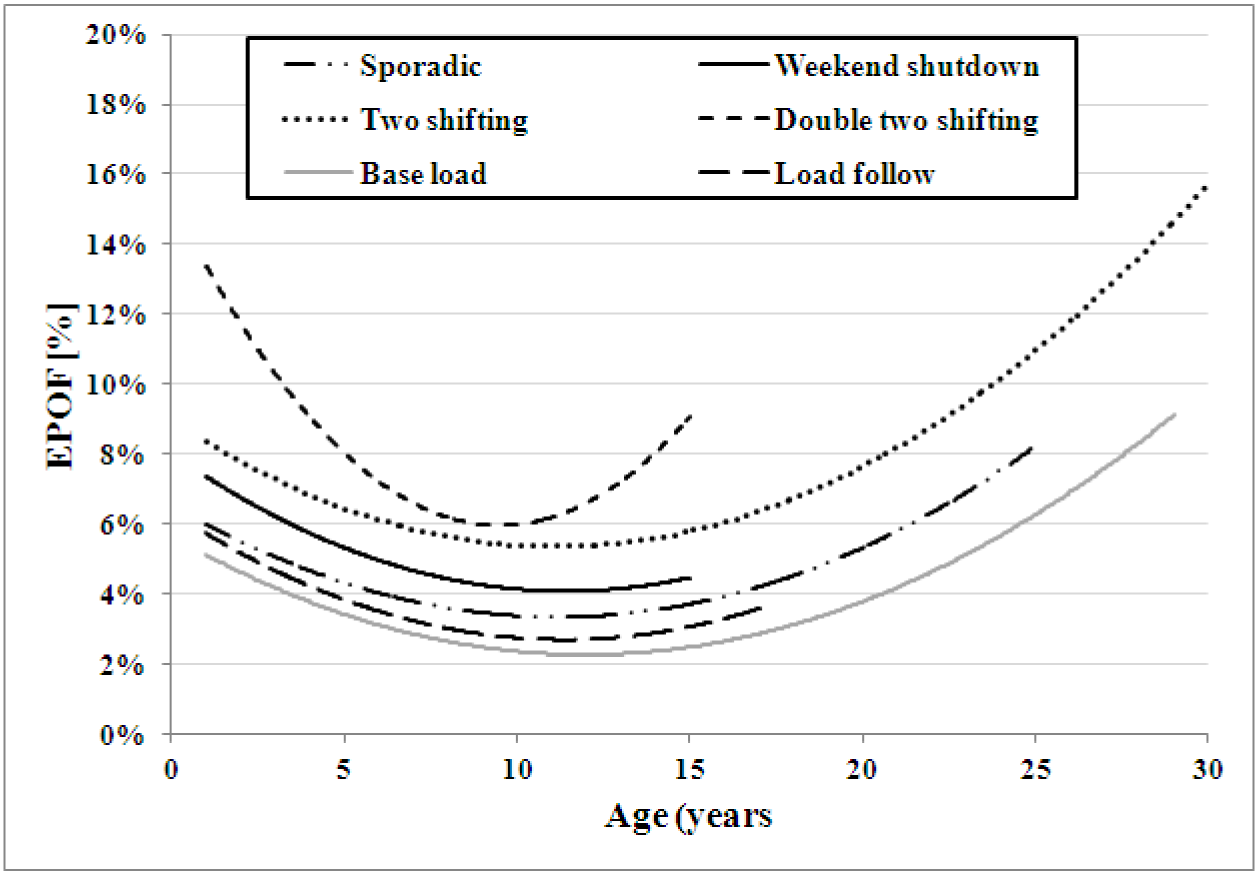


Figure 5-24
Equivalent planned outage factor (EPOF) for CCGT plants operating in the various cycling regimes

5.5 Availability (A) and Reliability (R)

The *Availability* of a power plant is the percentage of time the plant is available to generate power in any given period.

The *Availability* of a plant can be calculated using the following relationship, which takes into account the stoppages due to both forced and planned outages:

$$A = \frac{(PH - POH - FOH)}{PH} \times 100$$

Where:

POH = Planned outage hours

FOH = Forced outage hours

PH = Period Hours (8760 hrs/year) (as previously defined)

The *Reliability* of a plant is the percentage of time between planned overhauls when it is available for service and is defined as:

$$R = \frac{(PH - FOH)}{PH} \times 100$$

Where:

FOH = Forced outage hours

PH = Period Hours (8760 hours/year) (as previously defined)

Availability and reliability have major impacts on the economics of plant operation. Reliability is essential in the sense that when the power is required the plant must be ready to generate instantly. Planned outages are scheduled for non-peak periods. Peak periods are when the majority of the income is generated, as usually there are various tiers of pricing depending on the demand. Many power purchase agreements have clauses, which contain capacity payments, thus making plant availability critical in the economics of the plant.

The Reliability of a plant depends on many parameters, such as the type of fuel, the preventive maintenance programs, the operating mode, the control systems, and the firing temperatures. But it is directly related to the forced outage hours. Therefore, by reducing the plant forced outage hours, it is possible to improve its condition and hence reliability.

Figures 5-25 and 5-26 provide the life cycle evolution of Availability and Reliability for conventional and CCGT plants. These figures show that the availability and reliability have lower values during the earlier plant life and a steady increase from age 6 until age 20 years, achieving the maximum value possible between ~ 14 to 18 years of operating life. The availability and reliability decrease abruptly during the later part (after 20 years of operation) of the plant life as the components of a cycling plant show increased downtime due to increased failure rate. Therefore, the ‘initial’ and ‘end of life’ periods are the most critical periods as the initial reliability being the driving performance factor for plant operation. A low initial (also known as ‘inherent’) reliability is translated as a ‘low reliability’ towards the entire plant life cycle, therefore, it is crucial to achieve the highest inherent Reliability value possible during the plant initial period of operation. The ‘end of life’ period is described by a decrease in Reliability and Availability. To avoid this problem cycling plants need to improve equipment maintainability, maintenance effectiveness and implement a maintenance program direct to the most critical components. A solution would be the implementation of a condition monitoring system and a reliability centered maintenance program/ team to monitor continually the maintenance effectiveness and generate solutions to improve equipment condition and reduce downtime.

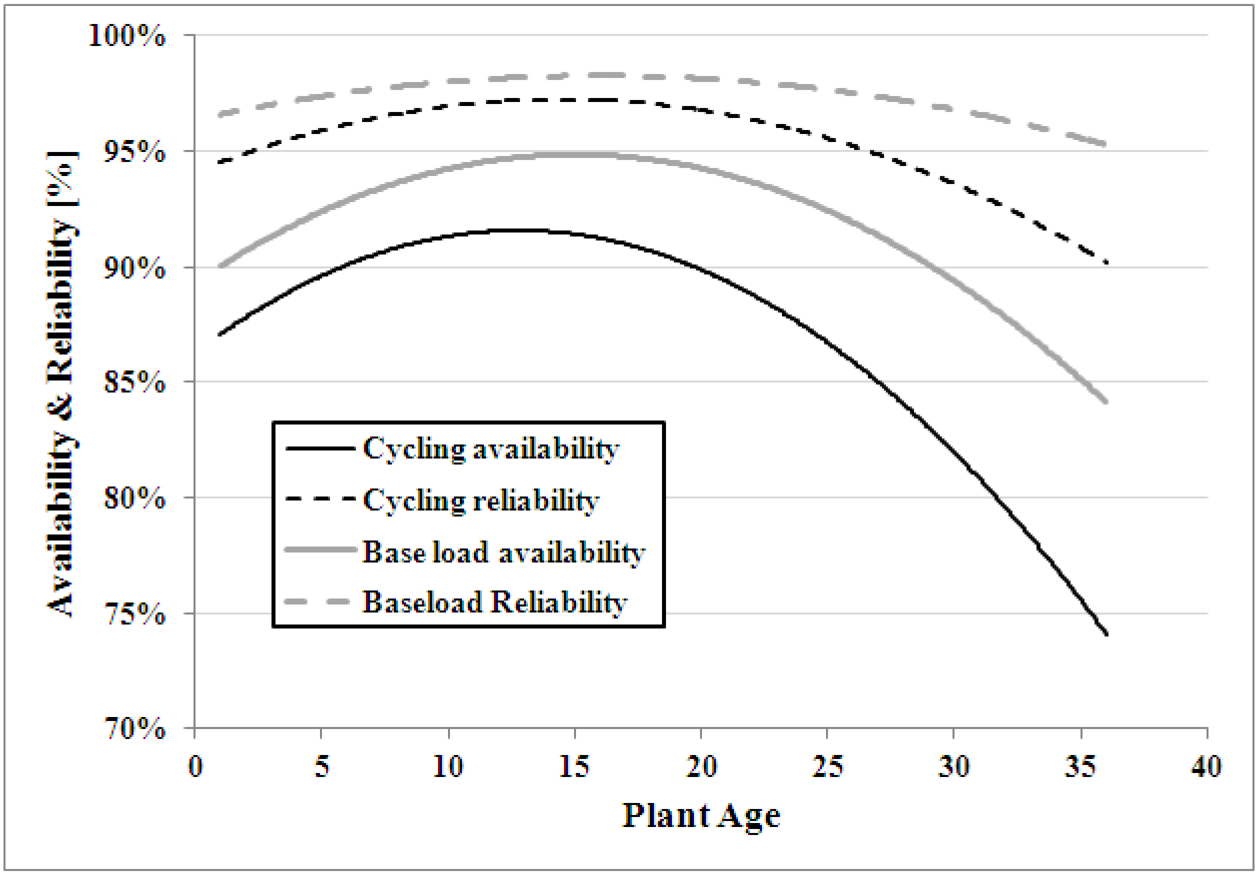


Figure 5-25
Life cycle Availability and Reliability for conventional plants operating in the base load and cycling regime

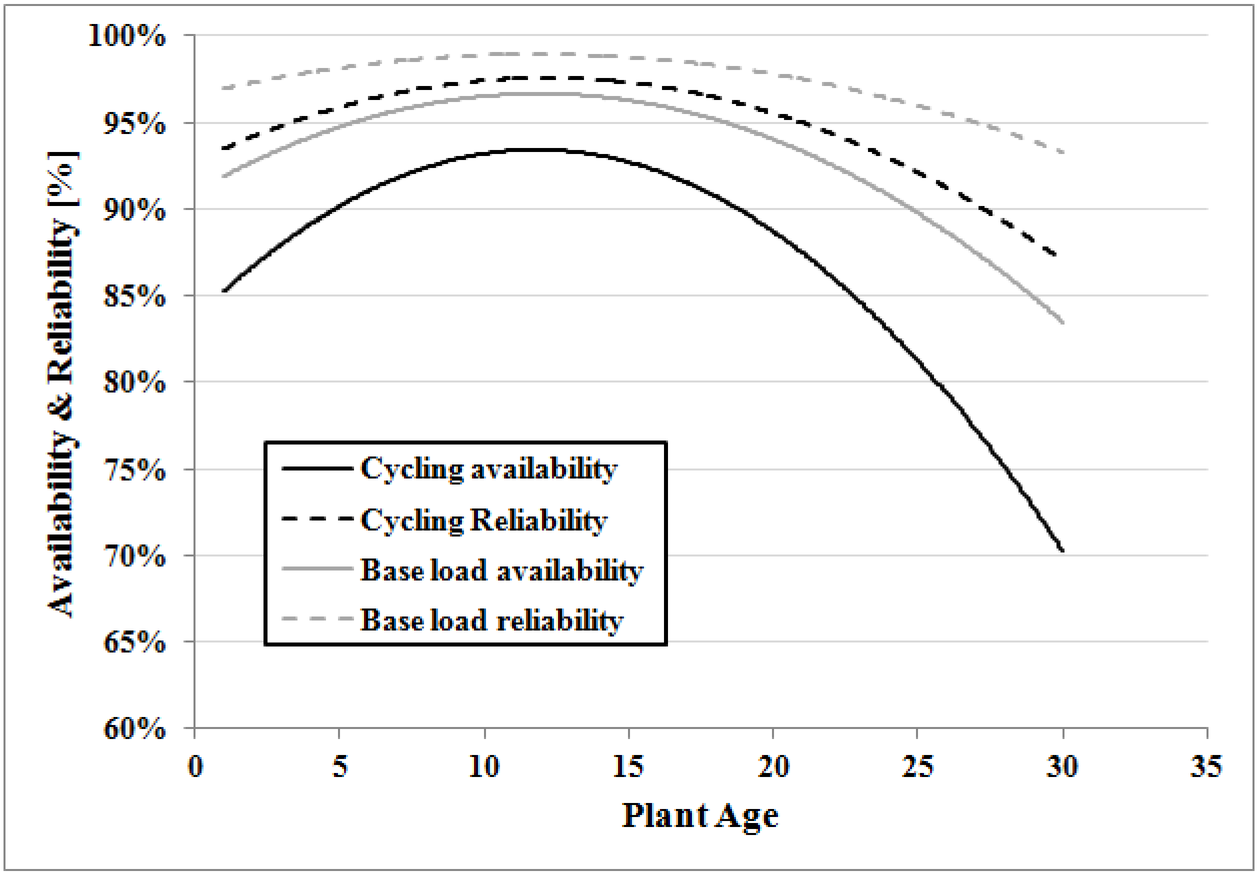


Figure 5-26
Life cycle Availability and Reliability for CCGT plants operating in the base load and cycling regime

Low reliability of plants gives rise to maintenance costs. In many large plants, about one-third of the failures are due to equipment failure; it is therefore necessary to redesign parts of equipment to improve reliability. Another crucial point already mentioned above is to guarantee a very high Reliability value from the first commissioning i.e. the inherent Reliability needs to be as high as possible to avoid redesign during the 'useful life period' of the plant.

One important phenomenon detected during the cycling failure analysis is the large amount of forced outages or trips that occurred during a plant start-up. The analysis demonstrated that ~ 20% of the plants had frequent start-up trips. This reflects that a reliability analysis should account separately for the failures occurred during start-up since this will be a limitation for the plants which should aim to provide reliable power during operation in the cycling regime. Based on this, it is clear that the Start-up Reliability needs to be clearly understood. It can be determined by the following relationship:

$$SR = \frac{NSS}{(NSS + NUS)} \times 100$$

Where:

SR = Start-up Reliability [%]

NSS = Number of Successful Starts

NUS = Number of Un-successful Starts

This factor was not considered in this study due to the lack of the availability of such data. However, it would be highly advisable to assess SR during a plant individual assessment to better identify its capability to reliably respond to the market demand.

5.6 Failure or Breakdown Frequency and Typology

Forced outages are without doubt the most important factor in a plant life cycle since their effect on a considerable percentage of the expected operating hours may have an impact on the plant Reliability and Availability. Loss of production and decrease of Availability & Reliability contributes to the increase of plant operating costs. The majority of losses have been unfortunately very difficult to recover since the electricity market became competitive. In order to avoid economic impact due to cycling, it is crucial to create a maintenance plan with a well-balanced proactive maintenance, predictive maintenance and preventive maintenance through the plant life cycle. This Subsection briefly discusses the most common component failures in conventional and CCGT plant. A detailed description of failure/ damages due to cycling in conventional and CCGT plants for each type of major components are described in *Appendices B* and *C*, respectively.

5.6.1 Failure Frequency

Component based failure frequency is determined from the plant experience and performance database and is shown in Figures 5-27 and 5-28 for conventional and CCGT plants. From Figure 5-27, it is observed that the number of failures is generally higher in superheater tubes in boiler units for conventional plants. Figure 5-28 shows that the number of failures is generally higher in GT units than in HRSG for CCGT plants.

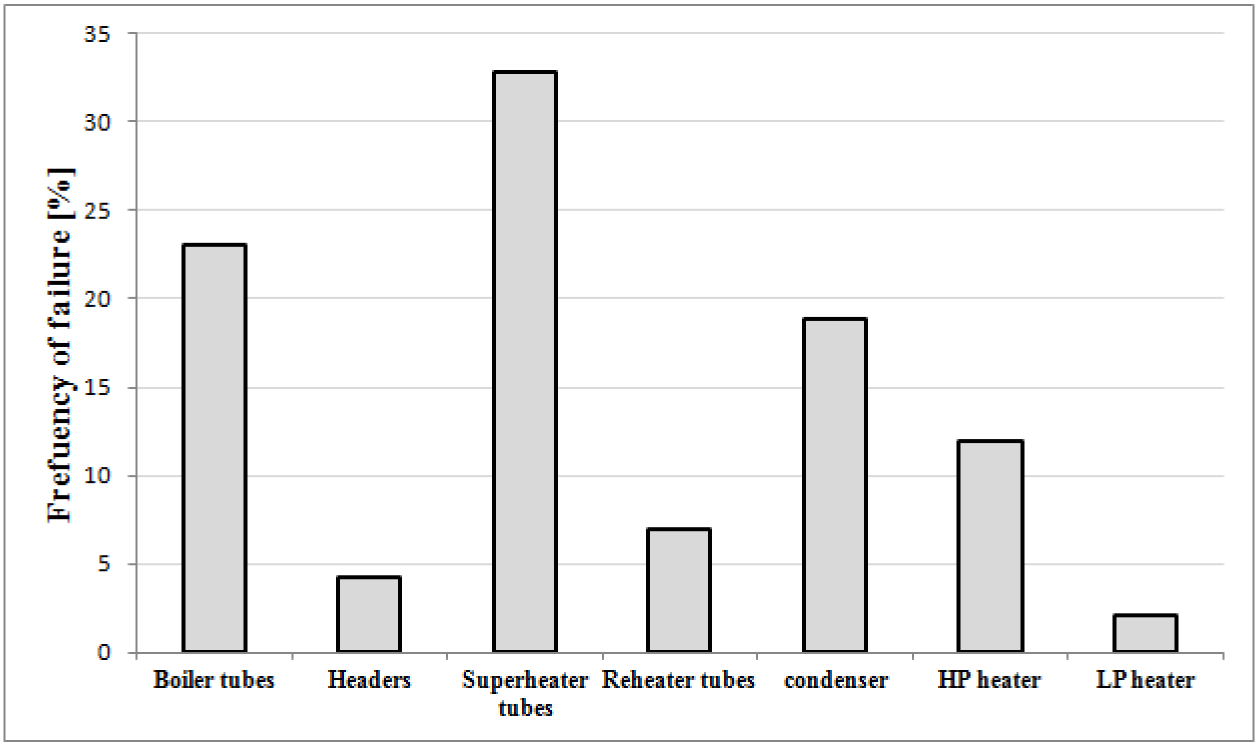


Figure 5-27
Component based failure frequency for conventional plants operating in the cycling regime

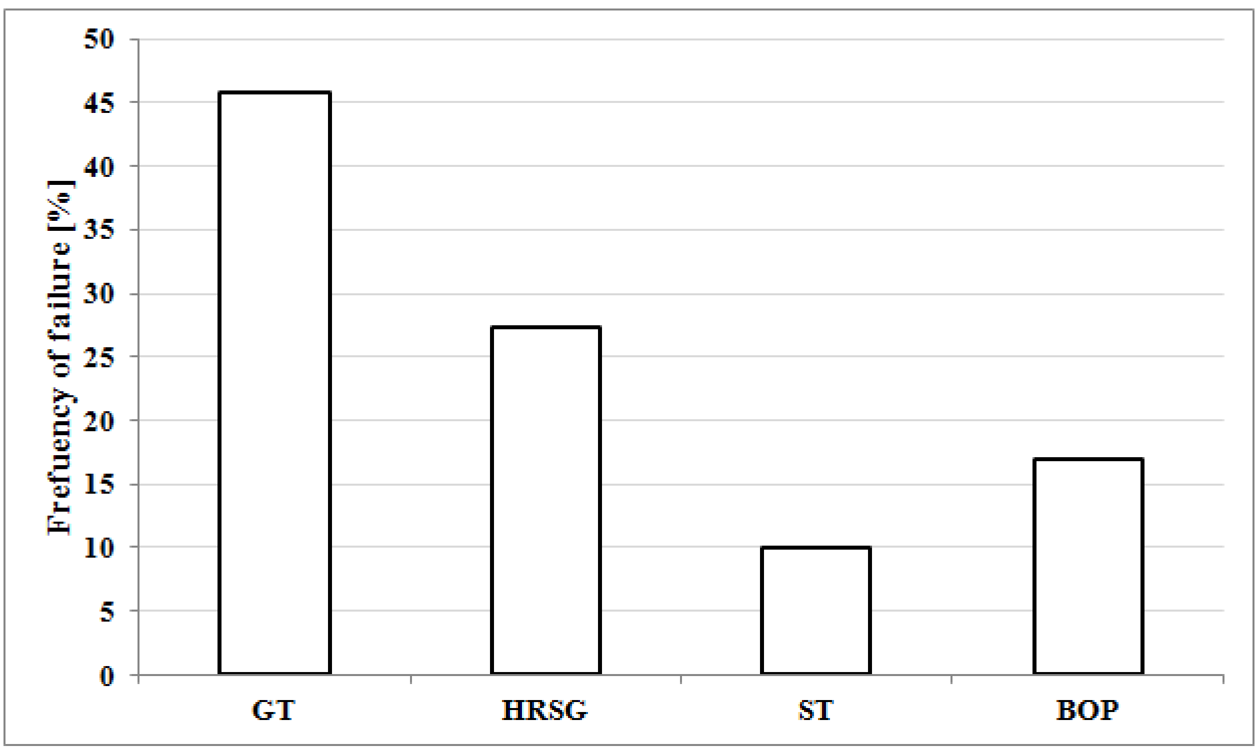


Figure 5-28
Component based failure frequency for CCGT plants operating in the cycling regime

The bar chart presented in Figure 5-29 shows a comparison between conventional and CCGT plants which shows the most frequent damage mechanism affecting plants operating in the cycling regime. According to this figure, fatigue is the most damaging type of failure followed by wear and erosion in CCGT plants and corrosion-fatigue in conventional plants.

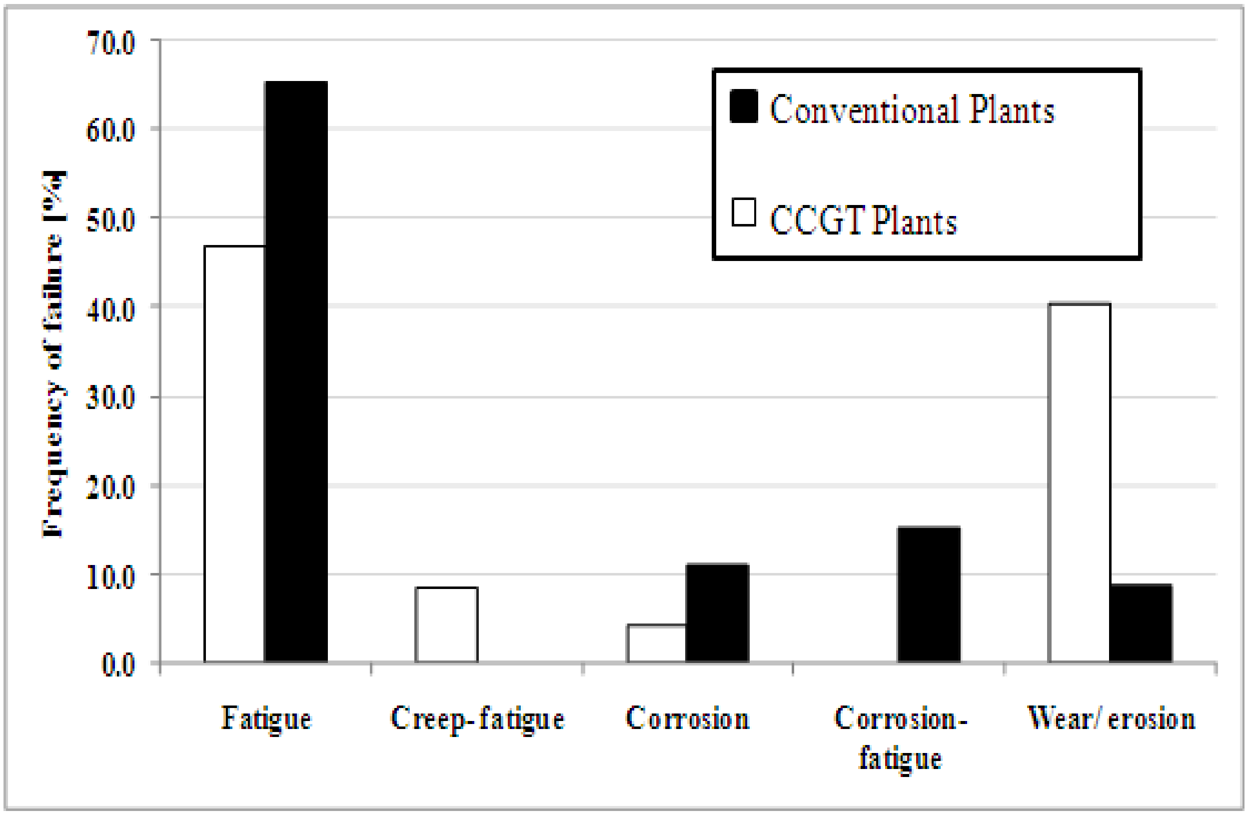


Figure 5-29
Failure frequency comparison between conventional and CCGT plants operating in cycling regime

5.6.2 GT Failure Typology

Figure 5-30 shows the higher occurrence failure mechanisms found in gas turbines. A more recent review of detailed historical data/documents from several benchmarked GT frames outlines some areas at risk in GT systems and their typical frequency of failures (Table 5-2).

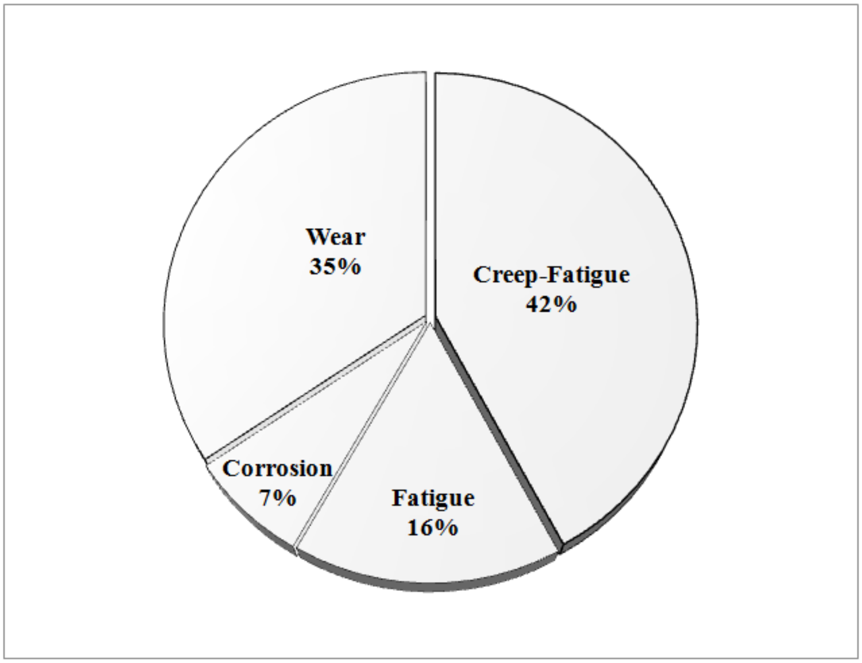


Figure 5-30
Failure mechanisms benchmarked in GT units

Table 5-4
Frequency of failures for GT components

GT Critical Parts	Frequency of failures
Rotating blades of the hot gas path section*	42%
Stationary parts (vanes & shroud) of the hot gas path section	21%
Bearings of the GT	14%
Compressor Rotating blades of the GT	12%
Exhaust Plenum area of the GT	5.8%
Compressor stationary blades of the GT	5.3%

*The failures of rotating blades were mainly associated with thermal fatigue, creep-fatigue interaction failure, wear (of coating), and oxidation.

5.6.3 HRSG Failure Typology

To be able to define the problem areas in an HRSG, one must define various components contributing to the overall decrease in plant Availability and Reliability.

From the benchmarked plant data it can be stated that the HRSG is usually not a major problem in the overhaul downtime of a CCGT plant. The problems in the HRSG are mainly located in the high temperature modules such as High Pressure Superheater (HPSH) and High Pressure Evaporator (HPEvap). More recently Low Pressure Economizer tubes have also been experiencing an increase in frequency of failure mainly due to cycling operation. HRSG units that were initially designed to operate in base-load regime are experiencing a change of operation regime. This change of operation is due to the market constrains where many base-load HRSGs are operating in load following or two-shifting regime in order to remain competitive when compared with modern and more efficient combined cycle plants.

A more recent review of detailed historical data/documents from several benchmarked HRSG units outlines some areas at risk in HRSG system (Figure 5-31) and their typical frequency of failure mechanisms (Figure 5-32).

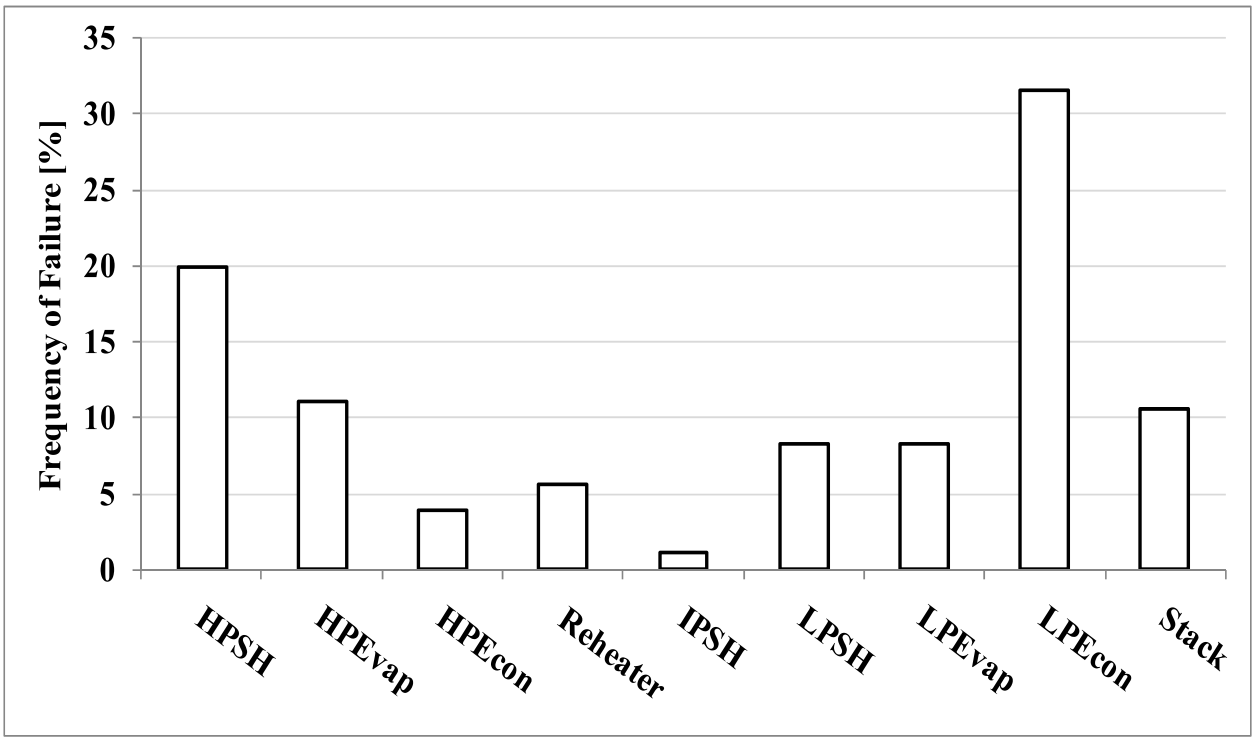


Figure 5-31
Frequency of failures for HRSG components

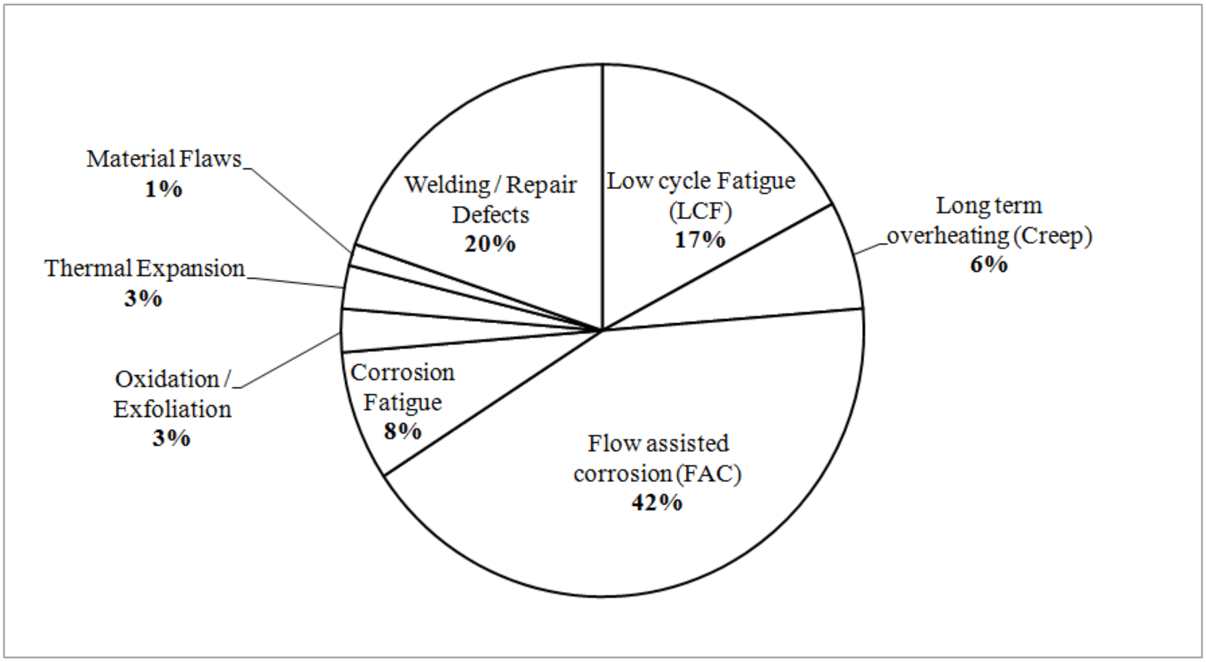


Figure 5-32
Frequency of failure mechanisms benchmarked in HRSG units

5.6.4 BoP Failure Typology

Balance of Plant (BoP) is composed basically of the auxiliary system such as cooling system, feed water pumps, air separation unit, condenser etc. Normally the components that are part of the auxiliary system are not considered major components/ assets such as HRSGs, GTs or generators but it is still important to keep in mind that auxiliary components play an important role in a plant. The availability and reliability of GT and HRSG depends on the availability and reliability of these auxiliary components. They link GTs to HRSGs, GTs/ HRSGs to STs, STs to condensers, condensers to cooling towers etc. Some of these "small" components are critical for the plant operation. A failure in one of these items may lead to a catastrophic shutdown with major damage and cost consequences. As a general conclusion a power plant cannot operate if the auxiliary system is not working properly.

The failure diagram shown in Figure 5-33 shows the areas where failures occur and the frequency of occurrence in BoP units. The results show the problems related with power supply system, fuel supply equipment, protection system etc, the areas where more frequent failures occurred that resulted in force outages. Protection systems such as fire, lightning and breaker protection are the most frequent types of failure in CCGT plants.

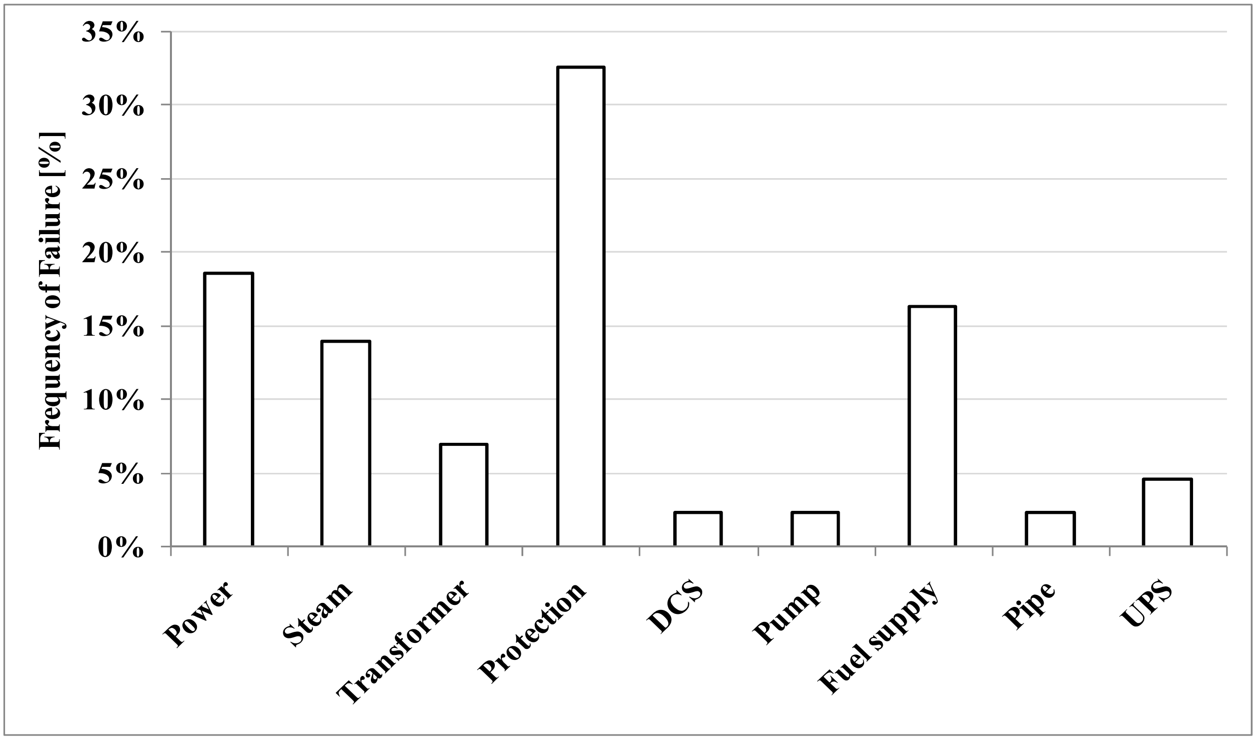


Figure 5-33
BoP frequency of failures for cycling CCGT plants

6
COST ASSESSMENT

The cost analysis undertaken for this Section of the report is based on a top-down statistical analysis. The cost analysis takes into account reported expenditure for each unit over a number of years to determine an annualized non-fuel O&M cost. Cost data are combined with each unit's historic operational data (online hours and unit starts) to develop a top-down statistical model of the O&M costs for a typical plant over its service life.

There are a number of potential limitations to the model. For example, differences in cost allocation between respondents, the specificity of failures to a particular generator type, changes in ownership and/or accounting practices (possibly more than once) during a unit's lifespan and possible accounting anomalies due to equipment sharing between units at the same power station may all have reduced the overall consistency of the cost data. The lifetime costing results produced by the model and described in the following Sections should therefore be viewed as indicative, rather than definitive. Despite potential disparities however, the analysis provides a simple and robust model which can be used to estimate unit lifetime maintenance costs in a range of plant-cycling scenarios.

6.1 Cost Assessment for CCGT Plants

The CCGT plants/units included in this analysis range in capacity from 110 MW to 492. They include units made by Siemens, GE, Alstom, Ansaldo and other major OEMs, ranging in age from less than one year old to over 20 years old. ***It should be noted that the operating history of these units shows no clear dateline between cycling and base load operations.*** Previous reports have in many cases examined units which had undergone a clear transition between modes of operation. The data for these CCGT units do not reflect that situation. There is a continuous distribution of life-consumption rates, from units that have had comparatively low numbers of annual starts through their lifetime, to those that have had higher numbers of starts. In some cases units have gone from having relatively low numbers of annual starts to having higher numbers of starts, and then returned to low starts (following changes in the relative cost of fuel, for example). While the overall trend is that units perform more annual starts as they age, the analysis is not based on simply dividing the sample into base load and cycling units; there is no discontinuity in historic modes of operation to support that interpretation of the lifecycle of a 'typical' unit in this sample.

This study seeks to establish the annual non-fuel O&M cost for a model CCGT over its service life. The annualized non-fuel O&M cost is calculated by summing the costs of maintenance and repairs, capital expenditure (for example, in plant modifications to enhance the unit's cycling capability), increased frequency of inspection and other operating costs. ***The reported costs of the units in the database are then corrected for capacity and related to two time-dependent variables; creep life (measured in online hours) and fatigue life (measured in starts)***.

6.1.1 Correction for Capacity

The CCGT units in the database range between 110 MW and 492 MW. Plant costs are broadly related to capacity; however, they are not simply correlated by a straightforward per-MW relationship. While some cost elements are capacity-dependent (for example, a superheater for a 400 MW unit will cost significantly more than that for a 200 MW unit), some O&M costs (such as labor for repairs) are incurred irrespective of capacity. After examining the reported costs for database units and established a cost-correction factor, the following formula was derived:

Cost correction factor = 0.002 * Capacity (MW) + 0.3

This relationship is plotted in Figure 6-1.

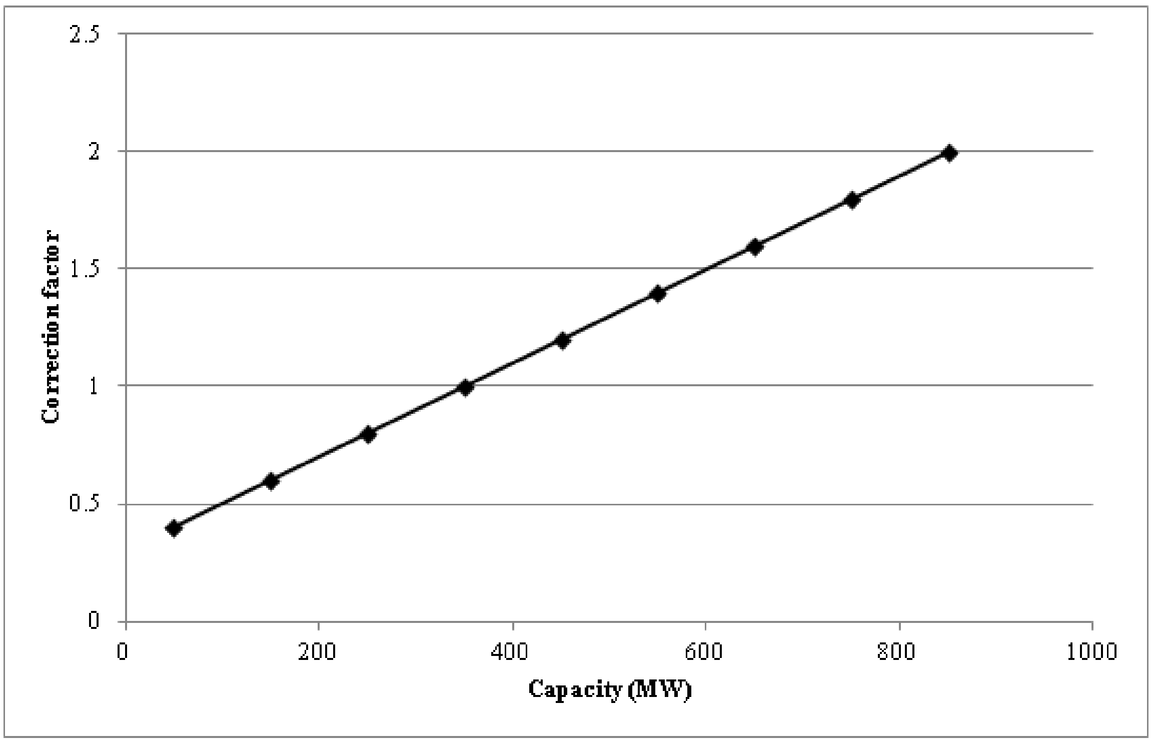


Figure 6-1
Derivation of Unit Capacity Cost Correction factor

The reported, ***annualized O&M costs are divided by the calculated cost correction factor to normalize them around 350 MW***, the mean capacity of the sample (note that this is not the same as calculating a per-MW value and multiplying it by the capacity of the unit in question). It should also be noted that calculated costs do not include estimates for related expenditure, such as long-term increase in heat rate or foregone energy payments resulting from increased outage rates. The impact of cyclic operation on outage rates is examined elsewhere (*Section 5*) in this report.

Raw and corrected cost data were examined in relation to two life consumption metrics; online hours and starts. Most component-level life-consumption studies estimate remaining life by quantifying the creep and fatigue damage that critical plant systems and components have accumulated and relating it to the environmental conditions (temperature, pressure and stress) to which they have been exposed throughout their operating life. The rate of damage accumulation associated with a particular generation profile can then be gauged based on a straightforward ‘cause and effect’ correlation of projected operating conditions and concomitant damage and failure rates. The influence of each mechanism varies in accordance with operational conditions; in situations with low numbers of cycles and long dwell times at temperature, creep is the dominant mechanism, while the reverse is true when the component is subjected to high numbers of cycles and short dwell times. This analysis uses a similar life-estimation process to assess the relative influence of creep life consumption (measured in online hours) and fatigue life (measured in total starts) on damage accumulation (measured in money expended on annual maintenance). The relative influence of each mechanism on the annual maintenance cost is then assessed.

6.1.2 Equivalent Hot Starts

EHS have been calculated for all units in the database to determine the relationship between O&M costs and EHS to understand the cycling impact. Details of EHS consideration are available in *Subsection 4.2*. The historical assumption for EHS is that a hot start equals 1 EHS, a warm start equals 3 EHS and a cold start equals 5 EHS. Based on this assumption total lifetime EHS have been calculated for all units in the database.

Figure 6-2 shows the relationship between creep life consumption (measured in online hours) and raw (uncorrected) annual maintenance costs for combined cycle (CCGT) plants in the sample.

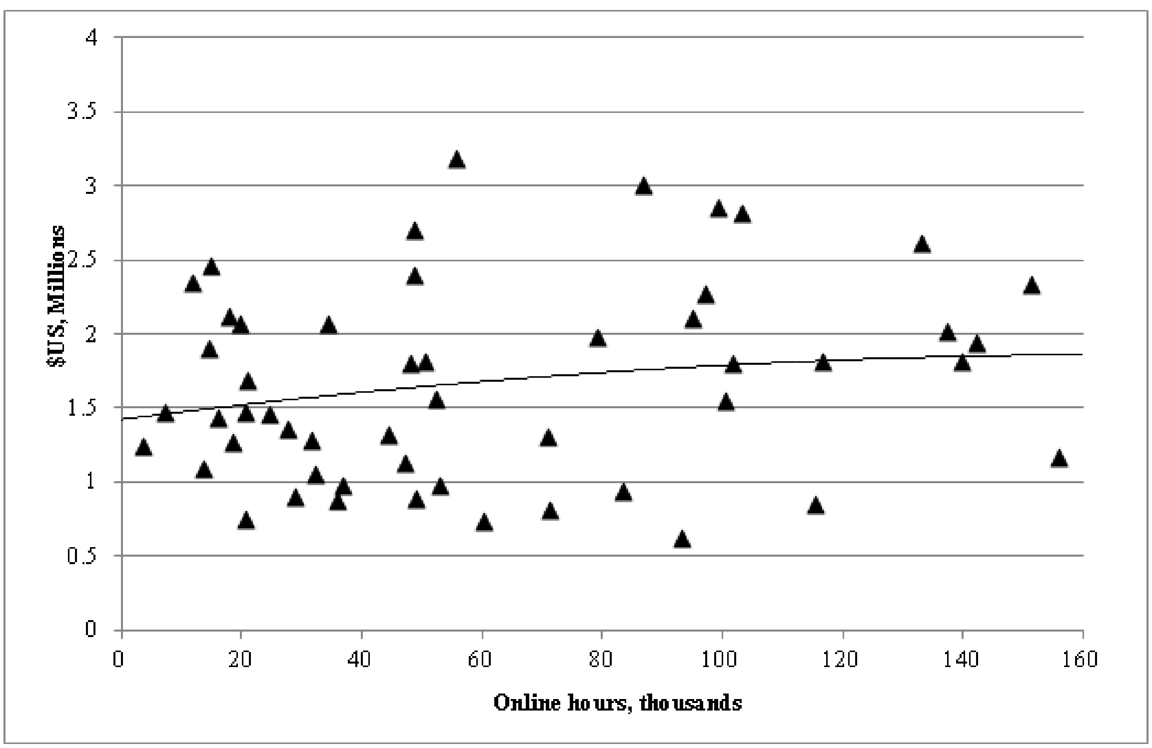


Figure 6-2
Raw O&M costs v. online hours for CCGT plants

Industry damage summation models assume that creep life is consumed in direct proportion to online hours, however, experience has shown that at plant (rather than component) level, the best fit for O&M costs is a 'bathtub' curve, reflecting a period of relatively high costs following commissioning when 'teething' problems are resolved, followed by a period of low O&M costs, which eventually increase as components begin to wear out. This profile is best described by a second order polynomial so the figures show the polynomial trend lines for the relationship, which for the raw data in Figure 6-2 has the form:

$$y \sim -2*10^{-5}x^2 + 5x + 1*10^6 \qquad \text{(Equation 1)}$$

Where,

x = Creep life consumption, in online hours

y = Annualized O&M cost

The goodness of fit between the data in this case is very poor, with a coefficient of determination (R2) value of 0.04, reflecting no practical correlation (the curve is also inverted from its bathtub shape).

Figures 6-3 shows the relationship between online hours and normalized costs (corrected for capacity by dividing reported costs by the cost correction factor) for the sample.

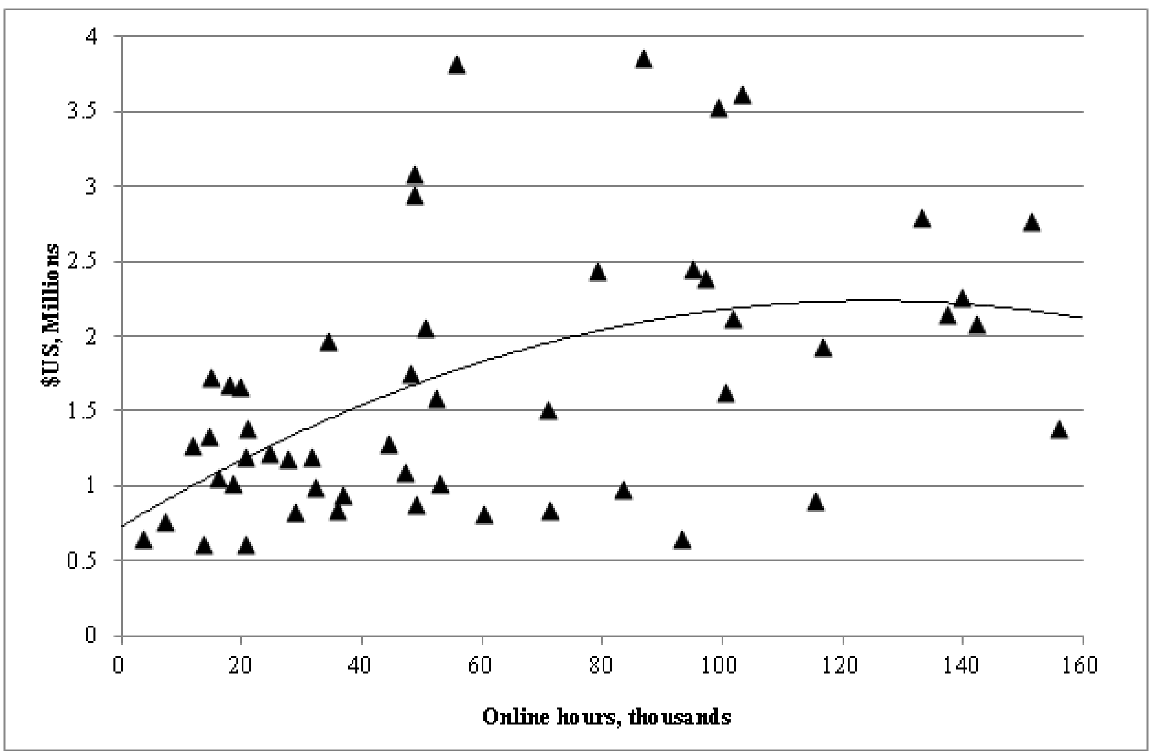


Figure 6-3
Corrected O&M costs v. online hours for CCGT plants

Color code the points to distinguish the various forms of cycling.

Applying the correction factor to the cost data results in a 2nd order polynomial trend line for Figure 6-3 of the form:

$y \sim -1*10^{-4}x^2 + 24x + 7.3*10^5$ (Equation 2)

Where:

x = Creep life consumption, in online hours

y = Annualized O&M cost

This results in a significant improvement in the coefficient of determination to a value of 0.19, however, this still represents a very weak correlation between online hours and O&M costs, and this curve also has an inverted bathtub shape.

Figure 6-4 shows the relationship between fatigue life consumption (measured in EHS) and raw (uncorrected) annual maintenance costs for combined cycle (CCGT) plants in the sample.

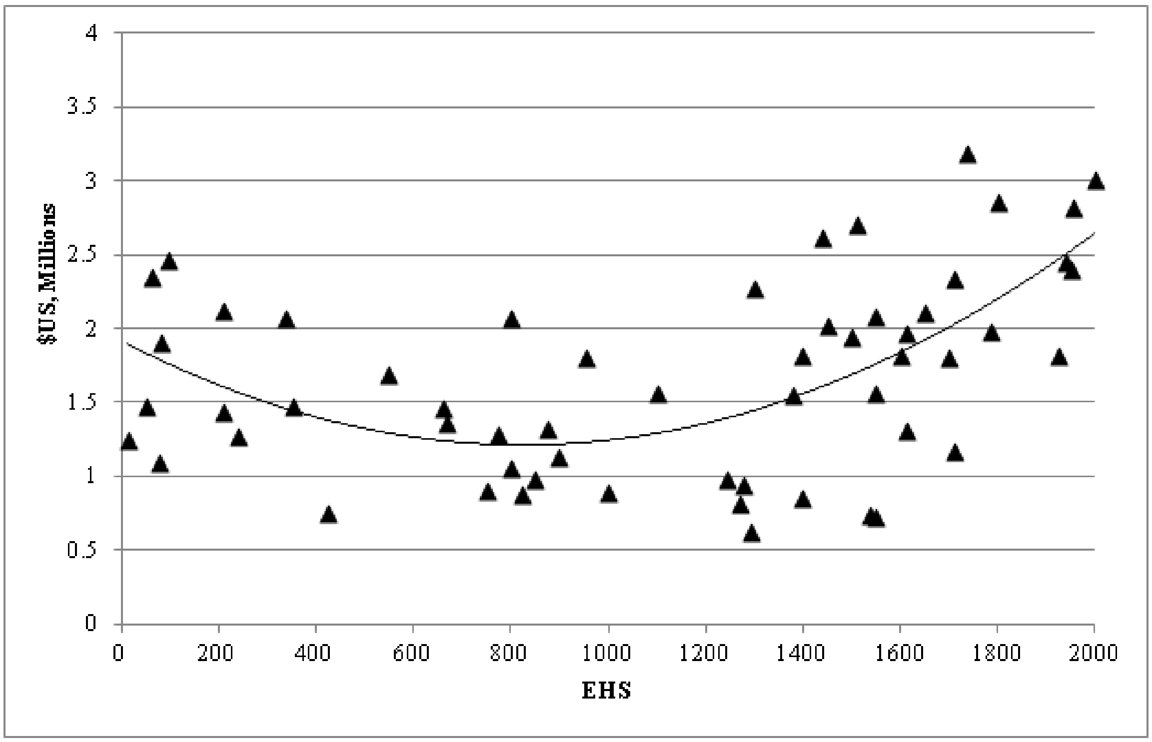


Figure 6-4
Raw O&M costs v. EHS for CCGT plants

Similar to Figures 6.2 and 6.3, the trend line is plotted as a 2 nd order polynomial which has the form:

$y \sim 1.04x^2 - 1720x + 2*10^6$ (Equation 3)

Where:

x = Fatigue life consumption, in total starts

y = Annual O&M cost

The trend line for this relationship shows an improved, though still relatively weak correlation with the data, with an R2 value of 0.35, however, because of the positive coefficient for the x2 term, the curve now has the correct profile.

Applying the correction factors to the cost data results in a significant increase in goodness of fit and a weak positive correlation of data, as shown in Figure 6-5.

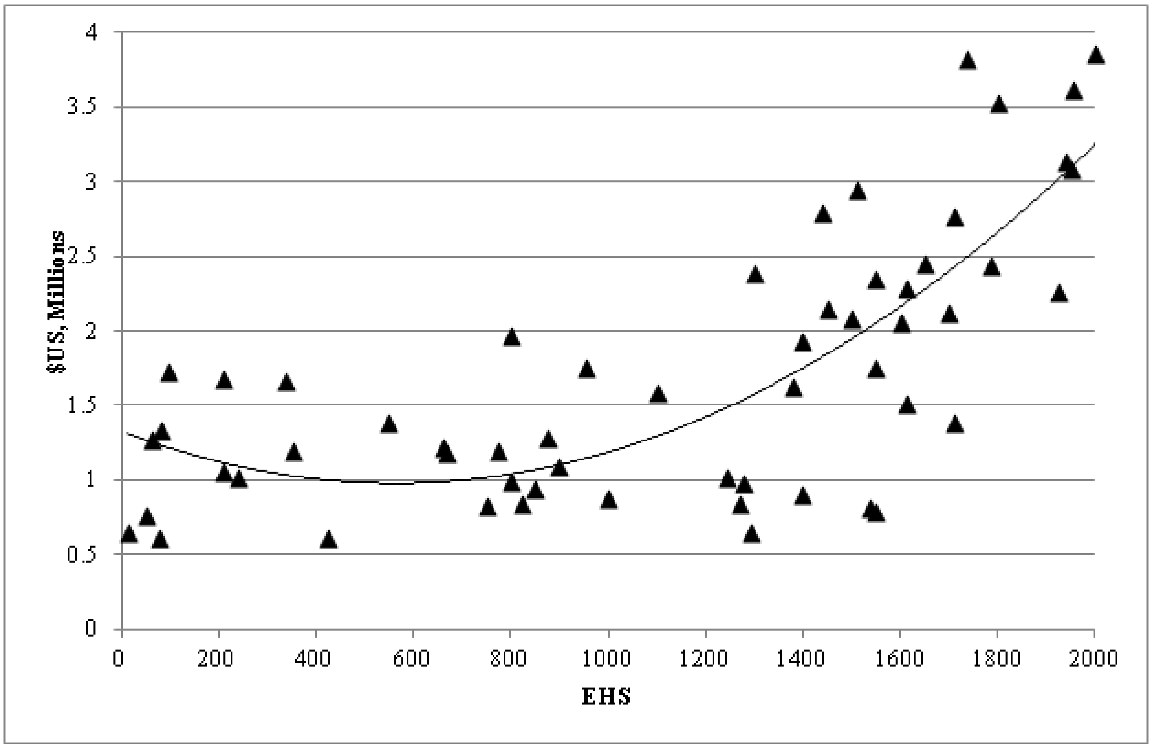


Figure 6-5
Corrected O&M costs v. EHS for CCGT plants

The trend line in Figure 6-5 is also plotted as a polynomial which has the form:

$$y \sim 1.1x^2 - 1250x + 1*10^6 \qquad \text{(Equation 4)}$$

Where:

x = Fatigue life consumption, in total starts

y = Annual O&M cost

This relationship shows a weak positive correlation of the data, with a coefficient of determination of 0.59. Again, the positive coefficient for the x^2 term means that the upright bathtub curve is restored, conforming to the observation that O&M costs are usually higher in the early (post-commissioning period) and late stages (wear out period) of the unit's service life.

Based on the analysis of the units in this database therefore, it appears that the strongest indicator of capacity-corrected CCGT annual O&M costs is the number of EHS that the unit has performed.

The trend line curve derived from the data is plotted for a model CCGT (350 MW), along with 95% confidence intervals (CI) in Figure 6-6.

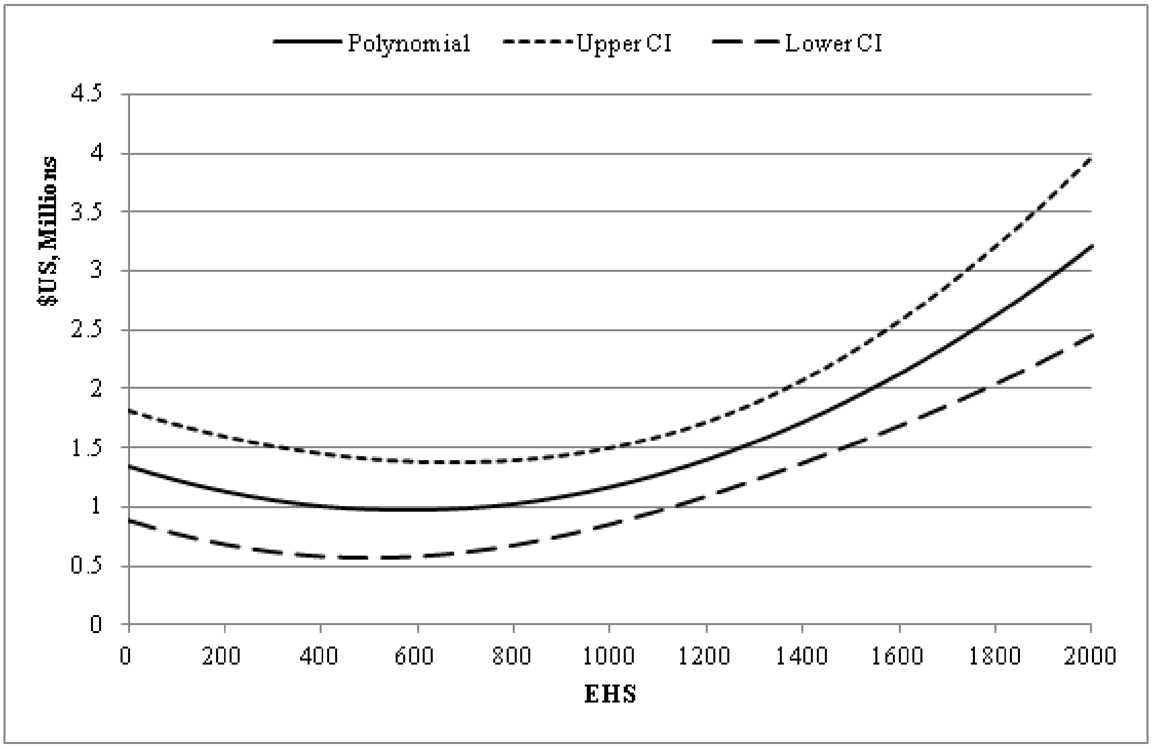


Figure 6-6
Predicted annual O&M spend for model 350 MW CCGT v. EHS with 95% Confidence Interval

In order to extrapolate these figures for a larger or smaller unit, the figures should be multiplied by the relevant correction factor. For example, for a model 450 MW units the correction factor is 1.2, and for a 250 MW unit the correction factor is 0.8, so after 1000 EHS the forecast annual O&M cost for these three units is shown in Table 6-1.

Table 6-1
O&M costs for model units 250MW, 350MW and 450 MW

Capacity	EHS	Polynomial	+95% CI	-95% CI
250 MW	1000	$940,960	$1,119,211	$762,709
350 MW		$1,176,200	$1,399,014	$953,386
450 MW		$1,411,440	$1,678,817	$1,144,063

6.2 Cost Assessment for Conventional Plant

The conventional units analyzed for this report included coal-, gas- and oil-fired generators ranging from 185 MW to 480 MW capacity. As previously described there were a total of 30 conventional plants with the cost data in the ETD database, of which relevant cost data were available for 20.

The scatter plot for the relationship between age in calendar years and total online hours is shown in Figure 6-7, while Figure 6-8 shows the scatter plot for EHS and age in calendar years. These plots are included in order to illustrate two important points regarding this analysis of conventional units; firstly, that all the units in the sample are over 20 years old; and secondly, that all have accrued over 150,000 online hours. This is in contrast with the CCGT sample, which comprised units ranging in age from less than one year to just over 20 years old, and all of which had accrued less than 160,000 online hours and 2,000 EHS. It should therefore be emphasized that because the cost data for these units is only available for recent years (i.e., the later stages of operating life), interpolation and assumptions regarding their maintenance costs in earlier years must be treated with caution.

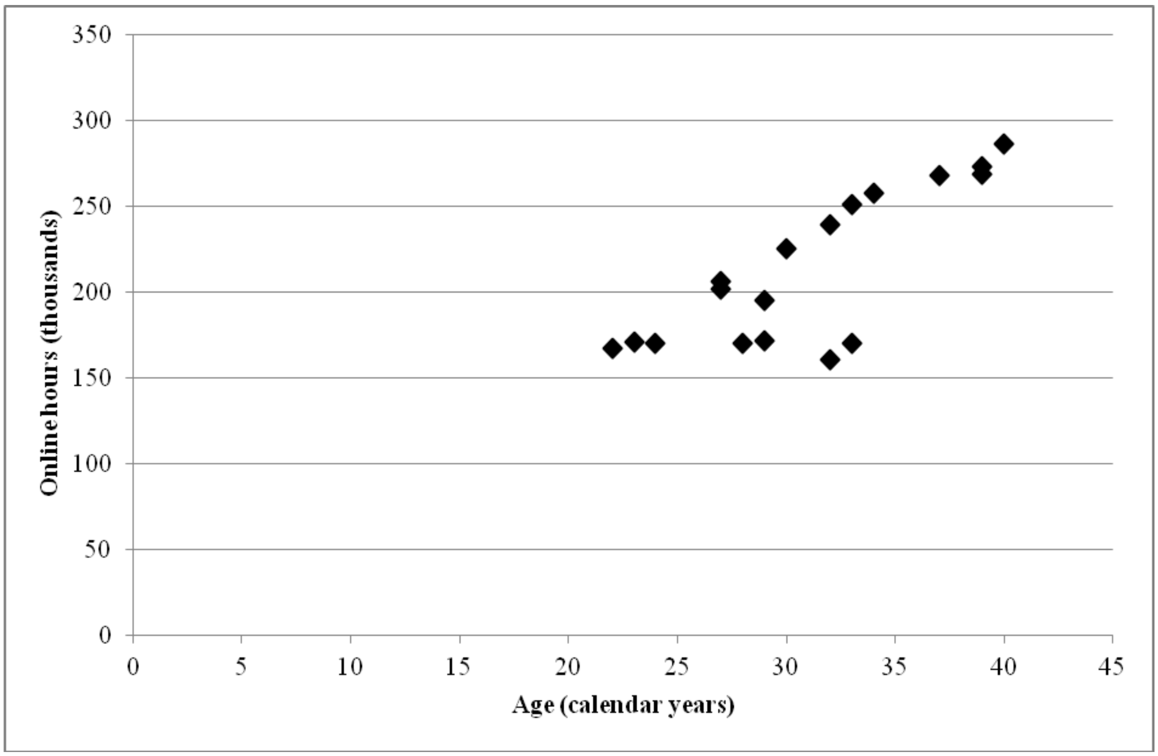


Figure 6-7
Age v. online hours for conventional plants

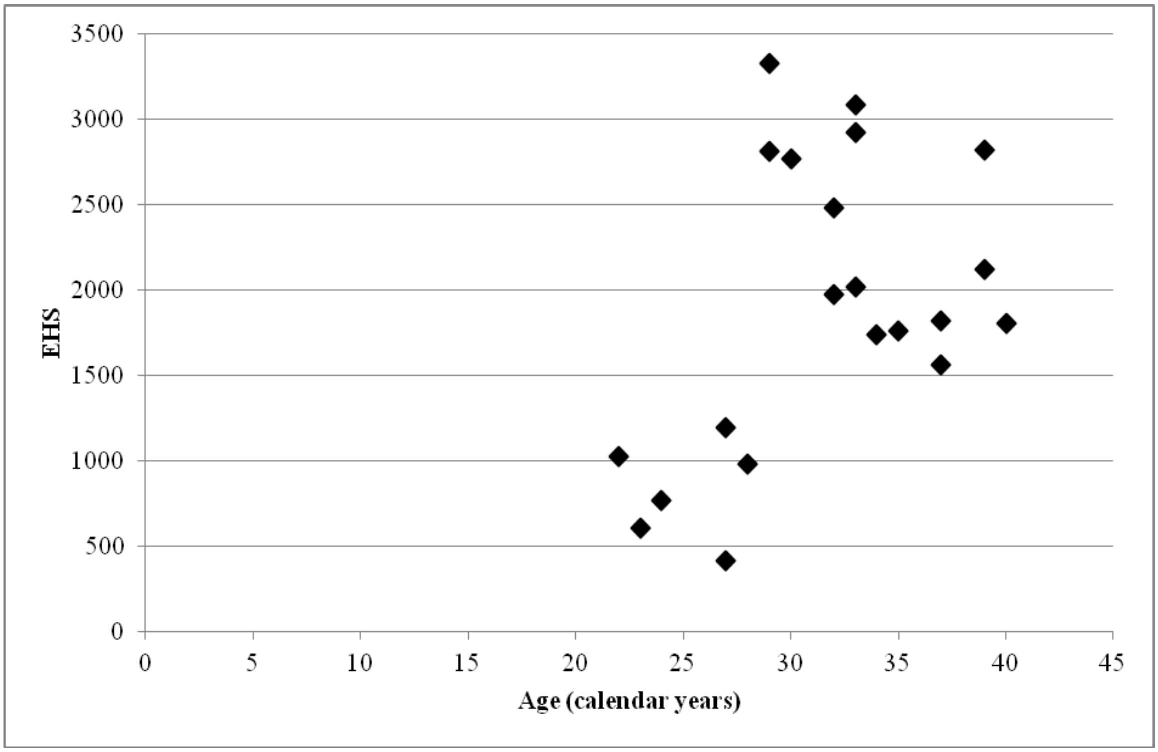


Figure 6-8
Age v. EHS for conventional plants

Bearing in mind, the proviso above regarding the interpolation of data points to the early years of the operating life of these units, a similar analysis to that undertaken for the CCGT units, is carried out here. In order to be able to plot a meaningful trend line for these plots, and to provide an estimate of the annual maintenance costs over the lifetime of these units, it was necessary to make an assumption about the initial costs of maintenance. In this instance, it is assumed that during the first year of operation, maintenance costs were $1.3 million. This is similar to, and based on, the figure derived for O&M costs for the CCGT units.

Based on this assumption, Figure 6-9 shows the relationship between EHS and corrected annual O&M cost. In contrast to the results for the CCGT units, the relationship shows weak correlation with an R^2 value of 0.48 and the inverted profile does not reflect the normal distribution of lifetime costs.

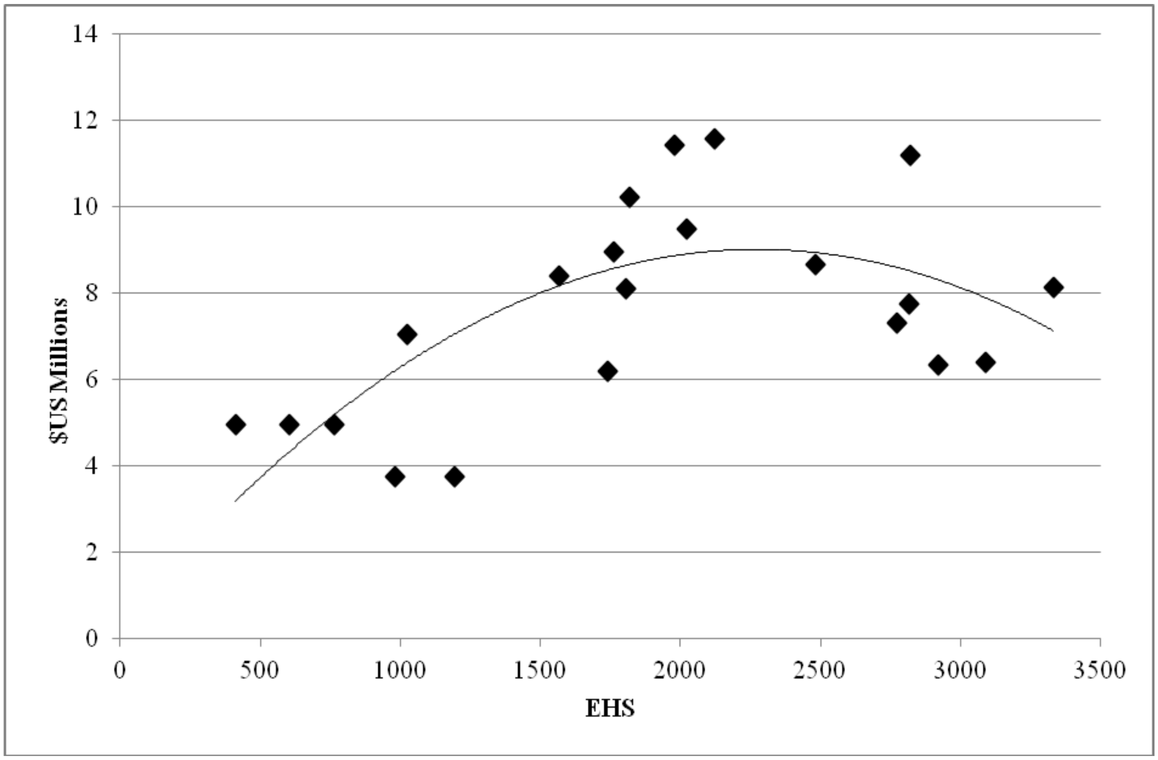


Figure 6-9
Corrected O&M costs v EHS for conventional plants

The trend line in Figure 6-9 has the form:

$y \sim -1.68x^2 + 7650x + 3.2*10^5$ (Equation 5)

(x = life consumption in total starts, y = annual O&M cost)

Coefficient of determination (R2) = 0.48

The relationship between lifetime online hours and annual O&M spend is shown in Figure 6-10. Overall, this relationship shows an improved reflection of the known life consumption profile of a typical unit. The coefficient of determination is slightly higher at 0.55 (close to weak positive correlation) and more importantly, the positive coefficient associated with the x^2 term.

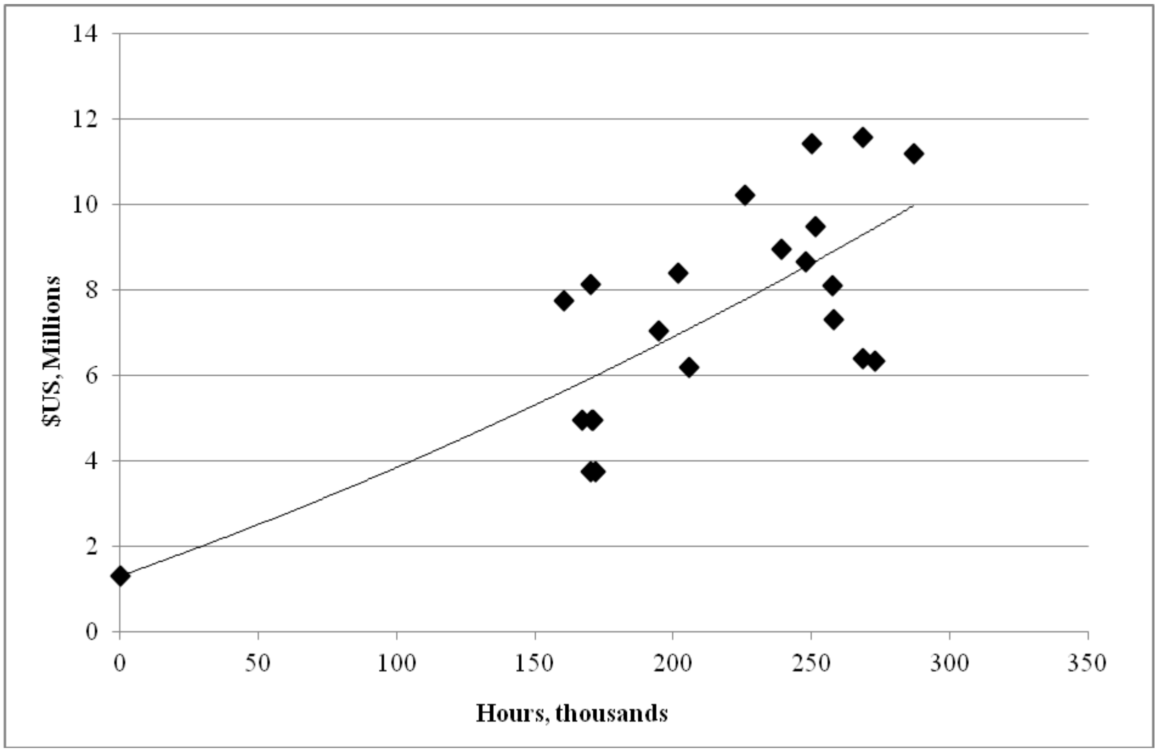


Figure 6-10
Corrected O&M costs v online hours for conventional plants

The trend line in Figure 6-10 is also plotted as a polynomial which has the form:

$y \sim 3*10^{-5}x^2 + 23x + 1*10^6$ (Equation 6)

(x = life consumption in total starts, y = annual O&M cost)

Coefficient of determination (R^2) = 0.55

The analysis of the conventional units suggests that in contrast to the CCGT units, annual O&M spend are more closely related to lifetime online hours. While the absence of cost data for the early part of the lives of these units, combined with the slightly weaker correlation between costs and hours (compared with the strong correlation between costs and EHS for CCGT units) makes extrapolation and interpolation slightly more risky for conventional plant, the best means of estimating plant O&M costs for these units is derived from Equation 6 above.

6.3 System Specific Analysis- A Case Study

The cost analysis in the previous Sections is based on units in many different power systems/electricity markets that have no strong correlation in operational history. Particular problem with extrapolating costs for CCGT beyond 2000 EHS or 150,000 online hours was that no data was available for this period. Similarly, there is a problem with the 'reverse extrapolation' of costs for conventional units in that the youngest unit available in the database was 23 years old. Therefore it was not possible to predict/ determine the O&M costs of, say, a 10-year old coal unit.

Those involved in this study have extensive plant, operating and cost information of a specific European system, which has allowed a far more detailed analysis to be performed within that system. The following analysis demonstrates the next level of assessment possible and is drawn from all the base load units in this particular system, including large coal and CCGT plant. The units were found to have very similar operational histories, whether conventional or CCGT. The results can be used as a guide for power generating units in North America (costs have been provided in $USD), however for more accurate / in-depth results it is recommended to perform specific analysis at the system level e.g. for ERCOT, PJM, MISO etc. territories, and would require data from generators within the system under analysis.

For units in this system the first correlations examined were those relating age in calendar years to creep and fatigue life consumption, measured in online hours and number of starts respectively. In order to avoid the possibility of individual units being identified only trend lines are plotted. The curve plotted on the primary y-axis in Figure 6-11 shows the power trend line for the relationship between the units' age and online hours, which has the form:

$$y \sim 7360\, x^{(24/25)} \qquad \text{(Equation 7)}$$

(x = age in calendar years; y = online hours)

The curve shows a strong positive correlation, with a coefficient of determination (R^2) of 0.9. It is also possible to plot the relationship as a quadratic (y ~ -109x2 + 9,440x) or higher degree polynomial, however, the simplest narrative description of the rate of unit creep life consumption, that the number of annual online hours performed by base load units tends to decrease with age, is most accurately represented by the simple power relationship in Equation 7.

The relationship between age in calendar years and fatigue life consumption, measured in EHS is plotted on the secondary y-axis in Figure 6-11. Data are represented by a power trend line for the relationship, which has the form:

$$y \sim 27\, \mathrm{x}^{(6/5)} \qquad \text{(Equation 8)}$$

(x = age in calendar years, y = EHS)

Again the curve shows a moderately strong positive correlation, with a coefficient of determination (R^2) of 0.8. As in the case of cumulative online hours, it is possible to use more complex functions to describe the relationship, however the straightforward narrative description of how units accumulate stop/start cycles over their lifetime, i.e., that there is a low rate of accrual of annual starts for new units, which increases over time, is adequately represented by a simple power relationship.

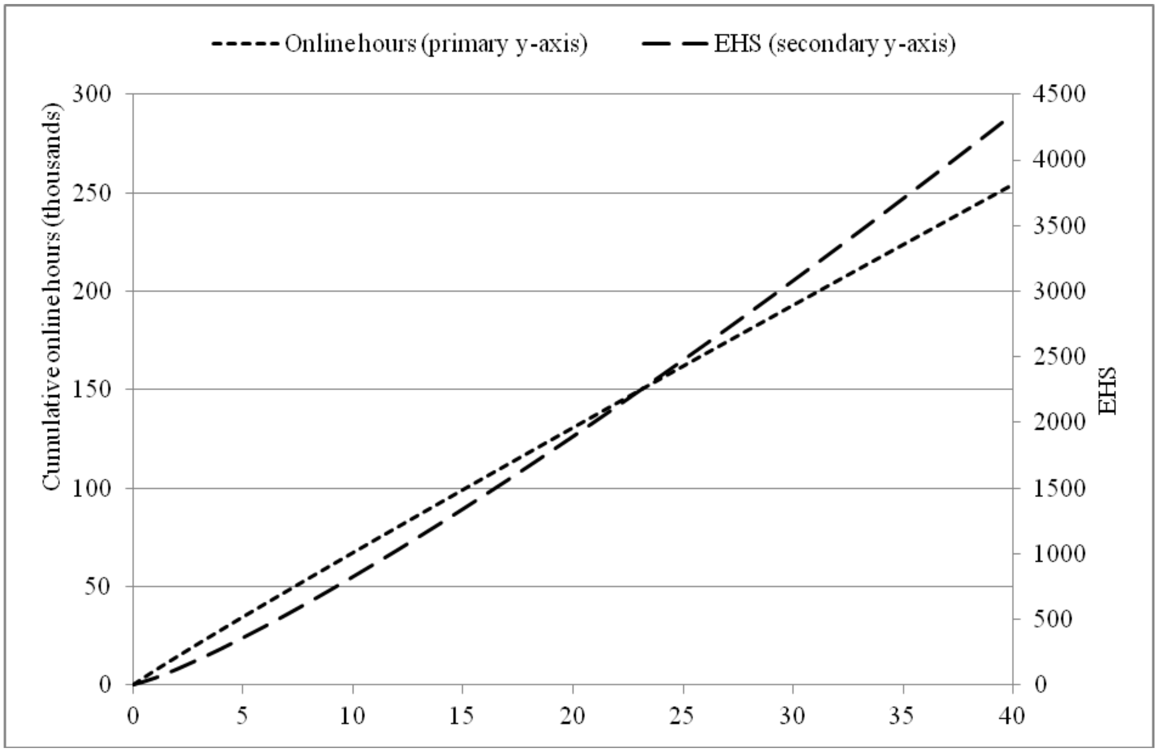


Figure 6-11
Power trend lines for creep life consumption (measured in cumulative online hours) and fatigue life consumption (measured in cumulative starts) v. Age (x-Axis) in calendar years

The relationships described by the curves in Figure 6-11 suggest that in line with the expected life cycle of most base load generators, as units in this system age, they accumulate online hours at decreasing rate, and starts at a higher rate. In terms of energy generated therefore, older units have to 'work harder', performing more stop/start cycles for fewer online hours. The high degree of similarity in the operational history of units in this system also allows broad assumptions to be made about the life consumption profile of a model base load generator.

In terms of absolute figures, according to Equations 7 and 8, after 15 years a model base load unit will have consumed most of its design life by accruing a total of just under 100,000 online hours and over 1000 EHS; after its effective life of 40 years this will have risen to over 250,000 online hours and over 4,000 EHS. By differentiating Equations 7 and 8 with respect to age in calendar years it is also possible to estimate the number of online hours and EHS that the unit will accrue in any one year.

For example, to estimate the online hours and starts in the unit's 30th year:

<u>Online Hours in year 30 (x = 30)</u>

Cumulative online hours after 30 years = 7,360 $x^{24/25}$ (Equation 7')

Online hours in year 30 = (7,360 * 24/25) $x^{-1/25}$ (by differentiating)

Online hours in year 30 = 6,167

EHS in year 30 ($x = 30$)

Cumulative starts after 30 years = 27 $x^{6/5}$ (Equation 8')

EHS in year 30 = (27 * 6/5) $x^{1/5}$ (by differentiating)

EHS in year 30 = 64

Cost Analysis

The polynomial trend line describing the relationship between creep life consumption (measured in online hours) and annual maintenance costs for units in this system was found to have the form:

$y \sim 8*10^{-5}\, x^2 - 10x + 7.5*10^5$ (Equation 9)

(x = online hours, y = annual O&M cost)

This trend line, however, exhibits a weak correlation with the data, with a coefficient of determination (R^2) of less than 0.2. For practical purposes therefore, it appears that there is no real connection between lifetime online hours and the annual non-fuel O&M cost for base load units in this system.

The polynomial trend line for the relationship between EHS and annual maintenance costs for base load units in this system was found to be:

$y \sim 0.25x^2 - 260x + 1*10^6$ (Equation 10)

(x = EHS, y = annual O&M cost)

This relationship shows strong correlation, with a coefficient of determination (R^2) of 0.8. The relatively high level of correlation between damage accumulation, as represented by annual non-fuel O&M cost, and fatigue-life consumption, represented by EHS, seems to confirm the assumption that for our model of a typical base load generator in this system, fatigue is the dominant indicator of annual maintenance costs.

Model development

Having established that Equations 8 and 10 (representing the relationships between age in calendar years and EHS; and between EHS and annual O&M cost) show good correlation with the data, it is possible to use them to represent a 'snapshot' of the service life profile of a typical base load unit in this system. This profile is used to create a model for this system which, by combining Equations 8 and 10 and correcting for capacity, can be used to generate a per-start cost for units of any age and capacity, from all cooling conditions. Using this information the hot, warm and cold per-start cost for a 350 MW unit after 15 years of operation can be calculated as follows:

Total EHS, year 15 ($x = 15$):

$27x\text{^}^{6/5} = 696$

Annual O&M spend, year 15 ($x = 696$):

1.1 (Capacity correction factor) * $0.25x^2 - 260\, x + 1*10^6$ = \$1,041,603

EHS in year 15 ($x = 15$):

Cumulative EHS = $27\ x^{6/5}$

EHS in year 15 = $(27 * 6/5)\ x^{1/5}$ (by differentiating)

EHS in year 15 = 56

EHS cost:

$1,041,603/56 = $18,600

Per-start costs for our model unit (not including fuel) in year 15 are therefore:

Hot start = $18,600 (EHS cost*1)

Warm start = $55,800 (EHS cost*3)

Cold start = $93,000 (EHS cost*5)

Modifications can be made to the model to adjust for changes in the number of annual starts performed due to factors such as high wind penetration or market liberalization. For example, analysis of the starts performed by base load units in this system before and after market liberalization showed an increase of 15%. If this increase is assumed to be an ongoing effect of market competition, an additional coefficient (1.15) representing this increase in starts can be incorporated into Equation 9' for the period following market liberalization. The impact of this is to accelerate the annual rate of fatigue life consumption, effectively speeding up the ageing process. Similarly, scenarios with further additional starts due to high levels of renewable energy penetration can be created by changing the coefficient to represent further increases above baseline (10%, 20% or 30%) in EHS performed annually.

Figure 6-12 shows the modeled 30-year cost profile of our notional 400 MW unit. The plot shows the baseline annual O&M costs (corrected for capacity) as calculated using Equations 8 and 10, along with the costs associated with the summation of four increased cycling scenarios; a 15% increase on baseline cycling as a result of marketization in Year 10; and further 10%, 20% and 30% increases in cycling, reflecting energy policies for low, medium and high wind power deployment from Year 5 onwards.

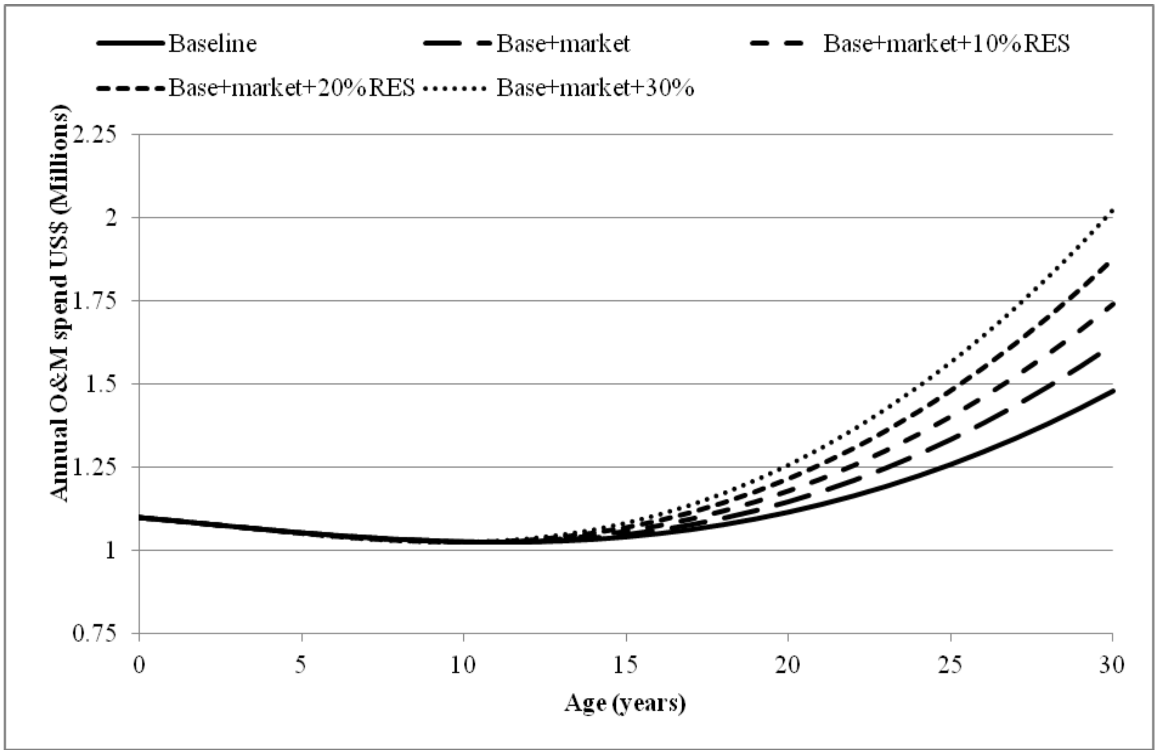


Figure 6-12
Annual O&M spend v age in a range of plant cycling scenarios

The example below shows the resulting per-start costs for a model 400 MW unit after 25 years of operation. In addition to the starts incurred in line with the pre-market profile, the impact of market liberalization after Year 10 and a 30% increase in the level of cycling associated with a policy of high wind power deployment after Year 5 are included in the forecast. This resulted in a cumulative total of 1788 EHS broken down as follows: 1305 baseline EHS (years 1 to 25); plus 137 EHS (years 10 to 25) due to the effect of market liberalization; plus a further 346 EHS (years 5 to 25) due to a high penetration of wind power, as summarized in the equation below:

Total EHS after 25 years =

$$\sum_{1}^{25}\left(\frac{6}{5}\times\ 27x^{\frac{1}{5}}\right)+\sum_{10}^{25}\left(0.15\ \times\frac{6}{5}\times\ 27x^{\frac{1}{5}}\right)+\sum_{5}^{25}\left(0.3\ \times\frac{6}{5}\times\ 27x^{\frac{1}{5}}\right)$$

$= 1305 + 137 + 346$

$= 1788$

Using this result as the independent variable in Equation 10, and correcting for capacity, we calculate that the annual maintenance cost for the unit in this after 25 years is:

Annual maintenance cost (x = 1788):

1.1 (Capacity correction factor) * $0.25x^2$ -260 x + 1*10^6 = \$1,567,807

Using the same process for calculating EHS cost outlined above:

EHS in year 25 ($x = 25$):

$= (27*6/5\ x^{1/5}) + (0.15*27*6/5\ x^{1/5} + (0.3*27*6/5\ x^{1/5})$

$= 62 + 10 + 19 = 91$

EHS cost

$1, 567,807/91 = $17,229

Per-start costs in year 25 in this scenario are therefore:

Hot start = $17,229 (EHS cost *1)

Warm start = $51,687 (EHS cost *3)

Cold start = $86,145 (EHS*5)

Note that the per-start costs have actually decreased, when compared with the per-start costs for the 400 MW unit after 15 years. This is due to the fact that although annual O&M spend has increased by over 50% (from $1,041,603 to $1,567,807), the number of annual EHS has increased by 63% (from 56 EHS to 91 EHS). In fact using this model results in a curve shape which largely follows the bathtub profile of annual O&M spend over the model unit's lifespan (see the base curve in Figure 6-13).

The process described provides a means of estimating the cost of a hot, warm or cold start in any year of our model unit's service life. Theoretically, because fatigue damage is a cumulative, continuous process, each individual start should have a unique cost associated with it, however this does not reflect the way that plant owners actually assess expenditure. Discussions with owners and operators show that in most cases O&M costs are re-calculated and changed only rarely, typically annually, following a detailed assessment of units' condition during annual shutdowns. The level of cycling carried out in the previous year is then related to the annual O&M costs to calculate a per start cost for the next year of operation. Rather than a continuous curve reflecting an incremental unique cost for each start throughout a unit's service life, the stepwise, annually changing distribution produced by the method described above is therefore a better representation of real world costing.

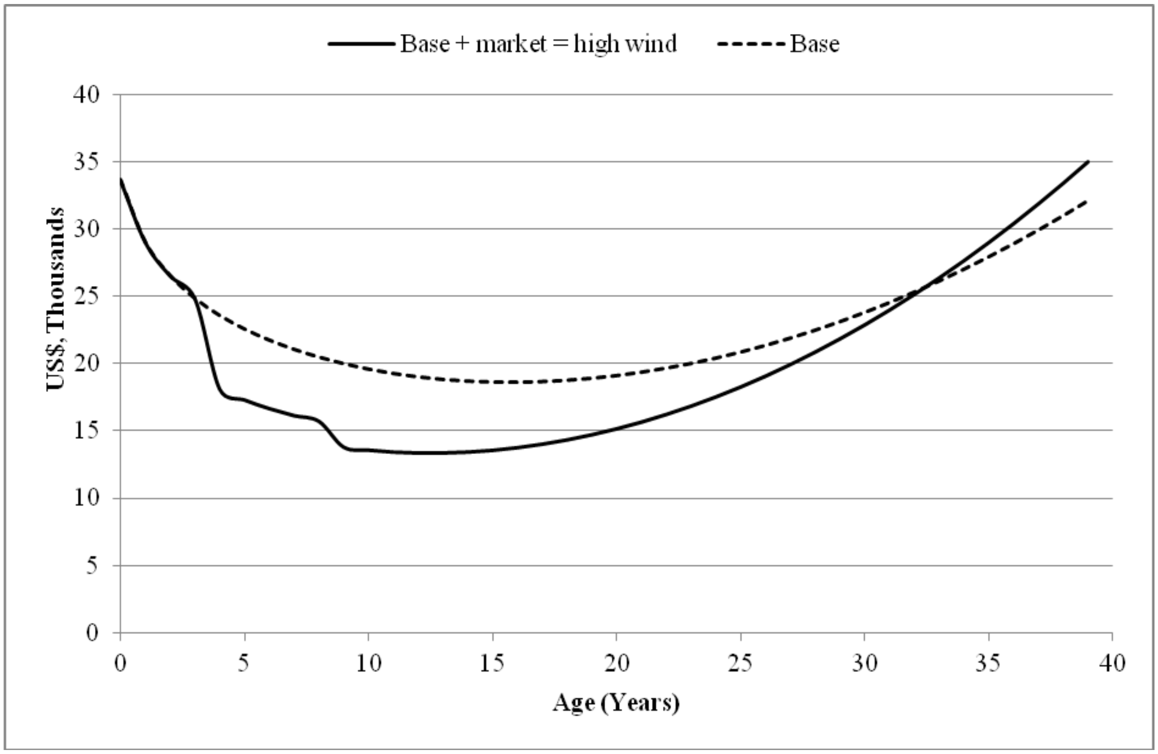


Figure 6-13
Hot start cost v. unit age

7
OTHER CONSIDERATIONS FOR CYCLING COST ASSESSMENT

Proper monitoring of start-up costs is essential to remain competitive in the electric power marketplace. This cannot be effectively done without detailed cost analysis and careful control of the Operation & Maintenance (O&M) activities. This Section provides description of some other factors which are market specific and were not included in the cost model.

The difficulty of a cost analysis process lies in establishing the degree to which each of the factors will be applicable to any specific plant. The cost model described above considered non-fuel O&M costs and also not included loss generation (replacement power costs) due to forced and planned outages which are market specific factors. These factors will also influence or even determine the degree to which cycling will impact on the overall O&M costs. Therefore, it is advisable to consider these factors as well for the proper understanding of the impact of cycling on O&M costs. These specific factors are described below.

7.1 Start-Up Fuel Costs

The start-up fuel cost depends on the type of start-up i.e. hot, warm or cold start-up (definition of these start-up conditions are described in *Subsection 4.2*). Fuel cost is one of the variables more difficult to predict since it depends exclusivity on the type of contract between the power plant and the fuel provider company and also on the current market price of the fuel. Generally, this cost depends on market changes but in some cases may also depend on lobbies and/ or other business strategies.

The start-up fuel cost also depends on the energy consumption by the plant. This energy consumption is clearly depended on plant size i.e. net capacity. This issue is well discussed in the following Subsection by plotting energy factor versus net capacity to develop the start-up energy consumption model. Start-up fuel costs may be determined from the current fuel price by considering this model

7.1.1 Start-Up Energy Consumption

This represents the energy required (energy consumption of the plant) to increase the initial temperature to the required operating temperature during start-up of a plant. Plant start-up energy consumption is broadly related to net capacity. The energy consumption generally increases with the unit size and remains practically the same until the end of the plant life considering the plant is well maintained e.g. chemical cleaning is performed on a regular basis in order to maintain the heat transfer efficiency. Figures 7-1 and 7-2 provide start-up energy consumption model for various plant capacity for conventional and CCGT plant. In these figures, the start-up energy consumption is represented as a factor against the plant net capacity.

The ***conventional unit*** in the database for start-up energy calculation range between 54 MW and 285 MW capacities. The power curves plotted from the conventional plants' data show a weak positive correlation, with a coefficient of determination (R2) 0.65, 0.67 and 0.72 for hot, warm and cold start-up, respectively (Figure 7-1). The start-up energy factor for conventional 'load follow cycling regime' is also represented in the Figure 7-1 and the power curve for load follow data show a weak positive correlation, with a coefficient of determination (R2) 0.64.

The equations of the power trend lines for hot, warm & cold start-up and load follow operation are provided below:

Hot start-up: $y = 0.0012x^{1.1732}$

Warm start-up: $y = 0.0132x^{0.835}$

Cold start-up: $y = 0.0158x^{0.862}$

Load follow: $y = 3E\text{-}05x^{1.6447}$

Where, y = Start-up energy factor

x = Net capacity, MW

These equations need to be multiplied by the known current market Unit price of the fuel to calculate the Unit cost of fuel/ energy.

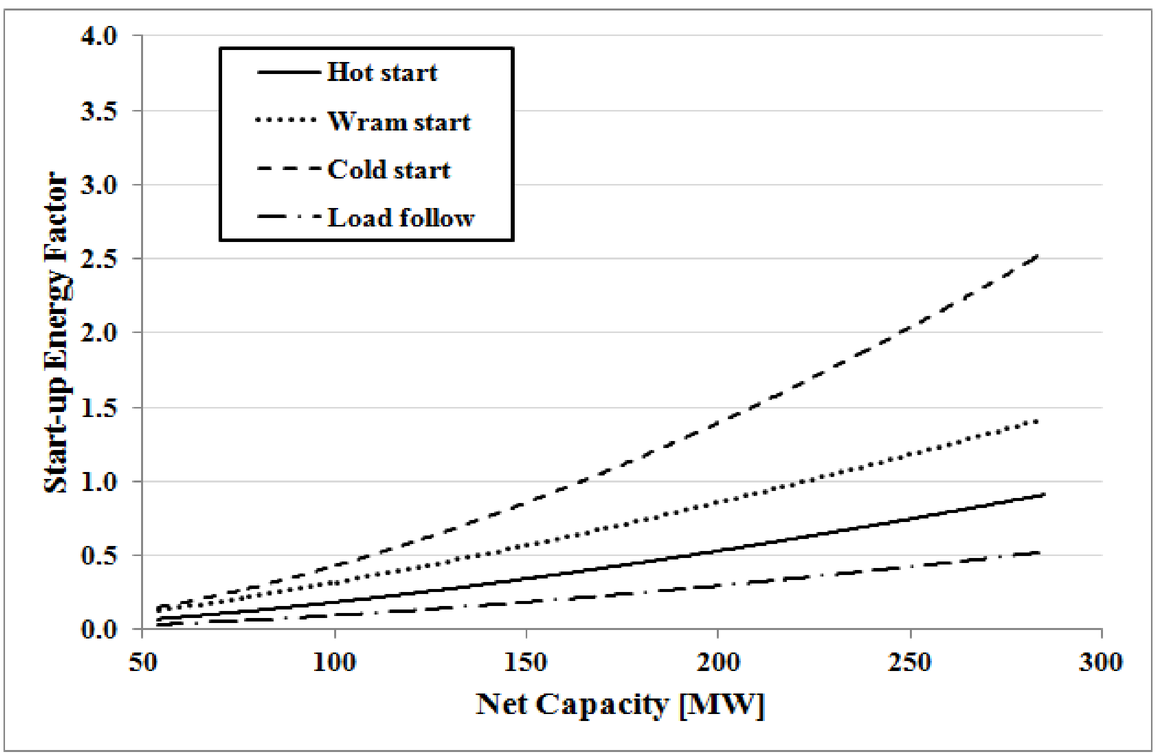


Figure 7-1
Start-up energy factor versus net capacity for conventional plant for various start-up conditions and load follow cycling regime

The ***CCGT unit*** in the database for start-up energy calculation range between 88 MW and 463 MW capacity. The power curves plotted from the CCGT plants' data show a weak positive correlation, with a coefficient of determination (R^2) 0.68 for hot start and a moderately strong positive correlation, with a coefficient of determination (R^2) 0.86 and 0.79 for warm and cold start-up, respectively (Figure 7-2). The start-up energy factor for CCGT 'load follow cycling regime' is also represented in the Figure 7-2 and the power curve for load follow data show a weak positive correlation, with a coefficient of determination (R^2) 0.62.

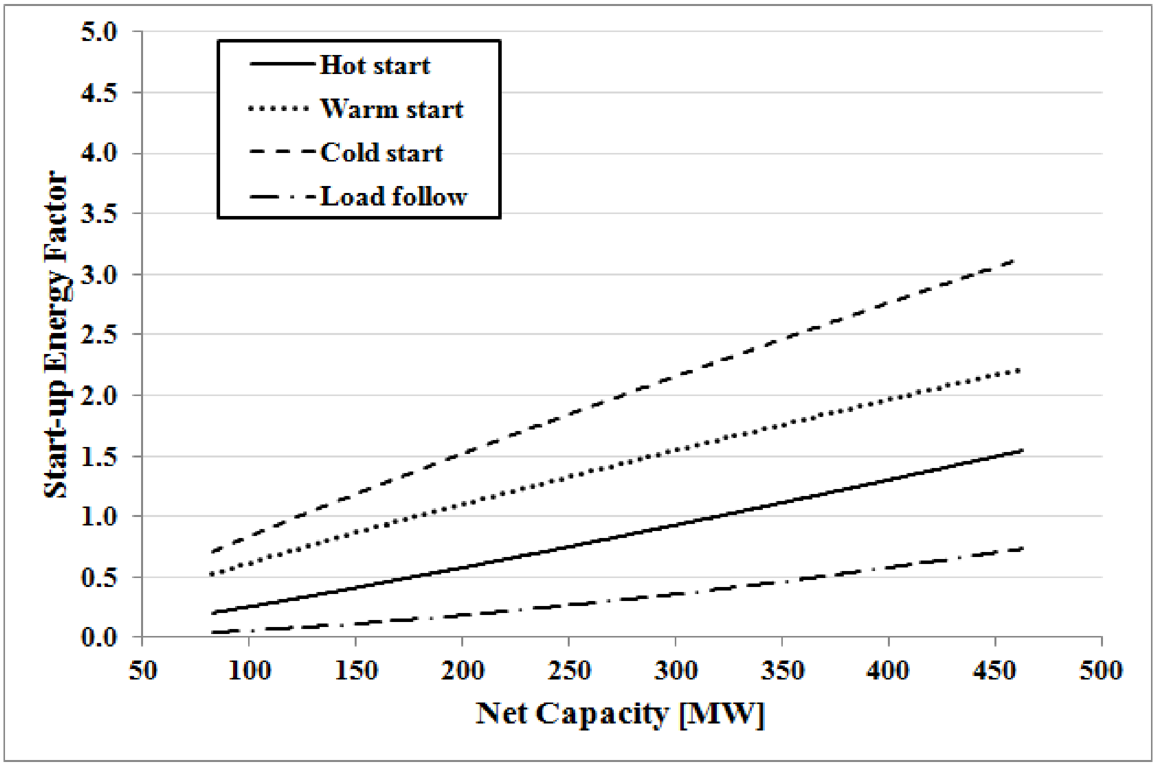


Figure 7-2
Start-up energy factor versus net capacity for CCGT plant for various start-up conditions and load follow cycling regime

7.2 Lost Generation

It is already mentioned that the cycling cost assessment procedure did not include loss generation (replacement power costs) due to forced and planned outages which are market specific factors. This is the replacement power due to loss of availability and the cost for this may be determined from the current market price of electricity of the specific system. Some other lost generation costs (i.e. load reduction due to demand) may depends on the power purchase agreement of specific unit.

8

MAXIMIZATION OF EQUIPMENT MAINTAINABILITY AND EFFECTIVENESS

Many utilities are nowadays focused on their performance and constantly worried with the possibility of not being capable of meeting the market demand. A plant is considered unreliable if it is not capable of coping with the market fluctuation as a result of unpredictable demand. Start-up Reliability, Availability and Life Cycle cost become the most important factors in the power business sector. The fact is that maintenance and operation performance achieves a far greater importance than plant thermal efficiency resulting in the development of new maintenance philosophies and techniques.

This Section made recommendations for the improvement of plant management, operation, monitoring, design, staff levels and training to optimize plant life cycle costs. These recommendations are based on the latest maintenance philosophies and techniques being applied at present in only very few plants.

8.1 Long Term Service Agreement and Its Limitations

Nowadays new power plants are presenting an integrated maintenance strategy e.g. for the case of a combined cycle the turbo generator groups generally present a maintenance schedule based on OEM agreement, HRSGs maintenance schedules are dictated by turbo generator schedule and normally involves life assessment. Auxiliary systems more commonly known as balance of plant (BoP) maintenance strategy is based on a combination of reactive, and in some cases preventive, maintenance. Only a few plants are implementing some degree of condition based maintenance that generally is concentrated on BoP systems leaving turbo machinery to be controlled by time based maintenance.

Long-Term Service Agreements (LTSA)/ Long-Term Repair Agreements (LTRA), sometimes also known as Contractual Service Agreements (CSA), for the long-term equipment maintenance and service programs in large plants (especially for advanced gas turbines) have become the norm, due to, in most cases, the insistence of the long-term note holder. LTSAs typically commit the original equipment manufacturer (OEM) to providing, on a “fixed-priced” basis, maintenance services for the complex and sometimes untested advanced gas turbines. LTSAs offer some advantages to owners and operators such as:

- Fixed long-term maintenance costs
- Availability of parts due to incentives for OEM support
- Contractually guaranteed availability and reliability of the plant
- Performance power and heat rate guarantees

Nevertheless, the LTSAs due to their very complex, legalistic language and high costs are hard for many operators to fully understand. Some of the disadvantages of LTSAs are:

- High maintenance costs therefore high Life Cycle Costs
- Long-term relationship that may not easily be dissolved
- Owner to bear an inordinate amount of risk if contract is not properly negotiated
- Lengthy litigation if contracts are not analyzed properly

Plant operators often come in after the contract has been negotiated and do not fully understand the scope of the contract, which may result in costly and time-consuming disputes with the OEM.

Time based maintenance that in many cases does not fit in with the operation strategy, may lead to an inefficient and expensive maintenance program without any contribution to improved plant reliability and availability.

In addition to the time based maintenance many utilities are focusing their attention on major components and forgetting that minor components have considerable impact on plant Reliability, Availability and more importantly on cycling costs, an example is provided in Figure 8-1. In many cases it affects the start-up reliability so much that it reduces the power plant ranking in the merit order.

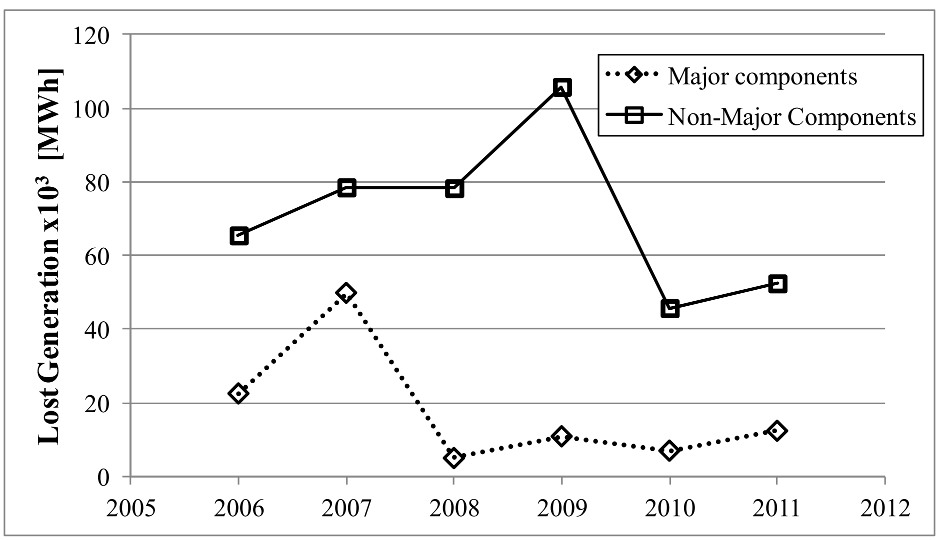


Figure 8-1
Lost generation due to major component failure versus Non-Major components

(Data based on a generation unit with a configuration of 2GT-2HRSG-1ST)

LTSA's are clearly a good asset during the plant first commissioning/ new start period since during this time there are a large amount of failures related to the quality of construction. After the new start period (~ 6 years) the long term agreements appear not to be entirely cost effective resulting in high planned maintenance and in most cases not completely designed for that particular plant. These limitations bring serious disadvantages for cycling plants competing for the merit order where plants with the lowest marginal cost of production are the first ones to be brought online to meet market demand. A solution to overcome this limitation would be the application of more flexible and specific maintenance strategies with clear goals.

As an example some of the techniques capable to overcome this problem could be:

- Condition Monitoring
- Reliability Centered Maintenance (RCM)
- Maintenance Basis Optimization

The following *Sections* describe the above mentioned techniques from the Life Cycle Cost optimization point of view.

8.2 Reliability Centered Maintenance and Condition Monitoring

A maintenance strategy which some utilities have been applying with success incorporates Reliability Centered Maintenance (RCM), Condition Based Maintenance (CBM) and Enterprise Asset Management (EAM).

Reliability Centered Maintenance is a technique that seeks to ensure that any equipment continues to do what it is designed to do. Reliability engineering is concerned with predicting and avoiding each failure as well as assessing the cost of a failure if it occurs. RCM identifies the most important functions of a piece of equipment to preserve with routine maintenance; the dominant failure status and causes are determined and consequences of failure ascertained. The consequences of failure are categorized into levels of criticality. Some functions are not critical and demand less attention while others must be preserved at all costs. This process directly addresses maintenance preventable failures. When the risk of failure is very high, RCM will sometimes demand that the user considers changing or re-designing a component which will reduce the risk to a tolerable level.

The maintenance management model of some power plants is driven by Reliability Centered Maintenance which maintenance strategies or tasks are designed to avoid failures or minimize the consequences of a failure. The maintenance package is then planned, scheduled and incorporated into the CBM and EAM.

To ensure maintenance management can achieve the business objectives such as plant availability and forced outage, maintenance performance needs to be reported. This would then be used as input for initiating the improvement cycle. This improvement program will trigger revision of the maintenance strategies and/or change in the way utilities operate the plant. This is performed continuously to ensure that any operator can deliver what is expected by the owner. The above can be illustrated in a simplified diagram as shown in Figure 8-2.

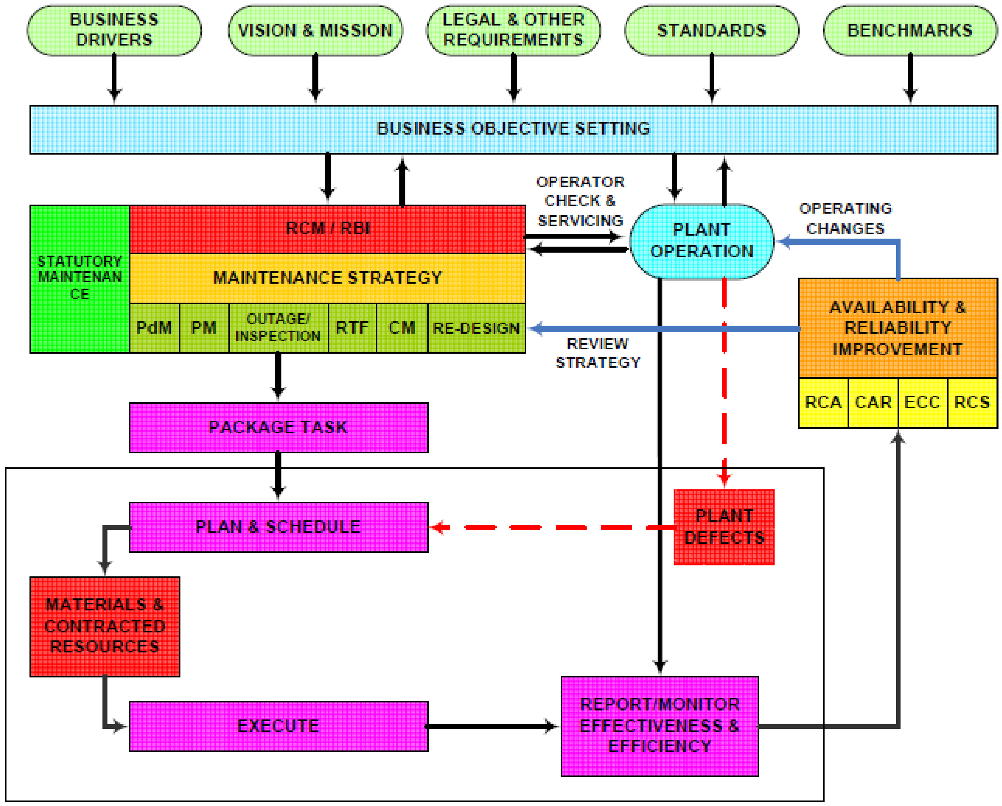


Figure 8-2
Example of a maintenance strategy based on RCM

The maintenance philosophy/ strategy should be based on the belief that it is possible to avoid unwanted equipment failures and unnecessary maintenance activities through appropriate application of various techniques supported by proven processes and procedures carried out by well-managed and competent personnel.

The need for both high Reliability and Availability of plant, and for cost-effective maintenance are compatible objectives. Operation and maintenance needs to share common objectives and are mutually dependent. The effectiveness of the maintenance effort is influenced by the relationship between those responsible for operating the plant and those responsible for its maintenance; therefore, it is crucial that operation and maintenance staff communicate with each other on a daily basis in order to effectively manage the plant life cycle.

8.2.1 Maintenance Basis Optimization

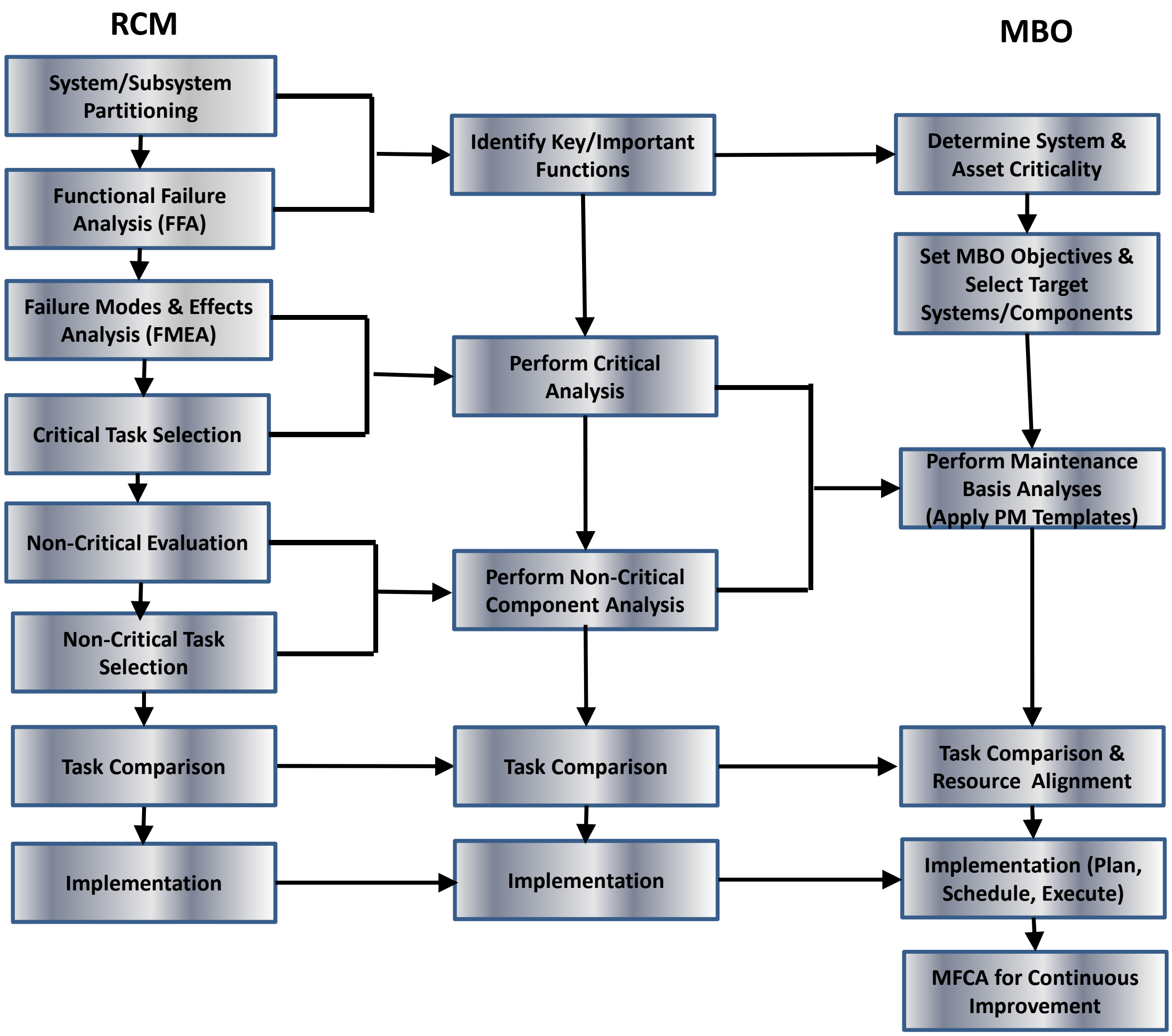
The application of classical RCM at fossil power generation stations required a significant amount of resources. In many cases, the resulting implementations were only marginally effective due to inability to complete the extensive analyses or failure to support the implementation of recommended PM task changes.

In the early 1990s, EPRI addressed the utility concerns that classical RCM required too many resources to perform the analysis and implement the resulting PM task changes. EPRI embarked on various projects to investigate possible methods of lowering the cost to perform an RCM analysis while maintaining the technical integrity of the process and results.

The Maintenance Basis Optimization (MBO) process resulted from the need for a simpler form of the RCM process that would minimize the effort to develop the optimum set of PM-RR and PM-CMT tasks to ensure high levels of equipment reliability at minimum cost. Over these past two decades, EPRI has developed and continually updated the Preventative Maintenance Basis Database (PMBD) [6]. The PMBD has incorporated more than 200 power plant components, PM templates, and bases information, and more are currently being added. This database represents years of collective RCM knowledge with power industry product manufactures, end users, and maintenance specialists. The PMBD component data templates have formed the foundation of many plants' MBO process implementation efforts. If used effectively, the PMBD can significantly decrease the time to perform the analysis associated with optimizing PM tasks.

The *MBO process* is defined as a process used for establishing or reviewing the maintenance basis for plant production equipment. This process typically compares existing PM tasks and MBs to best-practice PM templates and makes the appropriate adjustments—such as PM task additions, PM task deletions, or PM task frequency changes—to establish new MBs. The MBO process maintains the technical integrity as designed by the RCM process; however, MBO uses industry-standard component failure data and PM task information to minimize the analysis time.

Today, when a fossil power generation company embarks on improving their PM bases, the MBO process, which depends heavily on the use of component PM templates, is typically applied. Figure 3-2 depicts the transition and comparison between RCM and MBO processes. The MBO process is quite scalable and can be applied to all plant systems, targeted systems, or component classes, based on an organization's specific improvement targets.


RCM
MBO
System/Subsystem Partitioning
Functional Failure Analysis (FFA)
Failure Modes & Effects Analysis (FMEA)
Critical Task Selection
Non-Critical Evaluation
Non-Critical Task Selection
Task Comparison
Implementation
Identify Key/Important Functions
Perform Critical Analysis
Perform Non-Critical Component Analysis
Task Comparison
Implementation
Determine System & Asset Criticality
Set MBO Objectives & Select Target Systems/Components
Perform Maintenance Basis Analyses (Apply PM Templates)
Task Comparison & Resource Alignment
Implementation (Plan, Schedule, Execute)
MFCA for Continuous Improvement

9
CONCLUSION

- Conventional Power Plants and Combined Cycle Power Plants are increasingly being subjected to load-following and/or cyclic operation. However, cyclic operation introduces new types and higher rates of damage, and hence can result in *reduced plant performance* and *increased operation & maintenance and repair costs.*
- Plant performance was determined by two main factors; these are planned and forced outages described as Planned Outage Factor (POF) and Forced Outage Factor (FOF). The POF and FOF are used to derive the performance metrics of Equivalent Forced Outage Factor (EFOF), Equivalent Planned Outage Factor (EPOF), Availability and Reliability.
- A detailed analysis of the plant performance results shows that the cycling operation generates a considerable impact on the plant performance level. From the analysis, three distinct areas are identified for the entire life cycle of a power plant operating either base load or cycling regime and they form the classic 'bathtub' shape curve (typical to most plant equipment). These three areas characterize the different forced outage levels for the entire life cycle of the plant and consist of 'initial service period' in approximately the first 1-6 years, the 'useful life period' at approximately 6-20 years life and the 'major component wear out period' after these first two periods.
- Analysis of Reliability (R) and Availability (A) revealed lower values during the earlier plant life and a steady increase from age 6 until age 20 years achieving the maximum value possible between ~ 15 to 20 years of operating life. The A & R decrease abruptly during the later part of the plant life as the components of a cycling conventional and CCGT plant will show increase in failure rate and in downtime resulting in low Reliability and Availability.
- Analysis of the breakdown frequency according to plant area (or component) revealed that the number of failures is generally higher in superheater tubes in boiler units for conventional plants and the highest levels in the GT section of CCGT's, with creep-fatigue being the dominant failure mechanism and the rotating blades of the hot gas path section having the highest frequency of failures within the GT in terms of critical components. The HRSG section with the highest frequency of failures was the LP Economiser, and not unexpectedly FAC was the dominant culprit in terms of failure mechanisms. For the BoP the protection systems such as fire, lightning and breaker protection were the most frequent types of failure in CCGT plants in this study. A comparison between conventional and CCGT plants were also presented which shows the most frequent damage mechanism affecting plants operating in the cycling regime. According to this analysis, fatigue is the most damaging type of failure followed by wear and erosion in CCGT plants and corrosion-fatigue in conventional plants.

- Statistical analysis of annual O&M cost related to creep life consumption, measured in online hours, showed a very weak correlation between the data. Furthermore, because of the negative coefficient associated with the x^2 term of the data trendline, an inverted bathtub shape was found, implying that costs are highest during the mid-life period. This contradicts reason and experience, so for practical purposes, it appears that there is *no useful relationship between lifetime online hours and the annual non-fuel O&M cost for the units in the sample.*
- The statistical relationship between Equivalent Hot Starts (EHS) and annual maintenance costs showed a positive correlation between the data. Also, the positive coefficient for the x^2 term of the data trendline means that the upright bathtub curve is restored, conforming to the observation that O&M costs are usually higher in the early and late stages of the unit's service life. Based on analysis of the units in this sample, therefore, *it appears that the strongest indicator of annual O&M costs is the number of EHS that the unit has performed.*
- A relationship has been produced enabling annual O&M costs to be estimated based on the number of accumulated EHS; the relationship can be extrapolated for different sized units by multiplying the relevant capacity correction factor (formula provided in the report).
- A case study performed on a specific European system suggested that, in line with the expected life cycle of most generators originally designated as base load, *as units in this system age, they accumulate online hours at a decreasing rate, and accumulate starts at a higher rate.*
- The case study found (similar to the main study) that there is no real connection between lifetime online hours and the annual non-fuel O&M cost for base load designed units in this system. A relatively high level of correlation was however found between damage accumulation, as represented by annual non-fuel O&M cost, and fatigue-life consumption, represented by cumulative EHS, *which seemed to confirm the assumption that for our model of a typical base load designed generator in this system, fatigue is again the dominant indicator of annual maintenance costs.*
- The case study model was further developed to incorporate scenarios with additional starts due to factors such as high levels of renewable energy penetration above baseline (e.g. 10%, 20% or 30%). *This process provides a means of estimating the cost of a hot, warm or cold start in any year of a unit's service life within the system while accounting for varying demand for cycling in the future depending on market conditions.*
- The main analysis of this report is based on units in many different power systems/electricity markets that have no strong correlation in operational history. For more accurate / in-depth results it is recommended to perform specific analysis at the system level e.g. for ERCOT, PJM, MISO etc. territories, and would require further data from generators within the system under analysis (as demonstrated by the results obtained in the provided case study).
- In order to minimize the economic impact due to cycling, it is crucial to create a maintenance plan with a well-balanced proactive maintenance, predictive maintenance and preventive maintenance through the plant life cycle. There are many excellent maintenance tools now available in the market and (if not already implemented) it would be extremely beneficial to conventional and CCGT plants to consider their implementation to achieve a well-balanced maintenance plan going forward.

10
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A
PLANT DATABASE

The cost analysis presented in this study was performed based on a total of 30 conventional and 65 CCGT plants from ETD original database. This database includes O&M (performance and cost) data from a number of worldwide plants (Europe, USA, Asia etc.) collected by individual plant surveys and from earlier studies. Plants age ranges from 1 to 30 years for CCGT and 1 to 40 years for conventional plants. Capacity is ranges from 170 to 480 MW for conventional and 110 to 492 MW for CCGT plants. A detail of the plant database is presented in Table A-1.

ETD database has been divided into several categories such as age, equivalent hot starts (EHS), online hours etc. and then each category is divided by several groups (i.e. EFOF v Age, EPOF v Age etc.) (see Table A-2) and then the population mean value was determined for each individual group.

Table A-1
Details of ETD Plant database

	Conventional Plant	CCGT Plant
Regions	North America – 14 Europe – 14 Asia – 2	North America – 25 Europe – 35 Asia – 5
Capacity	170 – 480 MW	110 – 492 MW
Age	1 – 40 years	1 – 30 years
Online hours	- Range: 50,380 – 28,6874 - Lifetime online hours	- Range: 3,635 – 217,432 - Lifetime online hours
Equivalent Hot Start (EHS)	- Range: 500 – 4000 - Calculated from the lifetime hot, warm and cold starts	- Range: 40 – 4266 - Calculated from the lifetime hot, warm and cold starts
Equivalent Forced Outage Factor (EFOF)	- Calculated from the forced outage hours collected for last 3 to 10 years of operation - Average EFOF represents the average of EFOF values calculated for last 3 to 10 years of operation	
Equivalent Planned Outage Factor (EPOF)	- Calculated from the planned outage hours collected for last 3 to 10 years of operation - Average EFOF represents the average of EFOF values calculated for last 3 to 10 years of operation	

Table A-1 (continued)
Details of ETD Plant database

<table>
<tr><th></th><th>Conventional Plant</th><th>CCGT Plant</th></tr>
<tr><td>Operation & Maintenance (O&M) Cost</td><td colspan="2">- Calculated from the non-fuel O&M costs data collected for last 3 to 10 years of operation.
- The annualized non-fuel O&M cost includes the costs of maintenance and repairs, chemistry, modifications/capital expenditure (for example, in plant modifications to enhance the unit’s cycling capability), increased frequency of inspection and other operating costs.
- This non-fuel O&M costs does not include costs related to LTSA (Long Term Service Agreement).</td></tr>
</table>

Table A-2
Description of plant database divided into categories and groups

Plant Database	Categories	Groups
Conventional	Age (years)	EFOF v Age for base load plants
		EFOF v Age for cyclic plants (considered all types of cycling regimes)
		EFOF v Age for cyclic plants operating in the various cycling regimes
		EPOF v Age for cyclic plants
		EPOF v Age for base load plants
	EHS	EFOF v EHS for cyclic plants
		EFOF v Number of Starts for hot, warm & cold starts for cyclic plants
		EPOF v EHS for cyclic plants
		EPOF v Number of starts for hot, warm & cold starts for cyclic plants
		EHS v Annualized O&M cost
	Online hours	EFOF v Online hours for cyclic plants
		EPOF v Online hours for cyclic plants
		O&M cost v Online hours

Table A-2 (continued)
Description of plant database divided into categories and groups

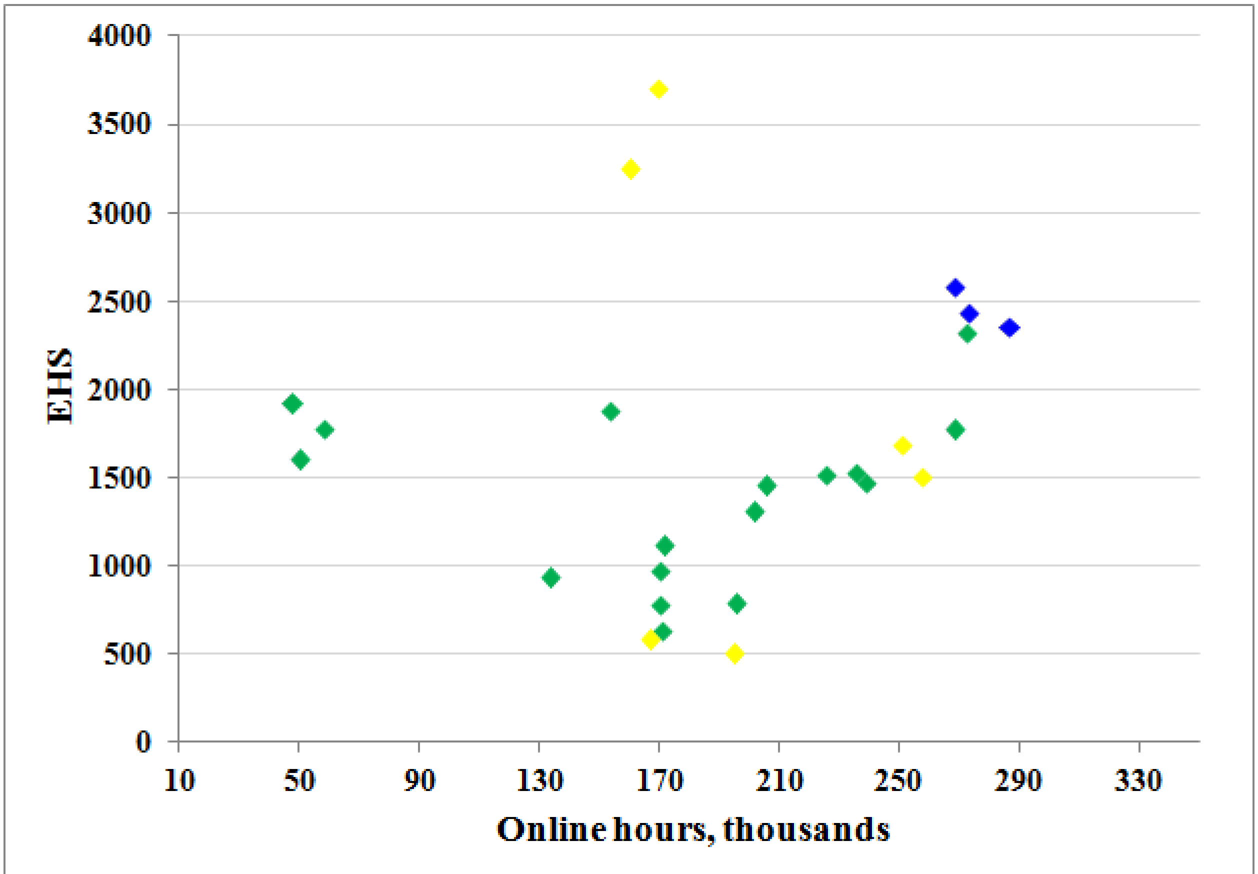
Plant Database	Categories	Groups
CCGT	Age (years)	EFOF v Age for base load plants
		EFOF v Age for cyclic plants (considered all types of cycling regimes)
		EFOF v Age for cyclic plants operating in the various cycling regimes
		EPOF v Age for cyclic plants
		EPOF v Age for base load plants
	EHS	EFOF v EHS for cyclic plants
		EFOF v Number of starts for hot, warm & cold starts for cyclic plants
		EPOF v EHS for cyclic plants
		EPOF v Number of starts/Year for hot, warm & cold starts for cyclic plants
		O&M cost v EHS
	Online hours	EFOF v Online hours for cyclic plants
		EPOF v Online hours for cyclic plants
		O&M cost v Online hours

Performance Data

Equivalent Forced Outage Factor (EFOF) and Equivalent Planned Outage Factor (EPOF) data were collected to assess the EFOF & EPOF performance for the entire life cycle of the plant. Impact of cycling on these outages was determined by plotting the graph for EFOF and EPOF against the lifetime Equivalent hot starts (EHS). Figures A-1 to A-4 show the accumulated lifetime EHS & average EHS v lifetime online hours for the entire database of conventional and CCGT plants, respectively. Color code is used to identify the plants operating under different cycling regimes. ***Note that various cycling regimes were identified from the annual EHS based on the definition provided in Section 4.***

From the analysis, *three distinct areas (Commissioning period, Useful life period & major component wear out period)* are identified for the entire life cycle of a power plant operating either in base load or cycling regime. These three areas characterize the different forced outage and planned outage levels, for the entire life cycle of the plant as follows. The following figures show the entire database for EFOF & EPOF for these three different areas for the plants operating under cycling regime; Figures A-5 & A-6 for conventional plants and Figures A-7 & A-8 for CCGT plants.

<u>Note:</u> Data for EFOF and EPOF were collected for the last 3 to 10 years of operation for each plant in the ETD database. Therefore, average EFOF & EPOF represents the 'data collection period' only. Data for EHS was calculated from the lifetime hot, warm & cold starts and the data for online hours represents the lifetime service hours.



Note for color code:
Sporadic - Green | Weekend shutdown - Blue | Load follow - Yellow
Two shifting - No data | Double Two shifting - No data

Figure A-1
Lifetime EHS v lifetime online hours for conventional plants

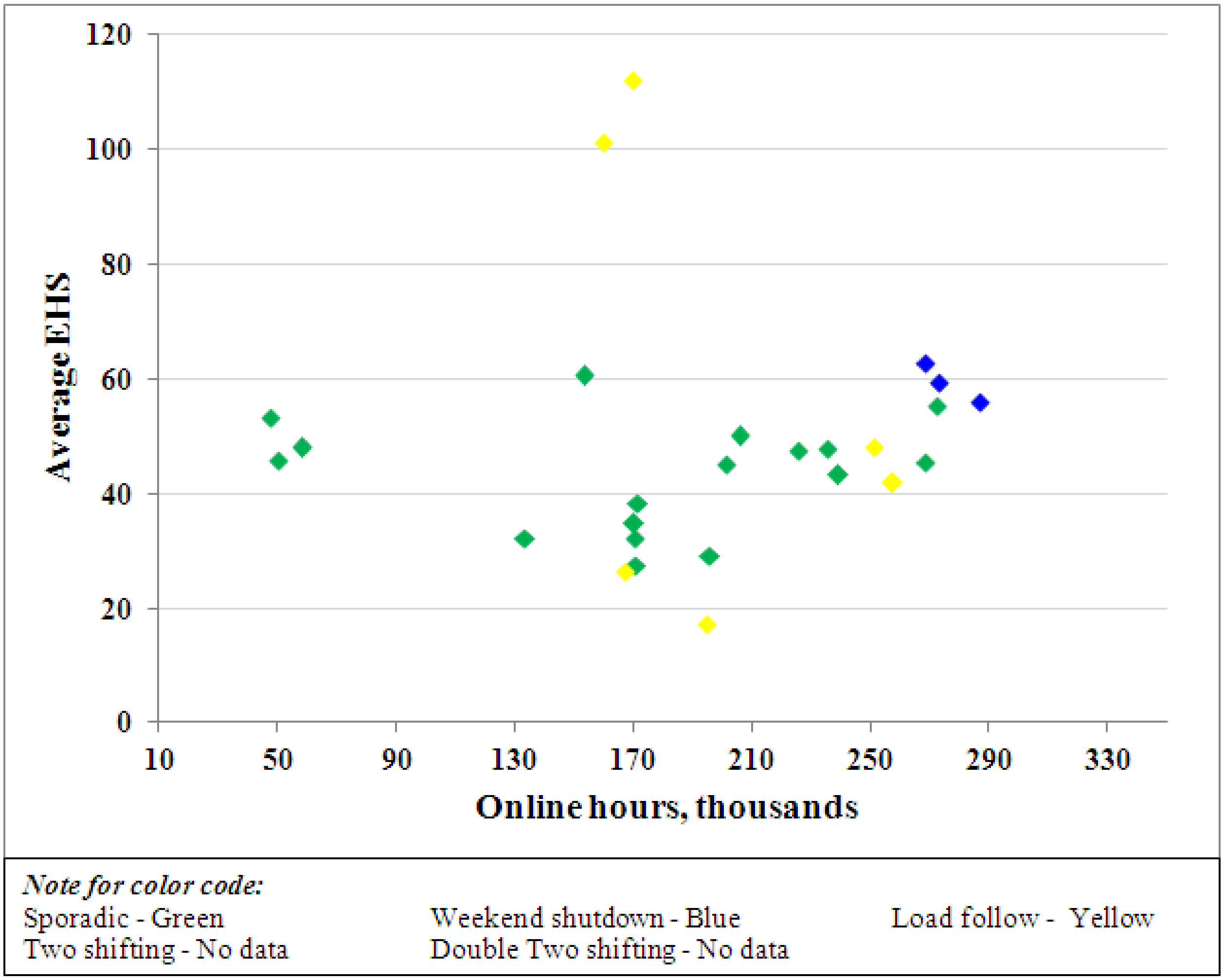


Figure A-2
Average EHS v lifetime online hours for conventional plants

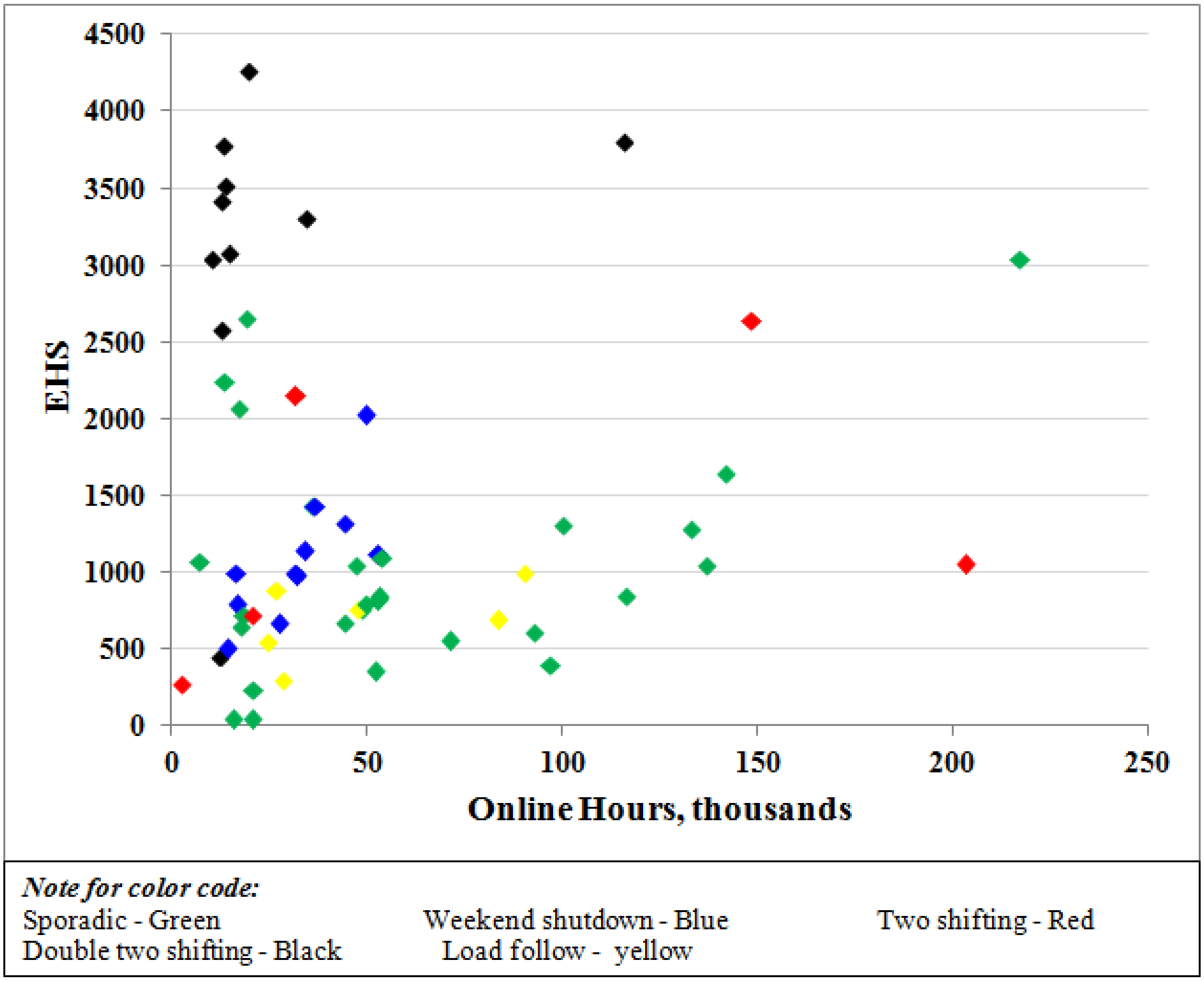
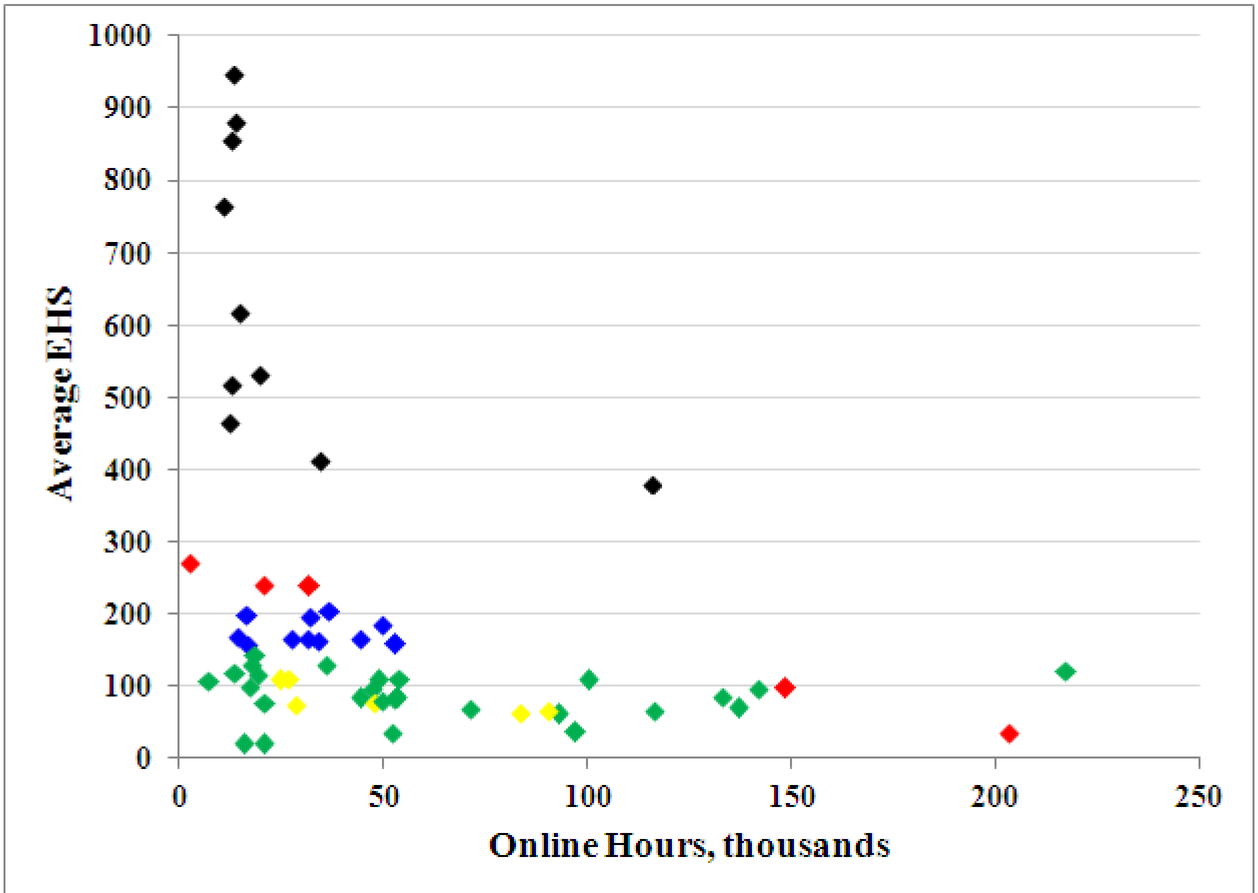


Figure A-3
Lifetime EHS v lifetime online hours for CCGT plants



Note for color code:
Sporadic - Green | Weekend shutdown - Blue | Two shifting - Red
Double two shifting - Black | Load follow - yellow

Figure A-4
Average EHS v lifetime online hours for CCGT plants

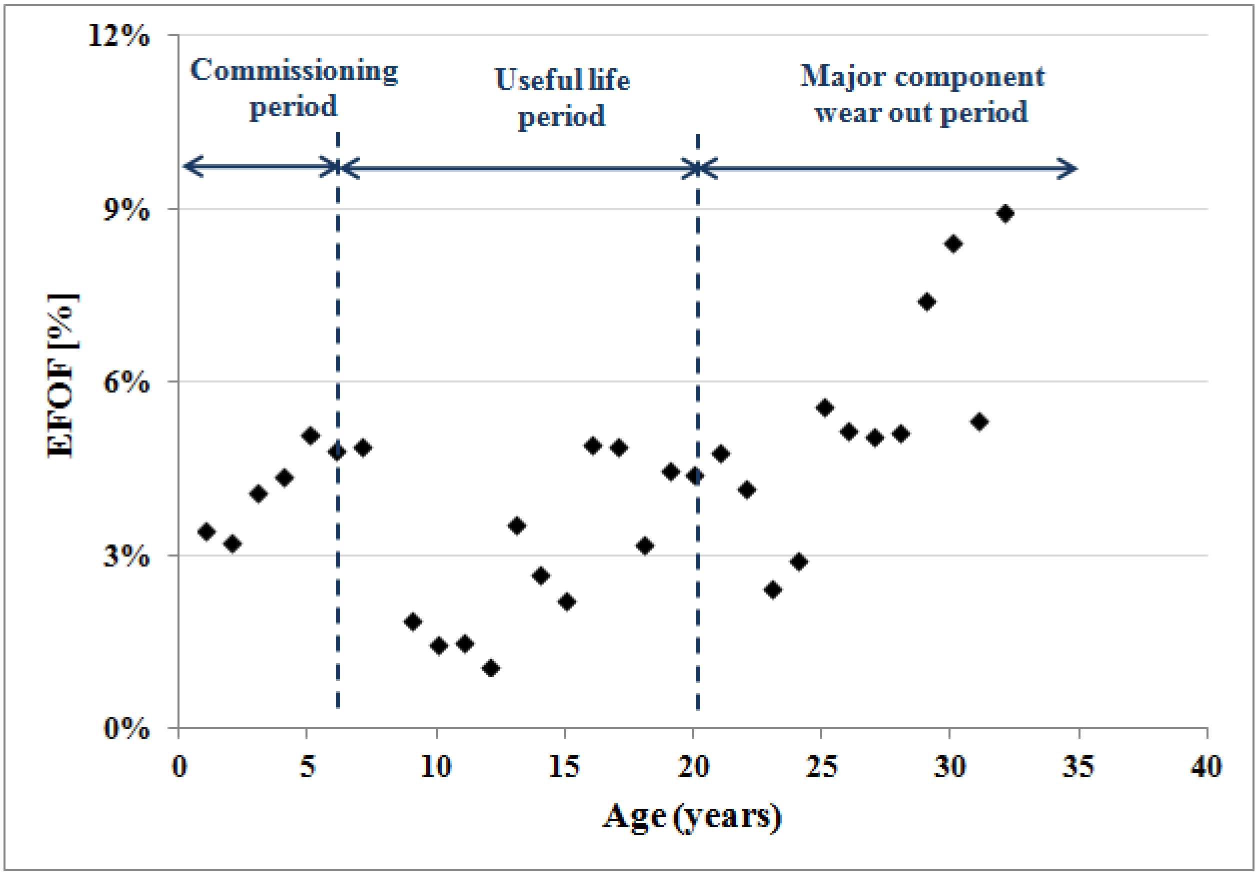


Figure A-5
Average EFOF v Age shows three distinct areas for the entire life cycle of conventional plants

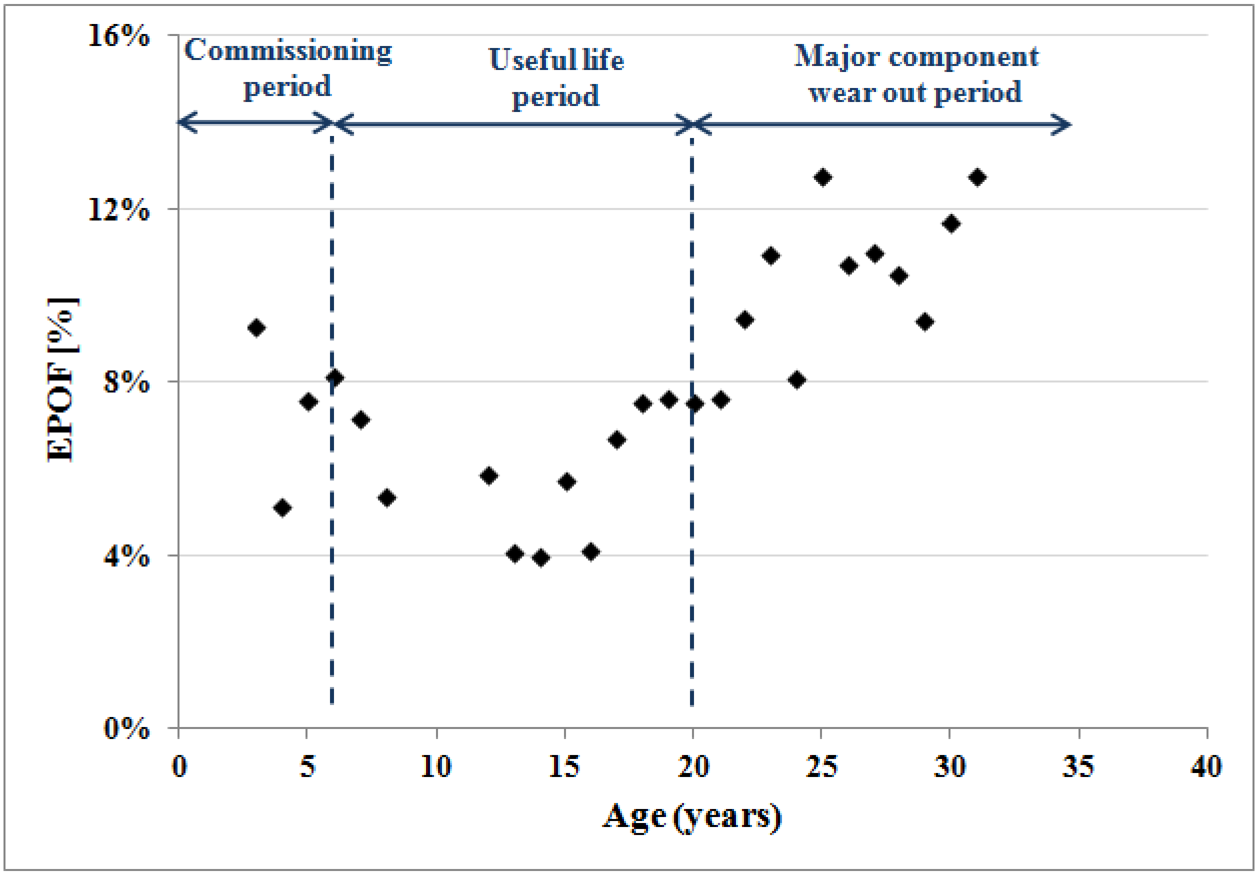


Figure A-6
Average EPOF v Age shows three different areas for the entire life cycle of conventional plants

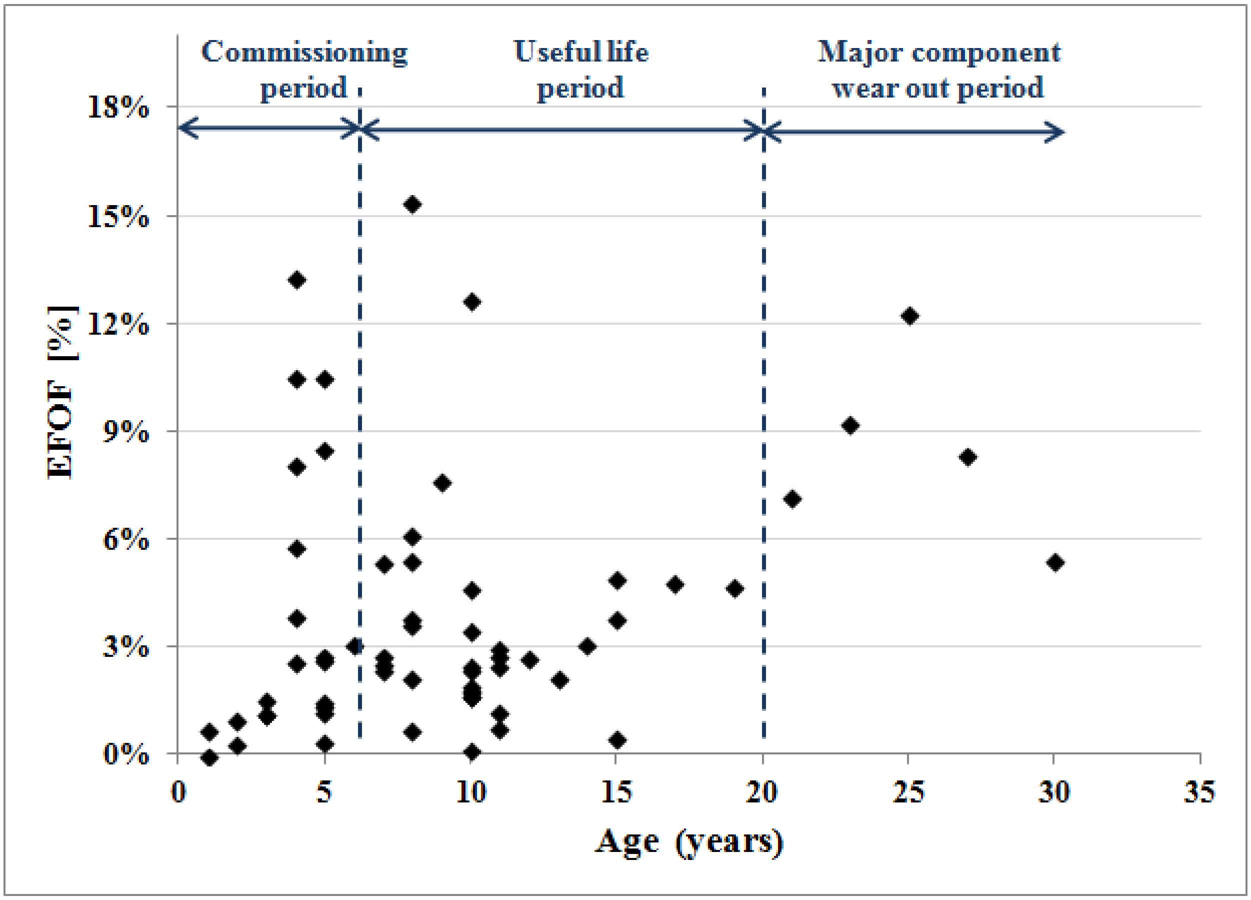


Figure A-7
Average EFOF v Age shows three different areas for the entire life cycle of CCGT plants

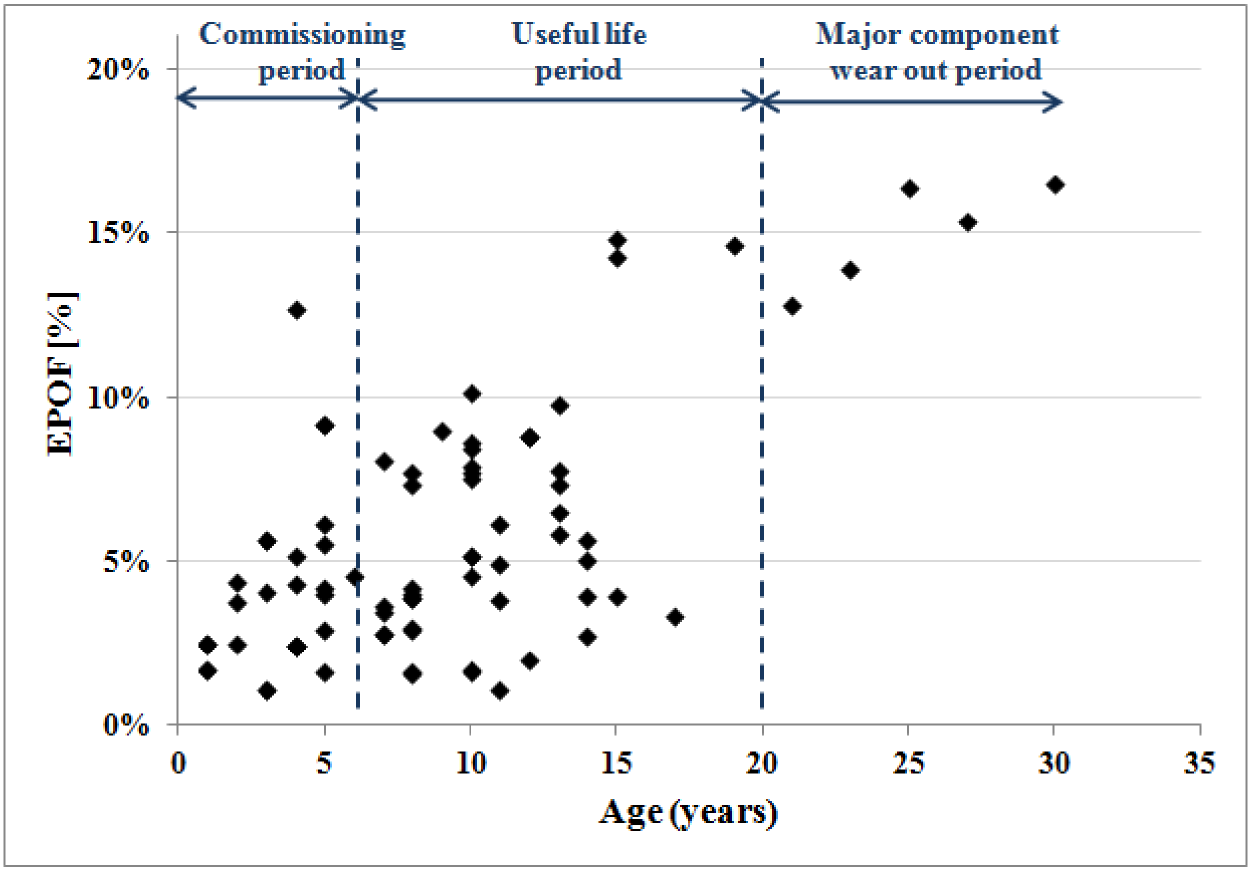


Figure A-8
Average EPOF v Age shows three different areas for the entire life cycle of CCGT plants

Online Hours

From the analysis, it is noted that the relationship for average EFOF & EPOF and lifetime online hours shows poor correlation of data for conventional and CCGT plants. Therefore, for these relationships no assessment has been made in the main report. The following figures show the correlation for 'average EFOF v lifetime Online hours' and 'average EPOF v lifetime Online hours'; Figures A-9 & A-10 for conventional plants and Figures A-11 & A-12 for CCGT plants.

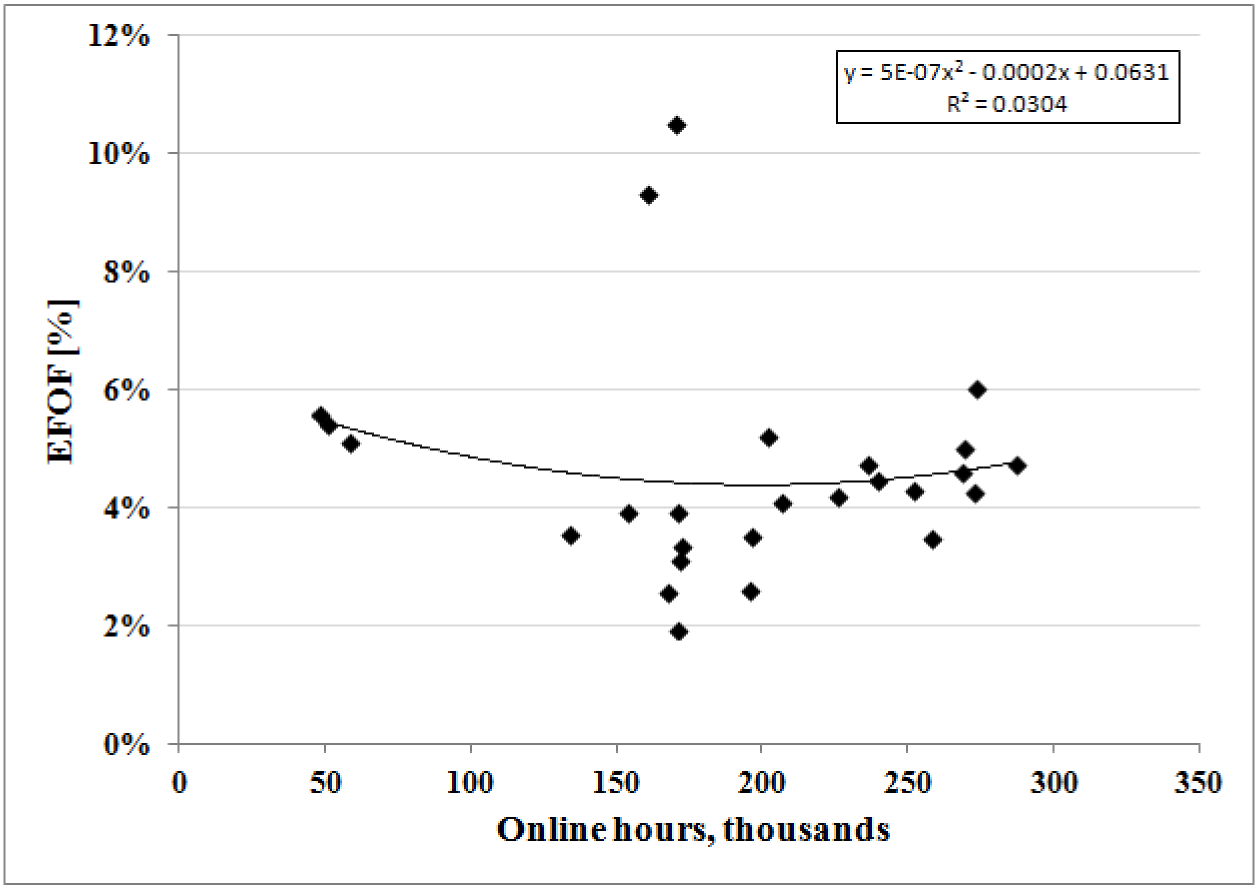


Figure A-9
Average EFOF lifetime online hours for conventional plants

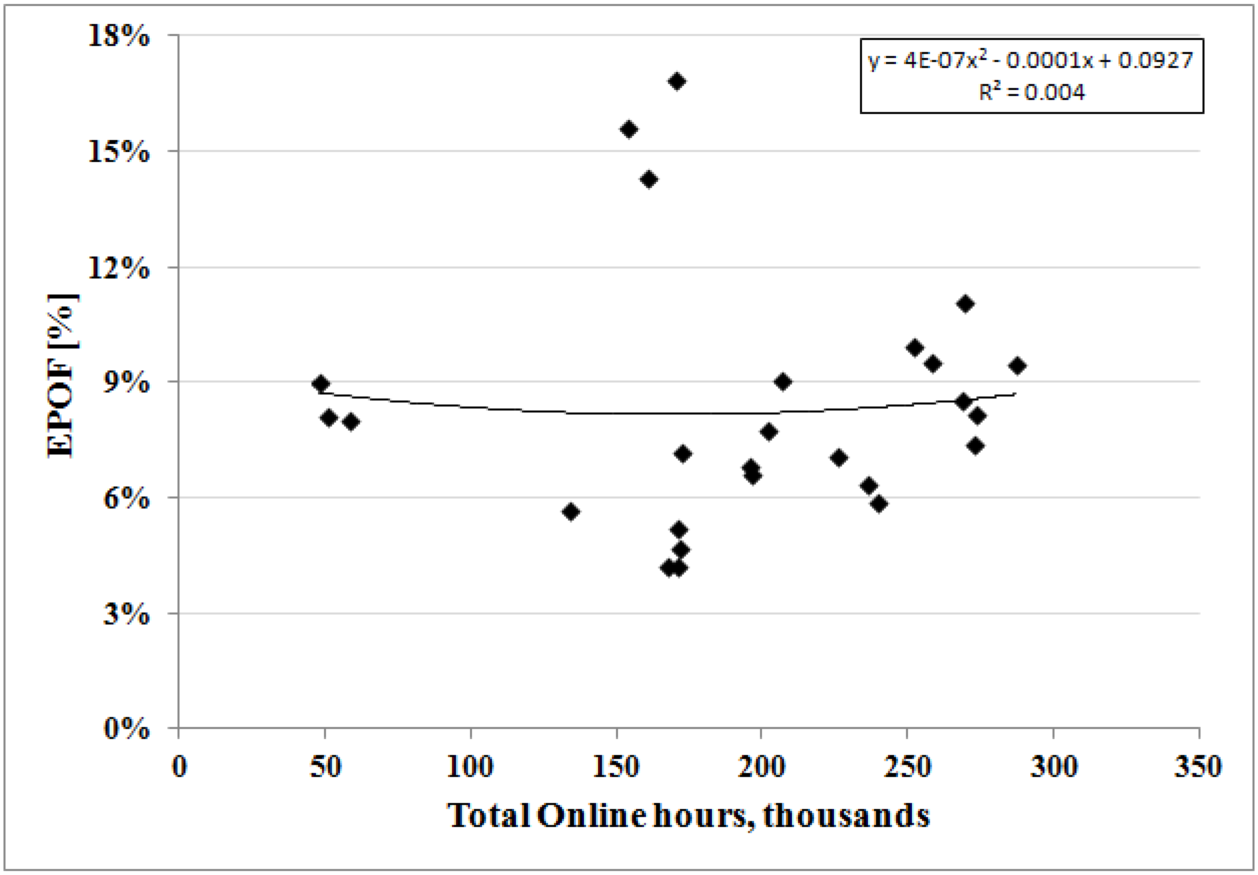


Figure A-10
Average EPOF v lifetime online hours for conventional plants

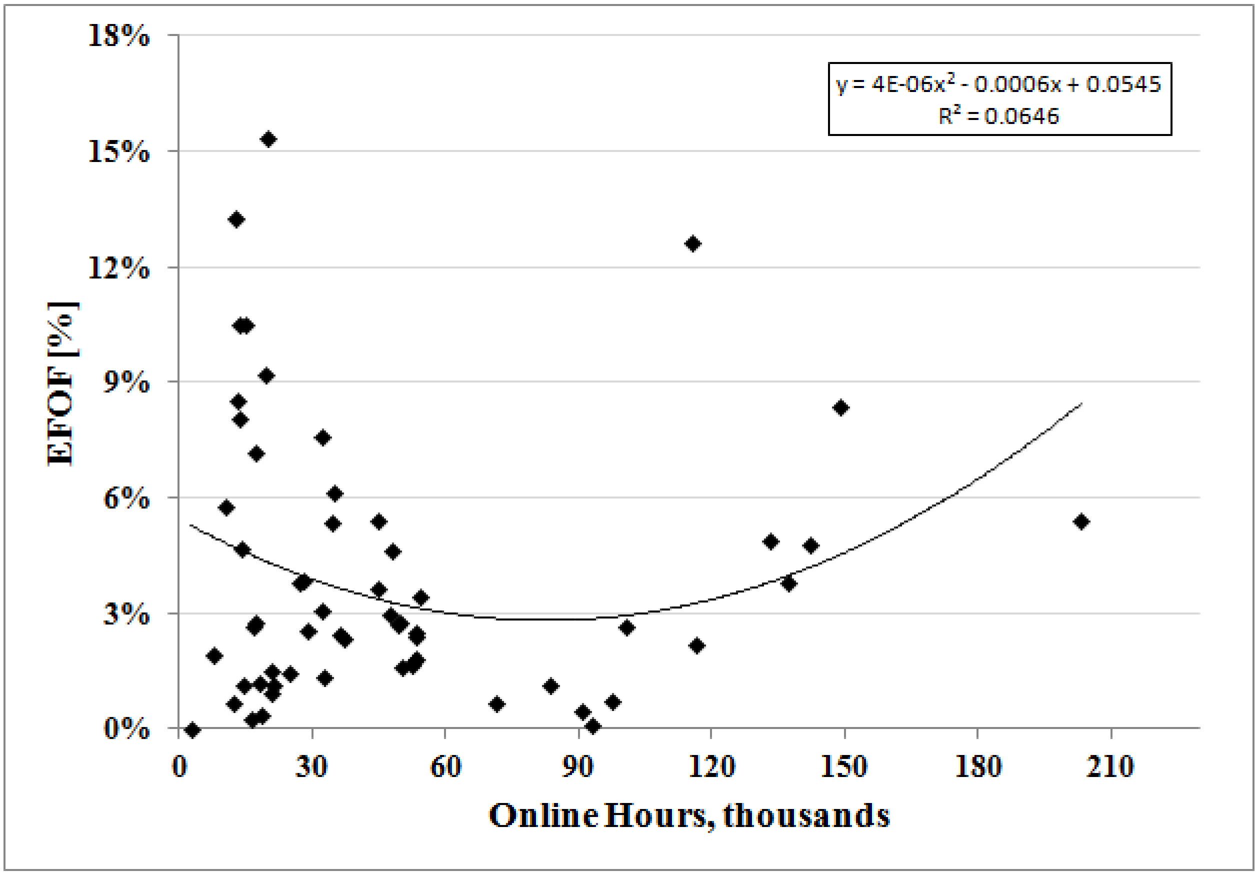


Figure A-11
Average EFOF v lifetime online hours for CCGT plants

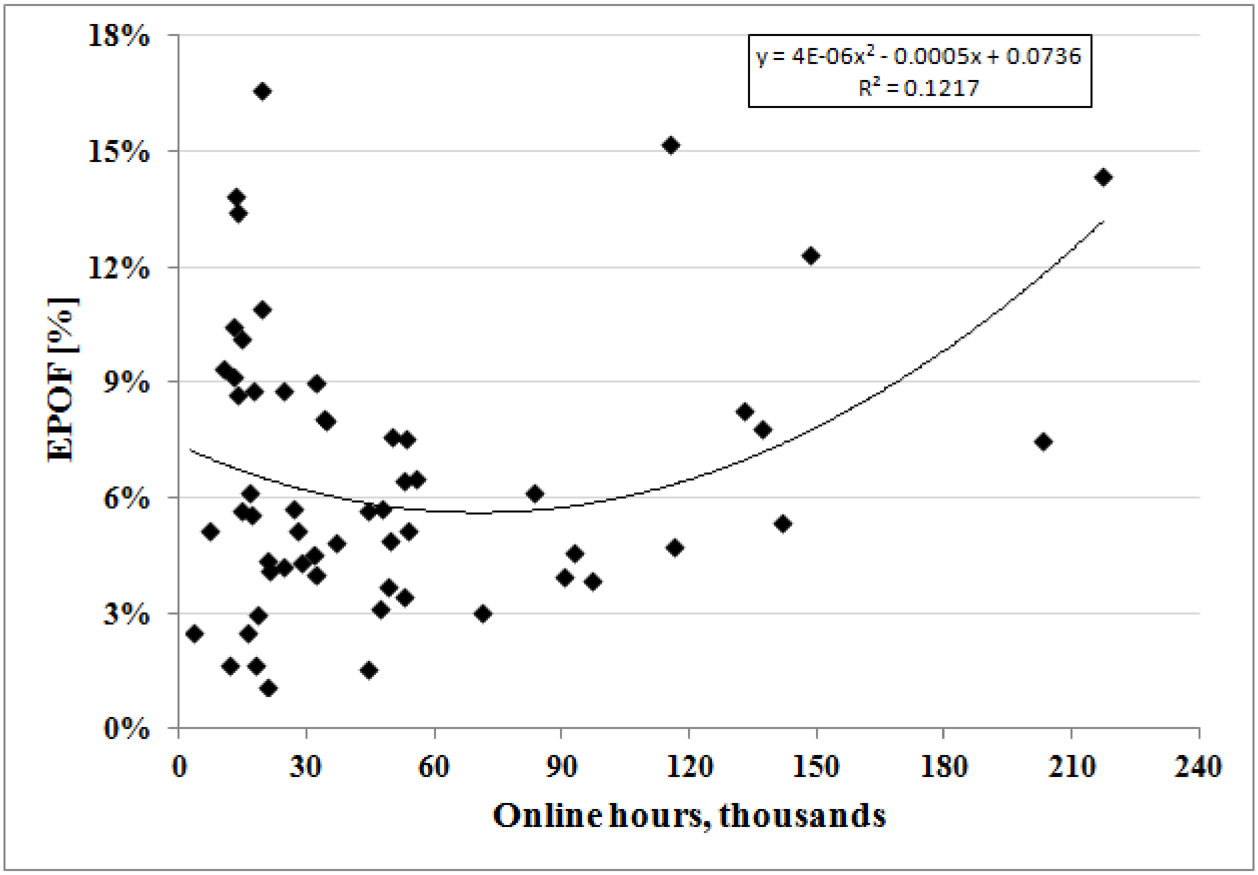
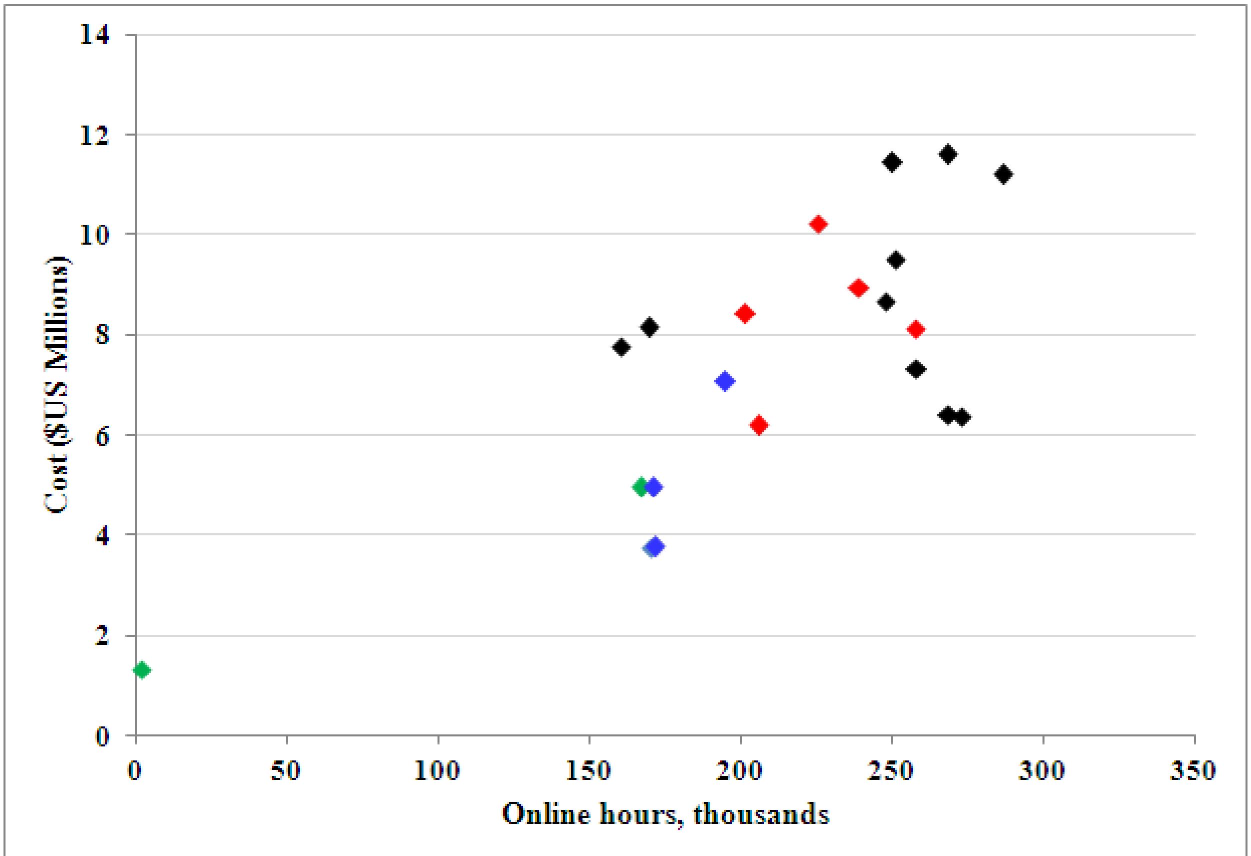


Figure A-12
Average EPOF v lifetime online hours for CCGT plants

Cost Data

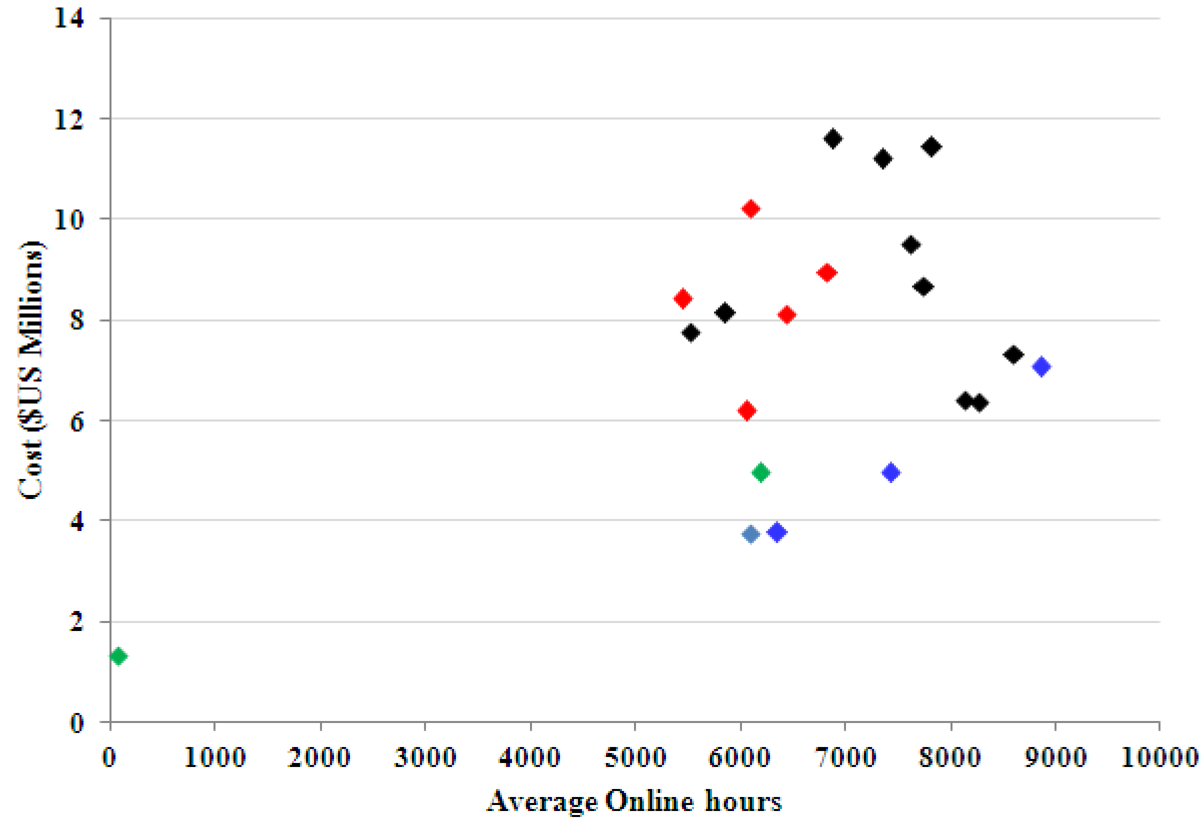
The cost analysis undertaken in this study is based on a top-down statistical analysis. ***The costs data considered in this study are not the cost of cycling but are the annual non-fuel operation and maintenance (O&M) cost.*** The cost analysis takes into account reported expenditure for each unit over a number of years to determine an annualized non-fuel O&M cost. During the survey costs data were provided in different ways (some are more and some are less detailed) by different utilities; only the general cost variables for the O&M costs were considered for this study. Therefore, the annualized non-fuel O&M cost is calculated by summing the costs of maintenance and repairs, chemistry, modifications/capital expenditure (for example, in plant modifications to enhance the unit's cycling capability), increased frequency of inspection and other operating costs. This non-fuel O&M costs does not include costs related to LTSA (Long Term Service Agreement). Fuel costs are not included in the cost model because this is a market specific factor and can change at any time. Lost generation cost due to forced and planned outages is also not included in the model as this depends on the current electricity price which is also a market specific factor. Following figures show the color coding used for each cycling regime; Figures A-13 to A-16 for conventional plants and Figures A-17 and A-20 for CCGT plants.

Note: Data for O&M costs were collected for the last 3 to 10 years of operation for each plant. Data for EHS was calculated from the lifetime hot, warm and cold starts and the data for online hours represents the lifetime service hours.



Notation for color code:
Sporadic - Green
Weekend shutdown - Blue
Two shifting - Red
Double two shifting - Black
Load follow - No data

Figure A-13
Annualized O&M costs v lifetime online hours for conventional plants



Notation for color code:
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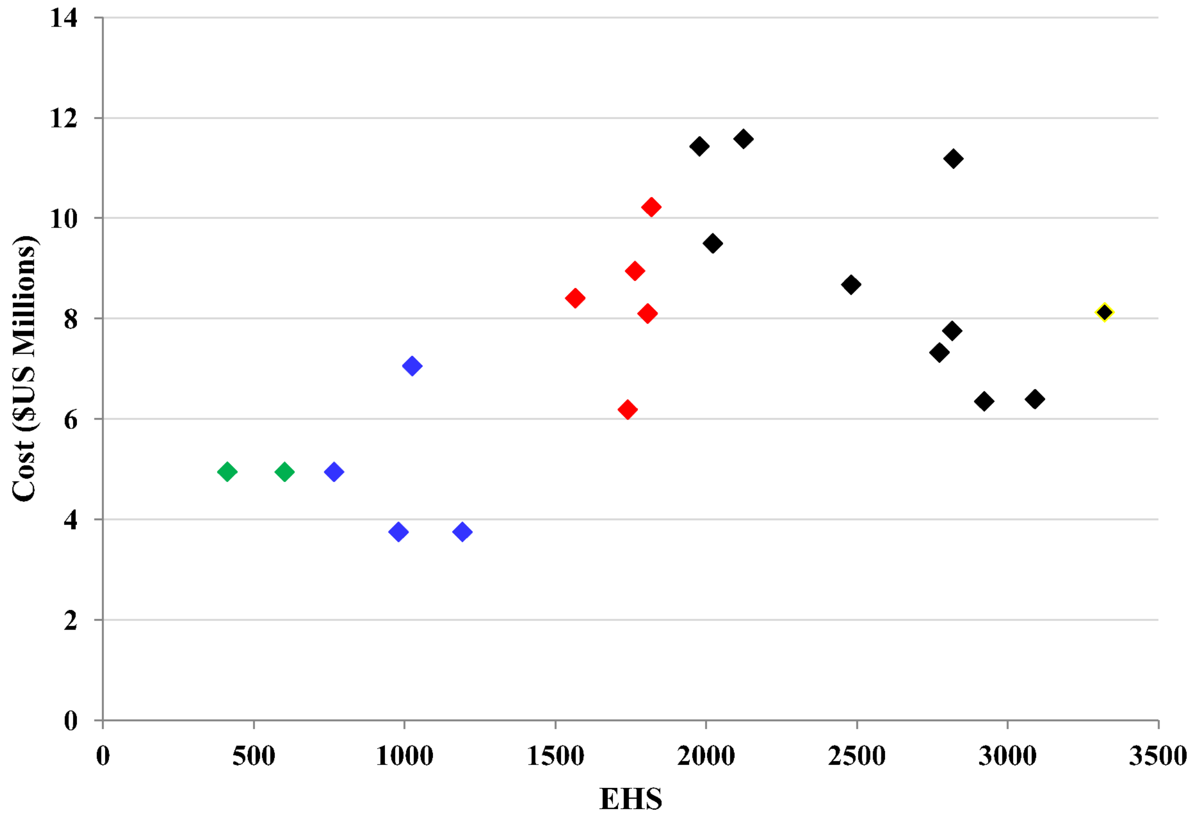
Double two shifting - Black Load follow - No data

Figure A-14
Annualized O&M costs v Average online hours for conventional plants



Notation for color code:
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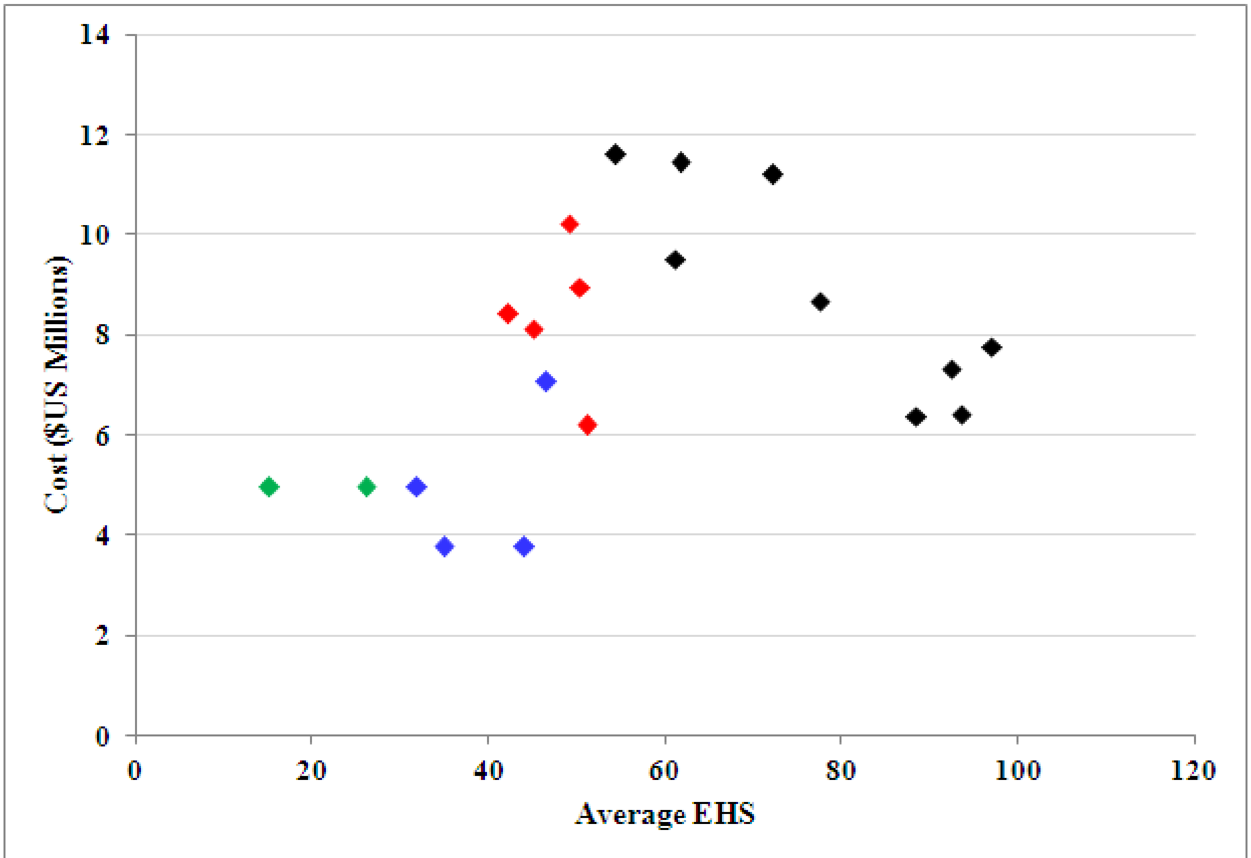
Weekend shutdown - Blue
Two shifting - Red
Double two shifting - Black
Load follow - No data

Figure A-15
Annualized O&M costs v lifetime EHS for conventional plants



Notation for color code:

Sporadic - Green
Weekend shutdown - Blue
Two shifting - Red
Double two shifting - Black
Load follow - No data

Figure A-16
Annualized O&M costs v Average EHS for conventional plants

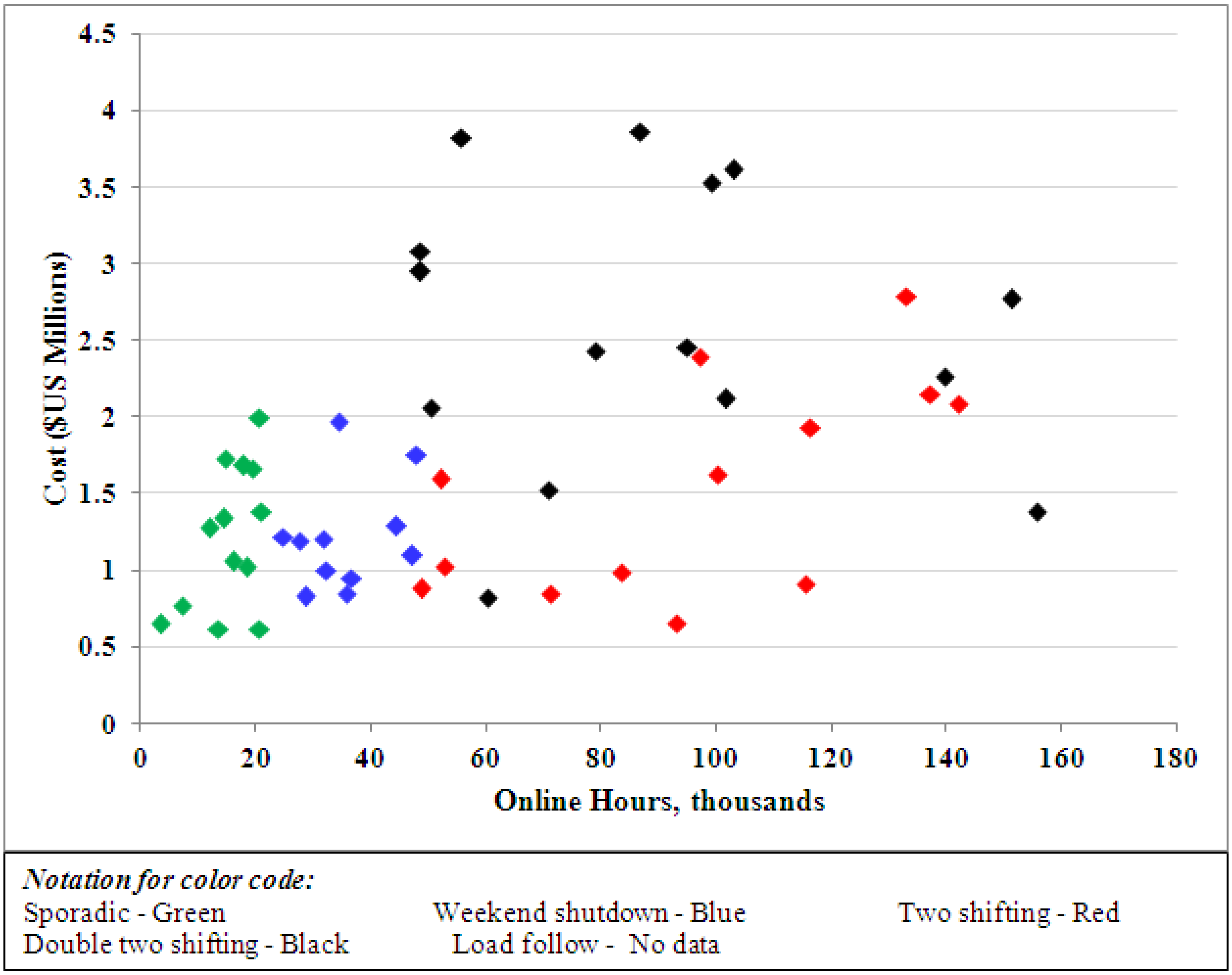


Figure A-17
Annualized O&M costs v lifetime online hours for CCGT plants

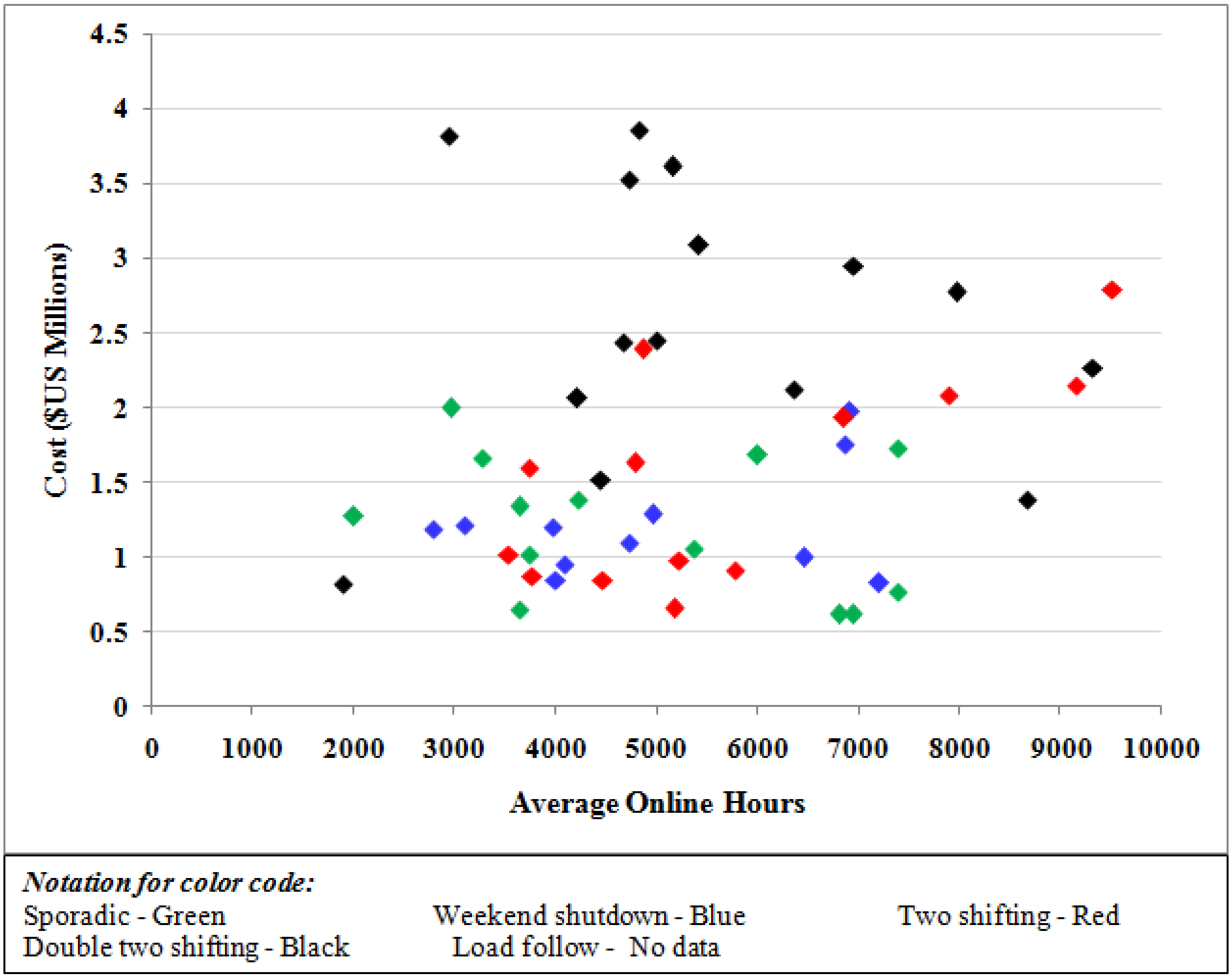


Figure A-18
Annualized O&M costs v Average online hours for CCGT plants

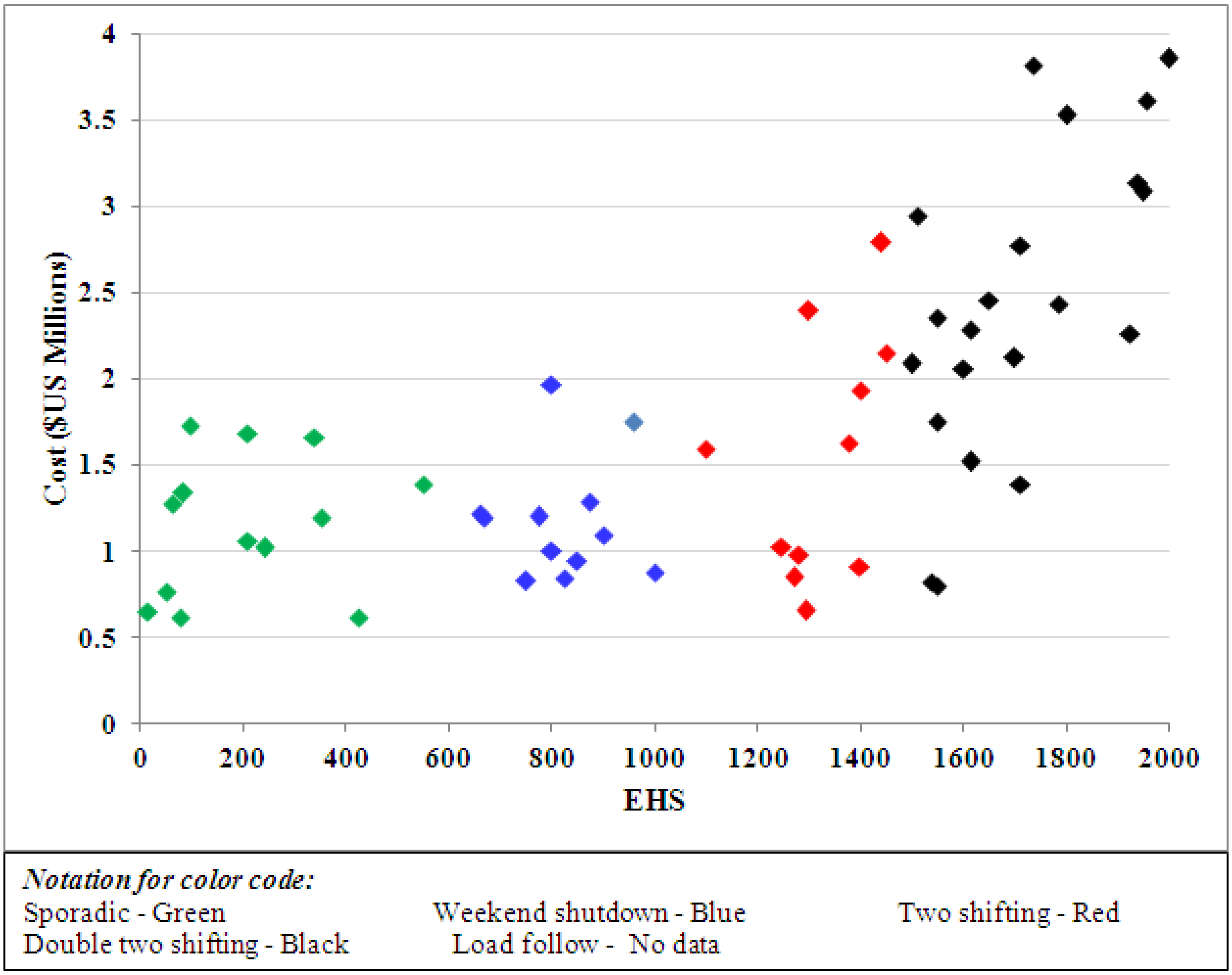
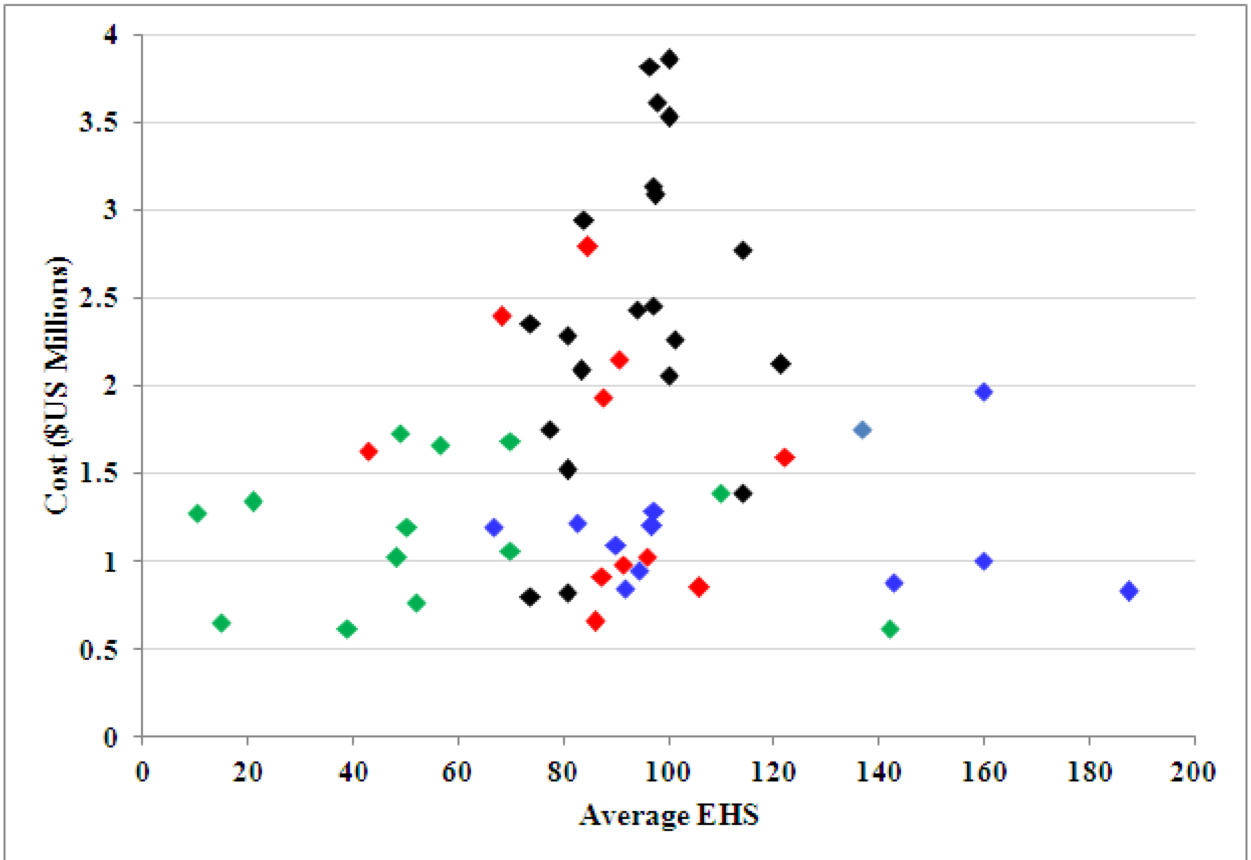


Figure A-19
Annualized O&M costs v accumulated EHS for CCGT plants



Notation for color code:
Sporadic - Green | Weekend shutdown - Blue | Two shifting - Red
Double two shifting - Black | Load follow - No data

Figure A-20
Annualized O&M costs v Average EHS for CCGT plants

B

DAMAGE MECHANISMS FOR CONVENTIONAL POWER PLANTS ASSOCIATED WITH CYCLIC OPERATION

Understanding of specific damage/failure mechanisms that affect various components of the conventional power plants and implementation of appropriate engineering solutions is crucial to avoid component failures. The typical damage mechanisms associated with cyclic operation for various components of conventional plants are discussed in this Section.

B.1 High Temperature Metallurgical Issues

Older base-load fossil units were designed, almost by definition, to operate predominantly under creep conditions. None of the older design codes for power plant (ASME, BS, DIN) placed any specific requirement on the designer to consider fatigue as a failure mechanism. The design codes merely made an implicit assumption that the effects of fatigue were contained within the conservatism of the design stresses. This was an adequate assumption for base load plant, but it is now recognized that fatigue, especially in conjunction with creep degraded material, is a significant concern for cyclic operation.

B.1.1 Creep and Thermal Fatigue

In principle, since creep is both time and temperature dependent, cyclic operation such as two-shifting and low load operation would be expected to reduce damage due to long term creep. During a unit start, or at periods of low load running, there may be some circumstances when localized overheating can occur. Whilst these problems are well known to plant operators, it should be recognized that the cumulative effects of repeated overheating during thermal cycling and load cycling can give rise to extended periods of operation above the design temperature which may result in accumulation of creep damage or its acceleration.

A creep related phenomena in cyclic units is degradation in the microstructure and the concomitant reduction in material properties which has already occurred during the previous term of base load operation. These microstructural changes will have occurred simply as a result of exposure to temperature, but they will have been accelerated by the presence of stress. The most obvious signs of such degradation are the onset of spheroidization in carbon- manganese and 2.25Cr1Mo steels. The implications of this are likely to be reduced material ductility, compared to virgin material and reduced resistance to creep and/or fatigue cracking.

However, by far the most common problem experienced as a result of cyclic operation is thermal fatigue damage. This can manifest either in the form of cracking of an individual component or by the mechanical failure of structures.

Cracking of a component is attributed to severe thermal gradients arising from excessive steam to metal and through wall temperature differences associated with rapid rates of change of steam temperatures as generally observed during start up, shut down and load changes. The principal components at risk typically comprise any thick walled sections such as boiler superheater headers, steam pipework, valves, HP and IP steam chests and turbine inlet belts. HP heaters and economizer inlet headers are also frequently exposed to similar effects due to rapid cooling by cold feed water.

Thin walled sections such as boiler tubes, reheater headers etc. are less prone to the problem. These thinner walled sections can, however, suffer damage due to accumulation of water inside giving rise to large temperature differentials or quenching. This is can be caused by inadequate drainage flow or incorrect supporting of components resulting in low spots which cause water to accumulate. Alternatively passing valves (reheater de-superheater spray valves, for example) which are barely noticeable on load can inject water into hot components causing rapid temperature reductions (when a feed pump is started, perhaps). On a wider scale, structures such as boiler framework and tube attachments, boiler supports and pipe work support systems are also vulnerable to thermal cycling.

B.1.2 Creep-Fatigue Interaction

Materials behave in a complex way when both creep and fatigue mechanisms are present. They usually act synergistically to cause premature failure. Creep strains can reduce fatigue life and fatigue strains can reduce creep life. The American Society of Mechanical Engineers (ASME) recognized the effects of interaction and provides guidance on the interaction between creep and fatigue and its effect on the life expectancies of materials. (*See ASME Cases of ASME Boiler and Pressure Vessel Code, Case N-47*).

The creep-fatigue interactions are not currently well defined, and the limit line as shown in green in Figure B-1 represents the design life limit, expressed in fraction of material creep life and fatigue life for a 2.25Cr1Mo steel. This limit line is used to establish the effect of combining the two mechanisms and demonstrates how they act together to reduce the effect of the individual processes. Original design criteria assumed that the two processes were entirely independent. The line is a highly conservative representation of the phenomenon. It does however serve to demonstrate the effects of the interaction.

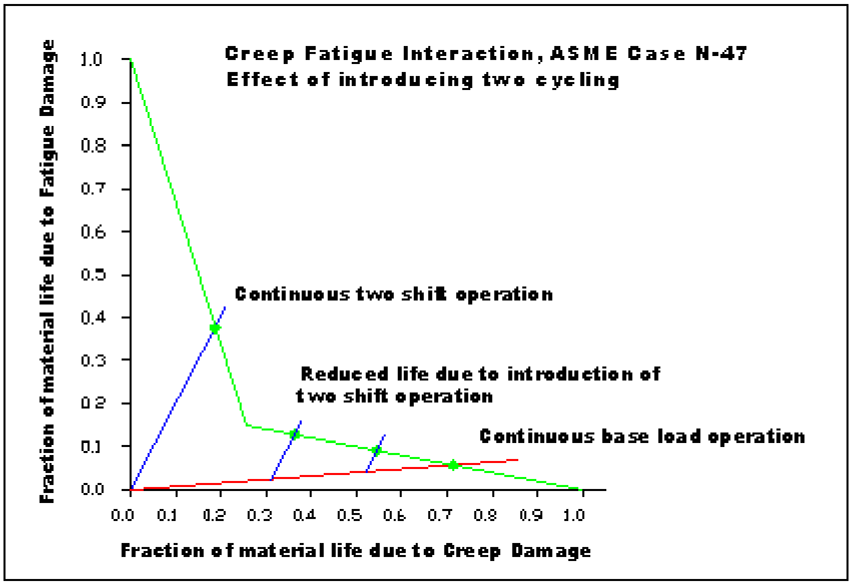


Figure B-1
Demonstrates the interaction and consequences of creep and fatigue (based on ASME N-47) for a typical power plant steel (2.25Cr1Mo)

By way of an example, consider a component originally designed for say 10,000 cycles, which might have been designed to operate in a unit which two-shifts on a daily basis over 30 years. Assume also that the component operates in the creep range and was designed for 150,000 hours operation. If the unit were to operate on a base load regime, it will of necessity, accrue some thermal cycles, probably of the order of 1000 over the projected life. The red line indicates the effective operation of the component. The actual component life is given by the point where it intersects the green line. This shows a reduction of the component life to about 75% of its predicted creep life. Similarly, if the component operates on a two-shifting unit with 300 cycles per year whilst operating in the creep range, the actual life may be as low as 40% of the anticipated fatigue life. Where operational cycling is introduced on a former base load unit, it can be seen that the residual life can be greatly reduced to between 40% and 60% of the original design life due to the combined effects of creep and fatigue.

The key implication is that older units designed for base-load operation and used in this capacity over many years are very susceptible to component failure when they are eventually forced to cycle regularly.

Thus, while increases in failure rates due to cycling may not be noted immediately, critical components will eventually start to fail. Shorter component life expectancies will result in higher plant EFOR (equivalent forced outage rate), longer scheduled outages, and/ or higher capital and maintenance costs to replace components at or near the end of their service lives. In addition, it may result in reduced total plant life or more capital to extend the life of the plant.

The above example, although only figurative, demonstrates how two-shifting plant originally designed for base load conditions can significantly reduce the integrity of components to a far greater degree than might otherwise have been anticipated. It should be noted that this methodology is highly conservative and does not take account of the timing or amplitude of the thermal cycles which may further influence the remnant life of components. However, in a recent review Gandi and Skelton showed from the metallographic examples of service-exposed components and laboratory-tested samples that a creep fatigue interaction may not necessarily take place at elevated temperatures. They argued that the deciding factor may lie in the strain rate of shut- down or start-up procedures of a power plant. They showed that even when creep damage has accumulated in the bulk of the material, exposure to a subsequent cycle at high strain rate tends to ignore the intergranular creep damage sites, favoring instead a transgranular path. Gandy and Skelton thus argued that several scenarios are possible:

- Plant which is cycled continuously would not undergo creep-fatigue interaction, components rather being prone to thermal fatigue cracking.
- Long periods of base load operation followed by relatively fast ramp rates during subsequent shut-down and start-up procedures would not necessarily induce a creep-fatigue interaction, mixed-mode cracking being more probable. This scenario will encourage thermal fatigue cracking.
- *However, long periods of base load operation followed by slow ramp rates during subsequent shut-down and start-up procedures provides the maximum opportunity for the damaging creep-fatigue interaction to prevail.*

Thus plant operators need to decide between fast ramp rates resulting in thermal fatigue cracking or slow rates resulting in creep-fatigue interaction. Gandy and Skelton have argued that high strain rates are insufficient to cause intergranular cracking and often induce transgranular cracking despite the presence of creep damage. Figure B-2 describes these two mechanisms observed in a microstructure of a failed 316L specimen showing carbides on grain boundaries after cyclic loading at elevated temperature. It shows the crack has initiated by fatigue but later propagated predominantly by creep crack growth.

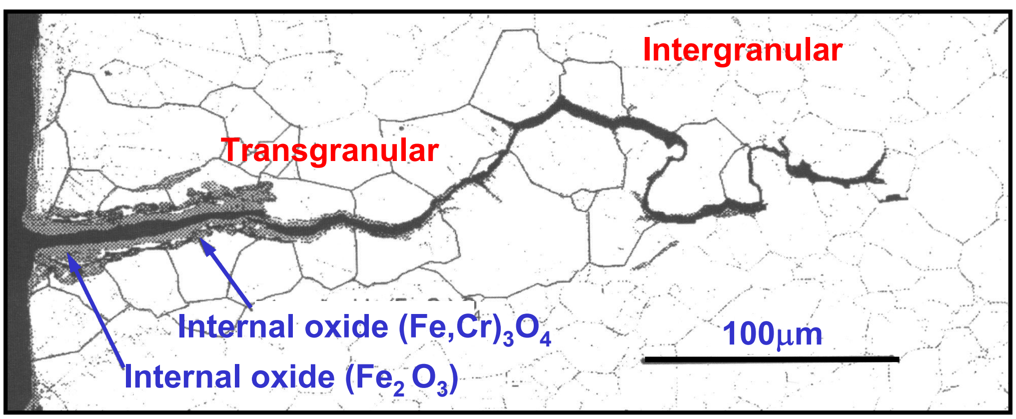


Figure B-2
Microstructure of failed 316L specimen showing carbides at grain boundaries after cyclic loading at elevated temperature

The reason is that creep damage requires time to build up within the material, whereas at early stages in life, corresponding to initiation, the grain boundaries have not had sufficient time to deform, hence cavitated. The grain boundaries in this case are strong therefore crack initiation is dominated by fatigue. After sometimes, the grain boundaries become progressively weaker and creep becomes the dominant failure mechanism in crack propagation. In other cases, we could expect a mixed mode (inter/transgranular) crack growth proceeding.

B.1.3 Equipment Failures Resulting from Creep-Fatigue Interaction

Major equipment failures resulting from creep-fatigue interaction are summarized below.

B.1.3.1 Cracking of Thick Wall Components

All thick sectioned components, such as boiler and turbine stop valves, governor valves, loop pipes and HP turbine inlet belts are prone to thermal fatigue cracking, due to through wall temperature differences during start up and shut down, as described above. These heavy section components are often produced as castings and tend to be thicker than the forged equivalents. Despite this they are generally regarded as being more tolerant of thermal transients.

In older castings, which have seen service, small thermal fatigue cracks can be ground out and the section re-profiled to reduce stress concentrations. In practice, such defects often regenerate. In such cases it is probably best to leave the defects in-situ and to monitor their growth as continued machining and repair welding may be more damaging in the longer term. The problem needs to be managed through a program of routine inspections and planned replacement and repairs at scheduled outages.

In the case of boiler stop valves, many of the valves fitted to older boilers were not designed with regular thermal cycling in mind. Many of them suffer the effects of thermal cycling resulting in cracking of valve bodies, valve seats and valve discs. They also exhibit operational problems as increased usage causes wear and tear on the valve stem and driving gear.

Recent developments include the use of P91 (9CrMoNbV) steel which significantly reduce the overall wall thickness and has thus been expected to reduce tendency to cracking (see later the discussion on P91, especially the Appendix on P91 performance). Where the cost of these valves is unjustified, an alternative is to have a set of spares which can be replaced during a planned outage and subsequently refurbished on-site for use at the next outage. The cost of a replacement boiler stop valve is in the range of US$40,000 to $80,000. Turbine steam chest and valve assembly costs vary considerably and cannot be quoted here.

Older designs of boiler stop valves often produced a higher pressure drop than newer designs so there may also be an efficiency reason for changing the valves, perhaps aided by rising fuel prices. This combined with the metallurgical reasons above may enable the financial case for replacement to be made.

B.1.3.2 Superheater and Reheater Header Ligament Cracking

Thermal fatigue cracking of the ligaments between header stubs and penetrations is recognized as one of the primary life limiting mechanisms on headers. The problem manifests itself primarily in the form of cracking in the bore of the header in the ligaments between stubs but can also be found on the outer surfaces around stubs and other attachments. Failure to recognize this problem could lead to catastrophic failures.

The susceptibility of any header to this problem is a function of its wall thickness, the spacing of stub holes, material of construction and the operating conditions. The headers perceived to be at greatest risk are the superheater outlet headers, especially those associated with horizontal tube self draining superheater elements. The cracking usually forms in the circumferential direction where the ligament efficiency is often low, but the phenomenon does appear in longitudinal ligaments. Isolated penetrations may also exhibit 'star' cracking where cracks radiate in all directions.

Intermediate headers running at lower operational temperatures are also exposed to the problem, especially if they have not been closely monitored. Installations of thermocouples have shown that these headers can be subject to high temperature swings at start up. Inner wall temperatures will be directly influenced by steam temperatures. Thus a superheater header which normally operates at high temperature (>500°C) and which is well insulated may retain high temperatures on shut-down for a short period. On start-up, should condensate form in the tube sections, or possibly saturated steam (usually at about 360°C or even lower if operating on a sliding pressure regime) be admitted to the header, rapid cooling will take place and could give up to 200°C temperature difference. Reheater headers, by virtue of their relatively thin wall construction are perceived to be at low risk of ligament cracking.

If cracking is present, it is possible to justify continued operation based on:

- The extent of cracking
- Any previous inspection history to enable crack growth rates to be estimated
- The likelihood of the cracked ligament resulting in a leak rather than rupture
- The consequences of failure and hazards arising
- A program of inspection based on predicted crack growth rate. CCTV inspection of the header internals is often used to assess the size of the cracks. Care has to be taken when making these inspections as any surface scale on the parent material can contain cracks which can be mistaken as cracks in the parent material.
- Monitoring of temperatures including through wall temperatures
- Structural assessment
- Changes to operating practices to reduce thermal stresses which could include more gradual loading / de-loading rates or even avoiding of cyclic operation completely until header repair or replacement.

The outcome of this assessment will determine the actions to be taken. Options include:

- Continued operation in the presence of cracks with associated program of monitoring and inspection
- Local repair of the cracks in the header (suitable only for isolated cracks - not very reliable)
- Replacement of worst affected sections of the header with short inserts
- Wholesale replacement of the header
- The choice of action will be influenced by the safety implications, availability of materials for repairs, the cost of the downtime to affect the repair and the cost of any operational constraints imposed by the defect and the cost of the repair.

The problem is now well understood and has been greatly reduced by improved header design to reduce stress concentrations and to improve ligament efficiency (Figure B-3). The latter include rearrangement of stubs in a diagonal format, application of manufacturing controls to ensure correct tolerances and alignment of apertures, and designing out of sharp edges and radii.

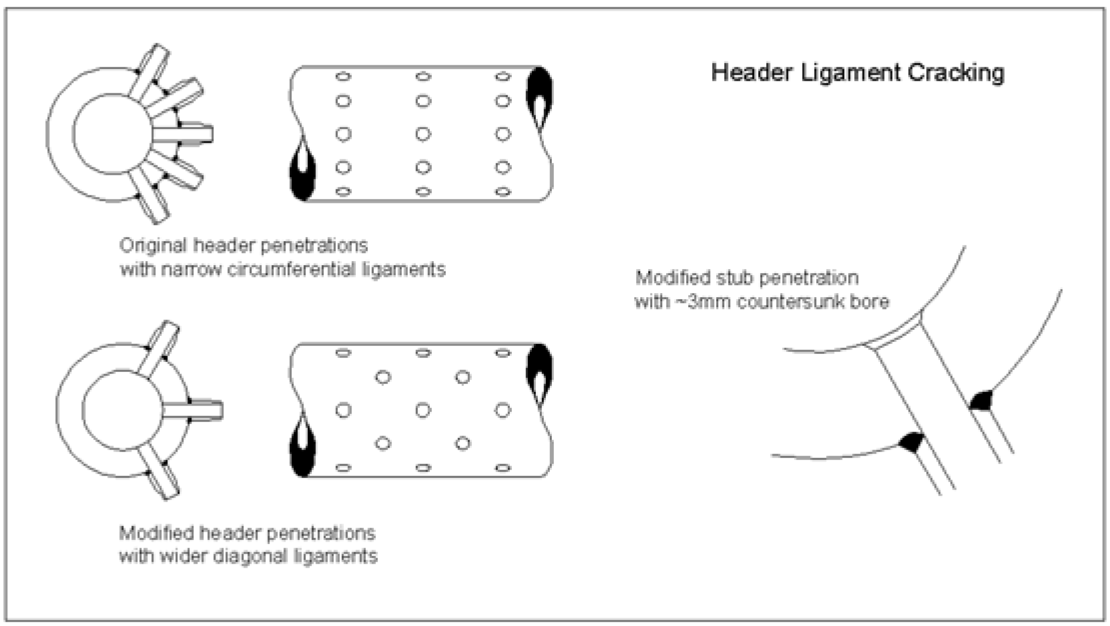


Figure B-3
Improved Ligament Design

Cracking can be avoided/reduced by implementing the following:

- Improved boiler operation to avoid severe temperature excursions and to drive to pre-set criteria identified above. This is achieved by monitoring temperatures at inner and outer walls, header inlet and outlet stubs etc. to establish actual rates of temperature rise (or quench) and through wall temperature variations and diametric temperature variations. Analysis of these variations, usually using FE modeling, can be used to estimate the stress ranges generated and to define acceptable limits to work to.

- The use of stronger materials such as P91 (9% Cr martensitic steel). This results in the use of thinner sections. However, more recent preliminary research findings (discussed in this report) show that P91 may have problems in terms of its vulnerability to Type IV and accelerated cracking under creep/fatigue interaction.
- The application of routine inspection using a combination of internal CCTV or intrascopic visual examination or external ultrasonic examination. Ongoing operation of the plant needs to recognize that cracking may occur at some time. It is therefore necessary to adopt an inspection strategy to monitor the situation. Typically this would comprise CCTV inspections of the bore of the header with some external NDT using ultra-sonic and magnetic particle methods, discussed elsewhere in the report.

B.1.3.3 Evaporator Header Stub Cracking

During boiler light up, the expansion of furnace wall tubes may not necessarily be uniform across the boiler. The centre tubes are more exposed to firing and tend to expand more rapidly than the wing tubes. If the tubes are connected to rigid headers at top and bottom, then the differential expansion has to be absorbed by flexibility with consequential development of internal stresses. A similar situation can exist when the unit is off load and uneven cooling occurs. This internal stressing will generally concentrate at the stub to header connection. Cyclic high stresses can lead to thermal fatigue cracking of the stub to header weld or the stub to tube weld especially on bottom water wall headers and economizer headers.

In the case of natural circulation boilers the tubes which initially receive more heat will start circulation in the upward (correct) direction. Ideally this will start a flow down the downcomers (the large bore external pipes from the drum to the headers at the base of the evaporator (or furnace wall) tubes. Sometimes, however, a downward flow is started instead in some of the evaporator tubes themselves. This can result in alternating areas of hot and cold across the tubes and headers thus increasing the stresses.

Whilst this problem will reveal itself as increased tube failures, its effect can be anticipated and minimized by early examination of susceptible header stubs. If the problem exists there are several options:

- Regular inspection and local repair at planned outages.
- Replacement of stubs with stronger (thicker) stubs able to accommodate the higher stresses. This solution may provide a short-term palliative but does not necessarily guarantee the longer-term integrity.
- Modification of the tubes to incorporate greater flexibility. This is the preferred option.
- Replace long headers with shorter interconnected box headers.

This phenomenon of differential expansion also puts the headers into bending. There have been some instances of header weld failures that may have been, in part, attributable to cyclic bending induced by the differential expansion of tubes. Replacement of the header with box headers eliminates this problem.

B.1.3.4 Economizer Headers

Similar thermal fatigue problems have also been found in economizer headers. These are subject to thermal shock on start up when slugs of cold feed water are injected into the boiler as flow is established. Although thermal fatigue is encountered during base load operation, the magnitude of cracking can crease quite dramatically. Header replacement or repair is one option. An alternative is to recirculate some of the water from the steam drum into the feedwater line, with the object of raising the inlet feed temperature.

In the case of a station with more than one unit then it is possible to run a line between the economizer inlet headers on the boilers of the generating units. As a result when it is necessary to top up the boiler of a unit either during the night or as part of the start-up operation warmer water can be taken from a running unit thus significantly reducing the thermal shock.

This does assume that at least one unit is on load. Even if all the units in a station shut down overnight one will be the last to shut down and one (perhaps the same or different) will be the first to start up. This means that there is a “donor unit” available for at least some of the shut down period for most of the units.

B.1.3.5 Feed Heaters

The temperature range experienced by feed heaters is much less than on boiler components, but the thick sections of tube plates and end covers of traditional designs are often a source of leakage under cyclic operation. The stiff sections of the joints are prone to thermal distortion and consequent failure of the seals. The bolting of these joints is usually highly strained, and vulnerable to high stress low cycle fatigue. It is important to monitor the integrity and life usage of these bolts with a clear policy on bolt replacement. Other areas of potential problems include thermal fatigue cracking of baffle plates and division plates.

More modern feed heaters utilize integral distribution headers which are inherently thinner section and less vulnerable to thermal fatigue problems. Some feed heaters utilize tube bifurcations which can also be prone to thermally induced failure. A replacement program may be required for these.

Most units have a bypass line around the high pressure (HP) feed heaters which allow the plant to run (albeit) less efficiently with the HP heaters out of service. This line can be used during cycling but it is a balancing act between protecting the HP heaters from thermal shocks and increasing slightly the thermal shock to the economizer inlet headers.

B.1.3.6 Tube Ties

Tube failures attributed to attachment weld failures are one of the main areas of reduced availability when boilers are in cyclic operation.

Boiler tubes are held in position by attachment either to adjacent tubes by means of slip ties (C’s and T’s) on platens, or by brackets to cold steelwork on furnace walls. Under cyclic operation, these attachments are subjected to a high number of cycles and are prone to thermo-mechanical fatigue failure, sometimes resulting in tube failures as fatigue cracks penetrate through wall. Some attachments, by virtue of their bulk, also generate local hot spots and can enhance the creep and fatigue effects.

There is little that can be done operationally to minimize attachment failures. The problem is essentially one of detail design and its solution lies in modification of the attachments. This problem is usually overcome by arranging for the cold steelwork to be connected to the hot components via a flexible link. Replacement of furnace wall tube ties with cranked ties gives greater flexibility. Platen alignment can be effectively maintained by the use of wrapper tubes.

The modification of tube ties is a relatively low cost activity when incorporated on an opportunity basis during planned outages. The main cost is related to gaining access and replacement of boiler casings and thermal insulation.

B.2 Expansion Related Issues

Expansion related issues associated with cyclic operation occurred in various components of conventional plants are discussed below.

B.2.1 Boiler Structures

Boiler structures are subject to considerable thermal movement. A typical large boiler will expand downwards from its roof supports by 250 mm, with lateral expansions of 150 mm. This expansion has to be accommodated by a ‘cold’ support framework that has to be designed to permit relative expansion. In particular the furnace wall buck stays, windbox attachment, gas ductwork and boiler supports have to accommodate considerable expansion.

Buckstays are usually attached to the furnace walls via a link and sliding clip arrangement. Under base load conditions, this mechanism is virtually static, but under thermal cycling conditions, the mechanism is required to flex on a regular basis. Failure of buckstay attachment and linkage is a frequent problem in older base load plant subject to thermal cycling.

Similarly, the attachment of wind boxes and air and gas ductwork to the boiler has to accommodate the thermal movement. This is usually achieved using a slip bracket assembly. This is prone to seizure and build-up of compacted dust thereby reducing its effectiveness.

Expansion joints are subjected to increase cycling. In addition to the increased mechanical cycling effects, entrapment of dust may cause a jacking effect, forcing the sections apart over a number of cycles.

Boiler supports is required to accommodate the thermal movement between the hot pressure parts and cold support steelwork. These are usually clevis pin or rocker type connections and do not generally pose a problem under cycling. However, should these connections seize, the sling rods will be subject to cyclic bending. This could result in failure, especially on the shorter sling rods near the wings of the boiler. Their failure could lead to collapse of boiler pressure parts. A potential additional problem is load migration where the support load is transferred across the boiler either by relaxation on highly loaded supports or their failure; this may lead to overloading of individual supports and their subsequent failure. Periodic inspection is advisable, possibly with check weighing of the load in supports, to identify where this problem might be occurring and to confirm the integrity of the support structure.

B.2.2 Pipe Work Systems

Steam pipe work between the boiler and turbine not only has to be able to accommodate its own thermal expansion, but must also accommodate the movement of the boiler and turbine. Most pipe work is inherently flexible, but can also generate extremely high system stresses if the supporting structure is not adequate.

Most pipe work systems use constant load supports to facilitate pipe movement, which can be in excess of 400 mm between its hot and cold positions. These support units typically have a load variation of less than ± 5% on supporting effort over the movement range. These supports are therefore very susceptible to changes in load, either by physical changes such as changes in insulation weight, changes in valve and actuator weights, from deterioration of the support mechanism due to increased friction (seizure) and build up of dust on the pipe work.

Over a period of time, especially when associated with thermal cycling, the net consequences can be for the pipe work to drop (or occasionally lift) due to a combination of weight change and deterioration of the support mechanism. If the pipe work becomes locked in position, the resultant system stresses will tend to focus on the terminal connections at the boiler or turbine and can give rise to creep and fatigue damage, usually in the welds.

It is important that under cyclic operation to ensure that the pipe work is free to move throughout its full design range. A full visual survey of the pipe work in both the hot and cold positions is required and the movements of individual supports identified and compared with design. In the case, the significant differences appear (say > 20% variation on movement), further analysis should be carried out to establish the likely consequences and the need for modifications. The installation of control points (limited movement) and the use of occasional variable load supports where the movement is small may prove to be beneficial.

B.2.3 Differential Expansion of Turbine Rotors and Casings

Expansion and differential expansion of the turbine rotor and casing are not usually a problem under cyclic operation, although it is essential to have good turbovisory indication of turbine movement and clearances.

Relative movement between the rotor and casing during turbine run up is always a potentially critical period when rubs can occur both on turbine blade tips and on shaft seals. Where cycling is introduced, it is important to understand what is happening within the turbine from the turbovisory equipment. Each case has to be evaluated on its own merits. Where problems arise, the solution may require changes to operating procedure or increasing clearances, albeit at the expense of efficiency.

The main issues are caused by the de-loading and loading rates. Any significant period of time where the steam temperature into the turbine is reduced (by either throttling losses or low steam temperatures from the boiler) to below the previous temperature of the inlet stage of the turbine will start to cool the rotor more rapidly than the stator. This is because of the differences in mass of the two items and the fact that the rotor is surrounded by steam whereas the stator is not.

If a situation arises where the rotor contraction approaches its limit relative to the stator then the operator's options become severely restricted. Basically the load and steam temperature needs to be increased rapidly to get the rotor to expand once more or the unit will have to be shut down quickly (without further chilling). If it is shut down then it will need several hours for the rotor and stator temperatures to stabilize thus returning the differential expansions to normal before another start can be attempted.

Although there is usually no throttling used on the steam valves supplying the Intermediate Pressure (IP) turbine the IP rotor can contract during prolonged low load running due to the temperature drop across the stages of the IP turbine. At low load (< 10%) the exhaust of the IP turbine is barely above atmospheric pressure and steam temperatures are correspondingly low.

Turbines are often built with complex greasing systems to feed the sliding surfaces of the turbine. As turbines are generally anchored at the condenser – hence the Low Pressure (LP) turbine the sliding surfaces are on the HP and IP turbines. These are the hottest areas which make the grease prone to overheating and drying out. As mentioned working on these sliding surfaces can be a major exercise. The pedestal between the HP and IP cylinders is rarely accessible without removing at least one cylinder. The front end (often called the governor pedestal) can often be removed by just supporting the end of the HP cylinder and removing the HP rotor.

It is also important to reliably measure the movement of the bearing pedestals. If it is known early that a pedestal is sticking then it is possible to take some action – operating the greasing system, for example. If a pedestal repeatedly exhibits problems then maintenance work can be planned. Without these measurements the pedestal problems can only be inferred from other indications such as differential expansion or thrust bearing position indication.

LP rotors have posed problems due their construction with shrink fit diaphragms. Cyclic loading may jeopardize their integrity. Some of these have been replaced with monoblock forgings. LP turbine differential expansion problems are often controlled by sprays located in the exhaust areas which counteract the heating which can occur at low steam flows.

When monitoring LP turbine differential expansion it needs to be remembered that many LP turbine rotors get longer as they run down (and shorter as they run up) due to the Poisson effect. The centrifugal force generated by the rotation of the blades causes strain in the rotor body perpendicular to the longitudinal axis of the rotor which then results in contraction along this axis which increases with speed.

B.2.4 Alternator Components

The stress variations which result from cyclic operation are more important, although variations in the electrical output can also induce fatigue, even though there is no change in the rotor speed. This is due to eddy current induced heating in the end ring.

Older designs of end ring are likely to be more susceptible to damage, since they are prone to high cycle fatigue. One of the older variants uses retaining rings that are shrunk onto the end disc, which itself is shrunk onto the shaft. Here a relative movement between ring end and the end windings occurs with each revolution. The other design can induce cracking in the shaft, in the rotor tooth region and in the end ring itself. Here the ring is shrunk on to the end windings and the shaft. The most modern design, which is less susceptible, to high cycle fatigue, is of the so-called “cantilever type”. Here the end ring simply grips the alternator rotor, and does not rely on support from the shaft. Another shortcoming of older designs was that the fixing of the end ring to the rotor was of a relatively simple form. Modified designs are available to reduce stress concentrations in this area.

B.2.5 Cycling Problems Associated with Steam Turbines

From about the mid 1970's, most turbines were designed on the basis of a 200,000-hour operating life with up to 5,000 hot starts, 1,000 warm starts and a few hundred cold starts. Evidence to date would suggest that in general most turbine plant is on course to achieve this objective. The forced outage rate attributed to turbines is historically quite low with values of less than 0.5. The general perception is that turbines do not suffer significantly from operation under cyclic regime, provided of course that due care is taken as set out below.

Most large turbine currently in use conform to a set of standard modules, usually comprising HP, IP and LP turbines based on a manufacturer's 'standard' configurations. HP cylinders are typically a single flow type with double shell construction, while IP and LP turbines are usually double flow single shell construction. The majority of rotors are monoblock type with two journal bearings located outboard at each end of the cylinder. The thrust bearing is usually located between the HP and IP turbines. Blading is usually a disc and diaphragm construction.

Operation under cyclic regime has two main effects:

- Thermal fatigue and associated creep fatigue
- Mechanical fatigue due to load and speed variations

Modern analysis methods, utilizing finite element methods are now widely available at reasonably low cost to permit modeling of components perceived to be at risk. Application of this type of modeling, whilst it may not be able to accurately predict the life of components, does provide a valuable understanding of the stress profiles within the component and identifies potential weaknesses and vulnerable areas. Armed with this knowledge, operational procedures can be optimized to minimize the effects of thermal fatigue and inspection procedures focused on selected locations at appropriate operating intervals. The addition of temperature and temperature differential instrumentation will enable the operator to minimize the intensity and duration of the adverse conditions which can then be incorporated into auto start sequences. The scope for modification to existing turbine plant is limited unless new rotors or casings are being fitted. Possible modifications to reduce thermal stresses include improvements to thermal insulation, pre warming (especially of half joint flanges) and slotting of flanges to increase flexibility. EDF are known to favor "skin peeling" of high temperature rotors where fatigue damage is of concern. This is basically skimming off a millimeter or so of material from the surface of potentially critical regions of the rotor e.g. radii etc. This effectively removes fatigue related damage what is essentially an "as new" surface. This process is typically carried out at midlife.

Another area of concern has been the effect of embrittlement and fatigue on the critical crack size of high temperature rotors. Major rotor failures in the 1980's (e.g. Gallatin in USA and Irsching in Germany) led to development of inspection methods (e.g. borosonics) and assessment procedures e.g. EPRI's Safer code. The potential for embrittlement is largely a function of residual or tramp elements which are strongly influenced by the steel making process. Generally only older rotors i.e. pre 1974 have a significant potential for embrittlement. Significant embrittlement can result in the material behaving in a brittle manner at high temperatures which could be experienced under weekend warm start conditions. Whilst severe embrittlement is relatively rare this aspect should be assessed for older rotors which are under cyclic operation.

The mechanical fatigue issues arise from two sources. Firstly, during turbine run up, the rotor passes through a series of critical speeds where vibration levels increase significantly. This is a well-understood phenomenon and the critical speeds are well defined for most machines. Clearly it is important to pass through these speeds as quickly as possible. Over a number of starts, the number of cycles at the critical speeds can accumulate to significant values and subject components such as turbine blades to unacceptable high cycle fatigue levels. The most vulnerable area for mechanical fatigue is generally regarded to be the LP blading. Obviously the length of the blades subjects the root area to very high centrifugal stresses. Any defects within this high stress area will make a significant reduction of blade integrity. The inspection of disc slots on some older units with large numbers of starts has identified the formation of fatigue cracks in the root serrations where stress levels are concentrated. These cracks are believed to propagate slowly and have not resulted in any major problems although blade replacements have been required. More modern units have employed modern design and analysis methods to improve the blade-root detail which should eliminate this potential problem. Where mechanical fatigue problems are encountered with LP blading (or indeed any blading) it is usually possible to re-blade the rotor with a modern design of blade to reduce or eliminate the problem. In some instances, re-blading may be combined with improvements in efficiency. Where cracking of the LP blades is present, the cracking may be exacerbated by the onset of corrosion fatigue. During this study the possibility of resonance related LP blade failures was revealed for large load cycling units, but as yet this has not been confirmed.

Four other lesser potential problems have been identified although evidence to date suggests that these are relatively minor problems:

- Increased wear and tear on turbine valve gear
- Overheating of turbines due to windage
- Turbine differential expansion
- Erosion due to oxide (scale) impacting on HP and IP blades

Clearly, operating in a cycling mode will require increased operation of turbine governor valves and stop valves. Inevitably there will be additional wear and tear on the valve seats and valve stems, especially under throttling conditions when flow induced vibration can lead to mechanical fatigue and wear. This can, in the main, be contained by redesign of the valve head, modification to the steam flow path and the use of stellite or similar hard facing materials on wear surfaces.

As the flow through a turbine cylinder reduces, conditions could arise where the turbine is actually driving the steam. This can lead to a degree of overheating. Two circumstances where this occurs are on HP cylinders where a by-pass system is engaged. As the discharge pressure of the HP cylinder increases, the flow through the cylinder decreases until a no flow situation can arise. This has resulted in high HP cylinder temperatures and subsequent damage. A similar problem can occur with LP cylinders at low loads where the flow is reduced to below the threshold value and the last stage blade may impart energy into the flow.

It is important when moving from a base-load situation, where efficiency is of particular concern (and hence the need to minimize blade tip and gland clearances) to review the clearances and make adjustments appropriate to a two-shift operating regime where reliability may be increased at the expense of some efficiency.

There is some evidence that cyclic operation can result in oxide in boiler tubes and steam mains becoming detached and carried forward into the HP or IP turbines. This problem is uncommon in the UK but more prevalent in the USA. It may be the result of excessive thermal shock under abnormal conditions or of rapid load shedding under fault conditions. Where these particles are small, they will be carried through the filters and enter the turbine. The high velocity impact on the nozzles and blades will result in increased wear which, over a period of time, can lead to significant levels of erosion. In one instance, the buildup of scale deposits on the filter resulted in increased pressure drop across the turbine and hence reduced performance. It is important to recognize the potential for this problem and, in the event of a thermal shock to the system, to check for and clean out any accumulation of scale. The presence/use of bypass systems (which are relatively rare in the USA) appears to minimize solid particle erosion.

Where new turbines are to be installed for cyclic operation, the following design features should be included:

- Use of a disc and diaphragm construction to reduce rotor construction.
- Ample axial and radial clearances to accommodate thermal expansion and differential thermal expansion.
- Use of high-strength materials to minimize wall thickness on steam chests, valves and turbine casings to maximize thermal response and minimize thermal transients.
- Application of FE modeling of the new turbine to optimize thermal transient effects
- Bypass system.

B.3 Corrosion and Fouling Related Issues

Corrosion and fouling related issues associated with cyclic operation are discussed in this Subsection.

B.3.1 Waterside Corrosion in Economizers, Feedheaters and Evaporators

Much of the increased incidence of aqueous related corrosion in cyclic operation can be traced to the interruption in condenser, condensate polishing and water treatment plant operation, which is likely to occur during the cyclic operation. This results in increased levels of oxygen and ionic species in the boiler water. The main difficulties will occur over a weekend shutdown, when, in most cases, water treatment plant is normally shut down for lengthy periods and boiler temperatures will drop to near ambient, so that there is no reserve of steam for de-aeration.

As a result of cyclic operation there will be a need for increased supplies of feed water. This is due to the necessity to drain off condensate from pipe work, manifolds and turbine casings, and to the problems in maintaining boiler contaminants below specified levels. Clearly, the more thoroughgoing the efforts to keep the boiler water within specification, the more likely the requirement for an increased supply of feed water.

It is likely that there are differences between once through and drum type boilers since the former are normally operated with minimal levels of inhibitors, so as to avoid deposition of solids at the steam water interface. Hence during steady state operation, the risks of contaminant induced failures, in once through systems, will be that much less. Conversely, it would seem that once through boilers could experience significant problems during cycling when the chances of contamination will greatly increase. With drum type systems there is the ability to blow the boilers down to reduce contaminant levels.

At higher temperatures, typical of evaporators and high temperature feed heaters, the autoclave studies showed a much more dramatic effect, with cracking occurring within a few tens of cycles when the dissolved oxygen was around 10000 ppb. However, it is considered that the risks of encountering this sort of level during normal operation is quite low, since the oxygen level would have fallen away by the time that the plant got up to temperature.

On the other hand it is possible to envisage cold slugs of oxygenated feed water being admitted to the plant, as it is going on load. The combination of high oxygen levels plus the thermal stress engendered by the "cold slug" sweeping its way through the system could be very deleterious. Clearly work is required to specifically address this area, although in principle it should be possible to eradicate the "cold slug" problem by changes to operating practice. Nevertheless, it is difficult to see how thermal stress can be eliminated during start up, and this does bring us to the central question.

The issue of sodium contamination increases with boiler pressure. Research indicates the more important problem is that of sulphate which leads to pitting corrosion. Other work has indicated that low pH is the principal cause of corrosion fatigue, which is more likely to happen with congruent phosphate or AVT conditioning. These depressions in pH may result from admission of CO_2 or inadequate amine injection rates. Equilibrium phosphate or sodium hydroxide control is better.

B.3.2 Deaerator Cracking

Recent investigation is targeted towards an understanding of stress corrosion cracking of weldments in deaerators. This is something of a historical issue since modern designs of deaerator are stress relieved during the final stages of manufacture. Furthermore modern steels, with low sulphur contents, are much less susceptible to this form of attack.

The in-situ studies also showed that during normal base load operation, when the deaerator output was essentially constant, the potential gradually fell until it was just below the critical value. The safe limit in this case was reached after some 400 hours of operation. However, if there was any disruption in output the potential would jump above the critical level. These increases were not necessarily related to oxygen level, or any other definite potential corrodants.

Clearly these findings will be of concern to operators since cyclic operation by its nature will result in poorly controlled oxygen levels and variations in flow rates. It is also possible that some of the corrosion fatigue phenomena, which occur on other parts of the system, may be related to these potential effects.

B.3.3 Steam Turbine Erosion, Corrosion and Fouling Aspects

B.3.3.1 Steam Turbine Erosion

Steam turbine blade erosion is of two types. Erosion by particulates is due to oxide scales and is found at the front end of turbines. Erosion by water droplets is essentially a back end problem.

Oxide scale erosion is normally due to the exfoliation of magnetite scales from the superheater and reheater. Some plants pointed out that during cycling, peak steam temperatures could be up to 30°C over design. This probably implies an even greater increase in metal temperature, perhaps as much as 50°C, in some localities. For 2.25Cr1Mo steel this would double oxidation rates, but this alone would not necessarily have too much effect on exfoliation. It seems likely that although cyclic operation will cause some initial difficulties with erosion, once loose oxide has been shed, the problem will disappear.

Of greater significance is the risk of erosion corrosion due to water droplets at the back end of the LP turbines. Off design operation can lead to increases in steam wetness, which results in more rows of blades being affected. An additional erosion problem can arise if water is deliberately injected to bring down the back end temperatures of steam turbines.

A.3.3.2 Steam Turbine Fouling and Stress Corrosion

These two issues are linked since they essentially result from the same phenomenon, the carryover of boiler water salts and impurities that deposit out on turbine blades and rotors. Obviously the problem of carryover is more likely with drum boilers that have continued to use the congruent phosphate treatment.

Cyclic operation implies that the difficulty in controlling feed-water and condensate quality would increase the risk of carryover. Nevertheless the risks of turbine fouling would appear to be lower than in normal operation. Steam temperatures and pressures will be below normal levels for most of the time, and the fact that Wilson line in the LP section of the turbines will move around during start up and shut down should tend to wash away deposits. The need to flush through the turbines, to control temperatures may also help.

Stress corrosion of turbine blades and rotors is likely to increase with cyclic operation. This will be due to greater steam contamination. Sections of the turbine, which are not normally susceptible to attack since they are operating above the dry-out line, will be operating in a hot wet condition.

Much will depend on the turbine construction. Older designs of rotor using shrunk on discs would appear to be more at risk than rotors of the monoblock or welded construction type. Accordingly there could be a plant-to-plant variation, although even with modern equipment some designs may be more susceptible than others. Laboratory work suggests that a major factor in governing resistance to stress corrosion is the yield strength of the rotor material.

B.4 Fireside Corrosion

Fireside corrosion associated with cyclic operation for various components are discussed below.

B.4.1 Superheaters and Reheaters

Serious fireside corrosion has been confined to UK plants and a limited number of units in the USA, where steam temperatures were at or in excess of 565°C. Here Type 300 austenitic tubing, in some instances clad with Type 310 or IN 671, was used to give additional protection.

The attack involves the formation of a molten alkali-iron trisulphate layer, underneath the ash layer. Below about 550°C the trisulphate is solid, but above about 780°C it dissociates, hence this compound is only corrosive over a relatively limited temperature range. The exact mechanism of corrosion is still open to conjecture, although it probably involves a fluxing mechanism combined with simple sulphidation attack at the deposit-to-metal interface.

One important question is whether operation in a cyclic mode is likely to alter the furnace environment so as to change the attack rate. Earlier views suggested that the level of SO_3 in the furnace atmosphere was critical. It is now considered that the bulk of the SO_3 forms within the deposit, as a result of catalytic reactions within the ash layers, thus stabilizing the alkali-iron trisulphates. The sulphur content of the furnace gas will only have a minor effect.

The effect of furnace (i.e. combustion product) temperature is likely to be more marked. High furnace temperatures are known to increase attack rates. In the opinion of the authors furnace temperature has an indirect effect, involving the decomposition of molten trisulphate near the outer surface of the deposit, thereby increasing the concentration of SO_3. This will increase the rate of attack either by combining with the metal, or by increasing the acidity of the melt.

It follows that the combustion of high sulphur fuel oil during start up should not have a dramatic effect on corrosion rates. Furnace and metal temperatures will be low for much of the start up and these parameters are likely to be more significant. Some acceleration will occur if the fuel oil contains significant quantities of vanadium.

However, although deposit spalling does occur, there is as yet no clear evidence of an increase in the rate of attack. On the one hand, the metal surface will be exposed to the full effects of the furnace environment. On the other, fireside corrosion involves the formation of a reasonably thick layer of ash and slag. Without a buildup of ash there is no way in which the activity of SO_3 can reach a critical value.

High chlorine levels in the coal can also increase fireside corrosion issues due to the lowering of the ash fusion temperature. Ideally the ash softening temperature should be at least 50°C above the gas temperature entering the superheater section. This will stop the ash from adhering to the tubes and being difficult to remove with soot blowers thus exacerbating the problems discussed earlier.

B.4.2 Furnace Wall Corrosion

Furnace wall corrosion is due to a combination of oxidation and sulphidation, the latter giving a marked increase in the rate of attack. Here it should be noted that, in attack involving sulphur, it is common to find a layer of sulphide beneath the oxide and metal. The sulphide layer can be regarded as the "shock troops" of the corrosion mechanism, since sulphidation attack is very much faster than simple oxidation.

The issue, however, is more than one of simple metal wastage. Thermal stress/fatigue in boilers leads to craze or elephant skin cracking of the tubing or water walls. Cracking of this type can be difficult to identify due to the layers of ash and slag that cover the affected parts. Good water treatment practice can help, reducing metal temperatures.

With the development of low NOx burners, furnace wall corrosion has become a very serious issue in some situations. Many of the designs increase H_2S levels. Because of these problems some utilities have resorted to the use of austenitic or nickel based welded overlay coatings. Although these have given protection, there is concern about distortion following welding and the levels of residual stress.

Cyclic operation seems certain to exacerbate the furnace wall problem. The short time of operation implies that the furnace structure will be cold and it will be difficult to ensure that the pulverized coal will be burning properly. This will lead to locally reducing conditions, which are known to add to the furnace wall sulphidation problem. It also seems likely that if there is a complete burner failure, this too would result in unburnt pulverized coal impacting on the furnace wall, increasing corrosion rates in that area of the furnace.

All of this has to be seen in the context of greater amounts of thermal and mechanical cycling, with an increasing severity in the amount of elephant skin cracking. Overlays may not be the complete solution, particularly in thermally stresses areas. Differences in expansion coefficient between the ferritic and austenitic alloys will result in thermal/corrosion fatigue, both on the fireside of the tubing and on the water side too.

B.5 Dust Removal

In general, electrostatic precipitators perform better at low loads due to the reduced proportion of unburnt carbon in ash and increased residence time of the gases in the precipitator thus allowing more of the dust to be collected. However, it is important to ensure that the temperature in the precipitators does not fall below the dew point as any moisture can result in a buildup of dust, which, if pozzolanic, can prove difficult to remove. The acid gases will also increase corrosion at lower temperatures. If low temperatures do become a problem, it may be necessary to install a warming system (e.g. a gas burner system) to pre-heat the precipitators when bringing the unit back on load.

During boiler start up, it is usual not to energize the precipitators until stable combustion has been established. This may give rise to emission problems into the local environment. Where this is not acceptable, it will be necessary to review the operating procedure with a view to energizing the precipitators earlier in the start-up procedure. In the case of bag filters, the main problem is to avoid temperatures dropping below dew point.

Electro-static precipitators need special consideration due to the potential risk of moisture causing dust to adhere to the electrodes which subsequently becomes baked. Maintaining temperatures above 90°C will reduce this problem.

In order to minimize the problems with the electro-static precipitators then unless the start up is quick and the gas temperature through the precipitators can be maintained then consideration should be given to isolating some of the banks of the precipitators during the early stages of the start up. Passing the gas through just one bank will restrict the cooling and potential depositing of dust and / or residue from start up oil burner combustion to this one bank. The impact on emissions of this one bank taking some time to recover to normal is less than if it affects all of the precipitator banks.

B.6 Corrosion and Fouling of FGD Systems

Very few Flue Gas Desulphurization (FGD) systems have been installed and most of those fitted are associated with high load factor plant.

The problems with FGD systems can be summarized as follows:

- Inability to operate the unit at all in cyclic operation due to lengthy warm up or attack by fuel oil residues on linings.
- Fouling due to build up of sludge.
- Cracking of reinforced polymeric linings and other components.
- Enhanced corrosion due to dry-out of solutions

There are, however, problems associated with taking units out of service. It is necessary to purge the units to remove solids settling out and solidifying. This can have a significant effect on the water consumption as a proportion of the generated load.

Lime or limestone are the usual reagents and are often fed into the absorbers at a fixed rate. Obviously, at low loads, there will be an excess of reagent in the slurry, which although it will increase the SO_2 removal, may result in increased scaling.

Further difficulties may be experienced during rapid load changes when it will be necessary to match the throughput of the scrubber with the required reagent. The time delay characteristics may need to invoke some form of control system to anticipate load changes.

Failure to balance the throughput with the reagent may lead to high alkali levels and the associated problems with corrosion. This can in part be overcome by improved water treatment management and upgrading of lining materials.

At reduced loads, the incoming gas temperature is likely to be low. This may influence the reaction rates. Where regenerative heat exchangers are used, the net effect may be a significant reduction in exit temperature which will reduce gas buoyancy and induce dew point corrosion in the ductwork and chimney. It is therefore usual, when cycling, to by-pass FGD plant until temperatures have been stabilized.

B.7 Stress Corrosion and Corrosion Fatigue of Alternator End-Rings

The most critical problem is that of stress corrosion of the generator end rings. This appears to be a problem associated with earlier Fe-18Mn-4Cr alloy that was extremely susceptible to attack if the hydrogen-cooling medium became contaminated with moisture. The more modern Fe-18Mn-18Cr steel is essentially immune to stress corrosion in these situations.

Scarlin *et al* stated that the newer alloy, although significantly, better than the Fe-18Mn-4Cr alloy, is somewhat poorer than most steels in a corrosion fatigue situation. Hence these authors advise that the rings be kept dry at all times. How practical this is with modern generators is open to question since many of these are air rather than hydrogen cooled.

Indeed, although the performance of the Fe-18mn-18Cr steel has been in general good, some plants experienced severe cracking of a retaining ring after six years of operation. The subsequent investigation emphasized the importance of copper in the failure mechanism, since deposits of this material in an elemental form were found on the crack surfaces, although Cl, Br and S containing compound were also found. The copper probably came from the windings, and would have become incorporated in some way in a corrosive solution, which in turn induced pitting and stress corrosion of the retaining ring. It seems possible that some of the corrosion took place when the equipment was off-line for an extended period. If this were so it would be significant for cyclic units, which are often stood down during the summer months.

In order to minimize the risk of stress corrosion cracking it is vital to keep the dew point of the coolant gas (often hydrogen) within the alternator as low as possible.

B.8 Other Issues

Most utilities engaged in cyclic operation have moved to a more proactive maintenance regime to anticipate wear and tear.

B.8.1 Pumps and Auxiliaries

Many of the auxiliaries are subject to increased wear and tear during cyclic operation. Boiler start up and standby pumps, which might otherwise rarely operate on base load, will be required to operate more frequently. Steam driven main boiler feed pumps, like the associated turbines, will also be subject to increased thermal cycling. Fans, vacuum raising plant, lubricating oil systems and condenser extraction pumps will be similarly affected.

Valves are normally subject to more frequent operation. A common problem is leaking of gland packings due to the increased usage. An effective remedy is to fir live (spring) loaded gland followers to keep the packings under a constant load.

B.8.2 Electrical Equipment

<u>General</u>

Damage to electrical equipment that could be attributed to the plant cyclic operation has not been reported. Normally cyclic operation does not necessitate departure from normal inspection/ maintenance procedures or intervals. However, station staff should be aware of a number of generic issues relating to cyclic operation and should inspect and monitor their plant as part of its normal inspection/ maintenance regime keeping these issues in mind. These generic issues, in the main, apply to the three areas outlined hereunder.

Motors

The number of starts per year imposed by cyclic operation should not normally adversely affect motors. However, the problem of rotor bar cracking on large motors has been encountered worldwide following a move to this type of running regime. Abrasion of the stator coil insulation at the slot emergence has also been experienced. This has been sometimes linked to the number of starts.

Generators

There are a number of generic issues relating to cycling of generators.

The material properties and machining quality of modern generator rotor shaft and end rings should ensure that they are not adversely affected by cycling. Some rotor designs have suffered from copper dusting whilst the machine is barring. This phenomenon is caused by the individual turns moving radially in the slot under the influence of gravity. The copper rubs against the slot insulation leaving deposits and possibly leading to an inter-turn fault.

The relative movement of copper turns during the heating and cooling cycling of cyclic operation can cause abrasion of the inter-turn insulation. Another common problem encountered is stick/ slip of the winding leading to transient vibration excursions.

Thermal cycling of the generator stator can lead to the development of partial discharge phenomena in the slot region. This is caused by expansion/ contraction of the conductor bars relative to the core, degrading the semi-conducting coating of the bar. This is a long-term issue and it is possible to detect this type of degradation by the use of on-line discharge monitoring equipment.

Another potential effect of thermal cycling is that stator slot wedges may become loose. Again, this is a long-term issue. It would not be expected to become a problem before the first scheduled removal of a machine's rotor and so the wedge tightness can normally be checked then and a suitable strategy formulated. Systems are available to perform wedge tightness checks with the rotor installed.

Switchgear

Cyclic operation would normally be expected to impose extra wear and tear on switchgear due to the increased number of operations imposed. This has proved to be the case in practice.

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C

DAMAGE MECHANISMS FOR COMBINED CYCLE POWER PLANTS ASSOCIATED WITH CYCLING OPERATION

Understanding of specific damage/failure mechanisms that affect various components of the combined cycle (CCGT) power plants and implementation of appropriate engineering solutions is crucial to avoid component failures. The damage mechanisms associated with cyclic operation for various components of CCGT plants are discussed in this Section.

C.1 Typical Damage to Gas Turbine (GT)

Typical damages for various GT components associated with cyclic operation are described below.

C.1.1 Gas Turbine

Nozzles

The stationary blades or nozzles suffer from cracking, especially at section changes and around the cooling holes and, if the cracks are allowed to grow, pieces of the nozzle can break off and cause extensive consequential damage downstream. This cracking is accelerated by load cycling and cycling. Thermal cycling can result in loss of thermal barrier coating, which leads to overheating of the substrate and reduction in life.

Blades

The problems with the rotating blades are similar to those of the nozzles, but with the added influence of the cyclic centrifugal stresses.

Turbine discs

The discs do not suffer from the same large thermal cycles as the blades and the nozzles, due to the relatively cool air from the compressor section supplied through this part of the turbine. The hottest area, and therefore the region in the turbine disc to suffer most, is near the blade root, but this is generally not an area that produces problems, although there have been isolated reports of disc cracking.

Turbine casing

The casing area is also kept relatively cool by the compressor air being channeled along this path to the nozzles for cooling.

Bearings

The thrust bearing experiences the greatest load changes in stop-start operation. The entire load is on the compressor end during start-up and rundown, with less duty on the bearing at full load. The bearings can suffer from increased wear when the turbine is barring or when off load, due to improper lubrication caused by the slow speed of the shaft or the oil being too viscous as the temperature is so low.

Middle and exhaust end bearings can experience severe temperatures if not properly cooled by a cooling medium such as water or air. These bearings have been known to give trouble due to insufficient cooling and exhaust end-bearing pedestals have suffered from creep.

Exhaust plenum

This section suffers from stop-start operation as the thermal stresses can cause severe cracking of the exhaust flow guides, which tend to have many changes of section and thickness, thus increasing the stresses.

Exhaust duct

This area is prone to many problems from cracking and leaking seals, which increase with cyclic operation. The duct cracks, and bolts holding the plates can also break off, allowing some paneling to come loose. The bolts and the seals can come loose and allow hot exhaust gases to escape to the outside, impinging on nearby sensitive plant components causing consequential problems. Insulation panels can break off and be carried into the HRSG. Vibration effects have also been known to affect the insulation (breakdown with subsequent compacting resulting in overheating).

Exhaust diffuser

As well as being subject to temperature changes, the turning vanes in the exhaust diffuser may be subject to untoward stresses, due to excessive swirl during start-up and shut-down, leading to cracking. Poor detail design on this component and difficulties in fabrication contribute to these problems.

Expansion joints

These joints suffer from fairly frequent failure as they provide flexibility for the differential movement between the boiler and the gas turbine exhaust duct. They are generally made from a mixture of insulating material and a metallic braid. They suffer more from a stop-start situation when the boiler movement has to be taken up by the joint. They also suffer from exhaust gas turbulence in the duct, making the joint flutter.

Baffles

These baffles act as flow-straightener and sound suppression barrier. The outside skin is generally made from a perforated metal sheet, which cracks along the holes, due to thermal stresses from start-stop operation.

C.1.2 Combustion Section

Can type combustors

These types of combustors suffer from cracking of the can and the cross-fire tubes between the cans. This area experiences considerable vibration from the combustion process. They will experience large thermal cycles due to cycling. Atomizing air and steam/water injection nozzles experience cracking. Spring seals holding the cans in place also suffer cracking due to thermal cycling and vibration.

Silo type combustors

All the combustion takes place inside a silo. Again nozzles for atomizing air, and water/steam low NOx injection nozzles can suffer from cracking. Wall tiles can come loose and crack due to the thermal-mechanical stresses.

Annular type combustors

This type of combustor is more common for aircraft-type engines but they suffer from similar problems with regard to cracking due to thermal cycling.

Transition piece

This section experiences the highest temperatures carrying the hot combustion gas from the combustion chamber to the 1st stage nozzles. This section does not tend to suffer as much from thermal cycling, as generally it has no sudden changes in cross-section and has an even temperature.

Igniters

These can get burned out if they remain stuck in the start position, and also fail stuck in the out (running) position preventing starting of the turbine. This problem increases with cycling.

C.1.3 Compressor and Ancillaries

Inlet filters & housing

This area should be little affected by cyclic operation. Generally this area has two types of filter: cyclone and screen type. Maintenance on this section is directly related to the number of running hours and the air quality.

Compressor

The compressor section will suffer temperature cycling, at the high pressure end, due to cycling and also pressure cycling of the casing, valve, pipe attachments and intercoolers. *Surging* is more likely during frequent cycling, leading to damage of compressor blades and overheating.

Inlet guide vanes

This mechanism will suffer under cycling, due to cycling of the vanes, with wear taking place on the vane shaft bearing, on the ring gear and pinions of the vane actuating mechanism. It will suffer leaks and wear on the vane actuating hydraulic ram. This type of problem will also arise due to load cycling.

Starting bleed-off valves (axial compressors)

These valves tend to get stuck in the open position during start up, or worse are stuck in the closed position during start-up, leading to compressor surging and blade failure. These valves and their pipe attachments will suffer from thermal cycling, as they are only required to work during start-up and shut-down.

Auxiliaries

A variety of problems are caused by cycling, ranging from heat exchanger leaks, stuck valves, leaking flanges, and failure of instrumentation.

Starting device

There are various means of starting: diesel engine, motor, gas expander, or generator. Frequent starting puts extra burden on these devices, increasing their failure rate. These items of plant experience a heating and cooling cycle for every start and for short duration runs.

Barring gear

The risk of shaft rubs on the stationary parts of the turbine increase with frequent starting and stopping, causing overheating where the rub takes place and also wear damage to blades and seals.

Lube oil system

The usual problems are stuck valves causing overcooling of the oil in the off-load condition and insufficient cooling when at full load. There is an increased risk of the failure of the heat exchanger which will allow cooling water to contaminate the oil supply and an increased risk of the AC oil pump failing to start when the turbine comes off load.

Hydraulic oil system

This generally uses the same oil as the lube oil system but on a separate circuit. It is sensitive to degradation if insufficient cooling is taking place, for the same reasons as the lube oil. The hydraulic oil is more sensitive to degradation as it has to work in equipment with very finely machined surfaces and clearances.

Atomizing air

Generally it is the coolers that leak due to thermal cycling from off load to on load, but again this is generally not a major problem. The atomizing air compressor can suffer from corrosion due to water vapor condensing in its housing, and this happens more with start-stop operation.

Gear box

Generally there is a single gearbox from which all the auxiliary drives are taken and through which the starting device operates. This is a poorly designed part of the plant in many cases (non-generator type starter). The gears can be overloaded driving auxiliaries that only need to operate when coming off or going on load. Small misalignments of the gear shafts cause gear teeth and bearing wear. Expansion and contraction of the gear wheel shaft results in increasing or decreasing clearances between the gears. These problems are more likely to develop with changing load conditions such as going on and off load.

Cooling water system

This system is used to take away heat from all the heat exchangers, cooling the various parts of the turbine and its auxiliaries. Again with cycling it introduces problems for control valves and temperature changes causing leaking flanges. Freezing while off load in a cold climate can happen if the antifreeze concentration gets diluted by make-up water.

Low NOx steam & water injection

This plant also gives trouble with stuck valves, leaks and un-drained steam lines, which happen more frequently when going on and off load. Dry low NOx type burners are now the norm on newer plant.

Heat exchangers

Heat exchangers suffer failure from cracked tubes due to load cycling and differential expansion between tube plate and tube. These leaks can result in consequential damage ranging from water in generators to hydrogen leaks into the cooling system, and also leaks into the lube oil system. The atomizing air system can suffer from thermal shock by cold water getting into hot atomizing air supplying fuel nozzles.

Some gas turbines have regenerative heat exchangers to preheat the combustion air from the exhaust gas. These heat exchangers suffer from thermal fatigue due to starting and stopping.

Valves, flanges, pipe work

The usual problems associated with these items of plant due to temperature changes, i.e. valves sticking and the consequential damage caused by such malfunctions. Flange bolts loosen and joints pull due to the contraction and expansion of the associated pipe work resulting from temperature changes (start/stop). Leaking flanges also can cause extensive consequential damage to other plant.

Gas compressors

Many combined cycles require the incoming gas to be compressed. This plant can itself be of considerable size and complexity, and suffers from many of the problems that the main plant experiences due to frequent starting and stopping. For example, heat exchangers, control valves, etc.

C.1.4 Cyclic Damage Experience per GT Models

It is observed that there are some variations in the type of cyclic damage experienced by different GT models. A summary of historical information on these damage experiences with the support of its worldwide collaboration and are below.

GE Frame 6, 6FA

Some utilities reported that their plants have experienced damage to the first stage nozzles and combustion cans, and the damage has increased due to operational cycling. One plant with 6FA machines reported cracking in the exhaust diffuser area and implosion of the combustion liners during cyclic operations, but these problems were not made worse by the cycling.

GE Frame 7

Combustion can cracking has definitely accelerated with much of the cracking occurring around the cross-over tubes. Blade seal wear tends to increase in cycling mode and thermal barrier coating (TBC) loss from the cans has also increased. Cracking has been found in the compressor disc of one machine. Cracking was noted by one user in the combustion liner transition pieces.

GE Frame 9B

Some plant operators have experienced increased damage to hot gas path components. One utility with a plant that has been two-shifting since 2004 (after operating in base load mode for many years) reported greater wear and tear of hot gas path components, including liners and burners, due to the two-shifting regime. Failure of the vibration sensor on a bearing was also reported; the issue here was not the sensor but the down-time for replacement.

GE Frame 9E

Some utilities have experienced cracking of the first stage blades and nozzles, TBC spalling, and damage to the exhaust diffuser and ducts, as a result of cyclic operating regimes. A number of plants have suffered from blade cracking in base load as well as cycling duty.

One plant that has been two-shifting occasionally since 2000 (after many years base load operation) reported a number of problems that had increased due to cyclic operation: torque converter failures; cracking of exhaust diffuser; load tunnel compartment hot gas leaks. The condition of hydraulic oil was reported to be possibly worse due to cycling.

One utility with a plant that has operated since 1994 and has been two-shifting since 2001 reported the following problems:

- Premature failure of first stage nozzles due to thermal fatigue caused by the cycling regime;
- Damage to the insulation boxes of the exhaust plenum;
- Damage to the exhaust ducts - loss of insulation, damage to the silencers resulting in pressure casing distortion and extensive cracking.

GE Frame 9F

Damage to various components was reported: deformation and cracking of outlet casing; cracking of silencers and loss of enclosed noise damping material; cracking in corners and deformation of duct between GT and HRSG; cracking of burners and creep damage due to higher burning temperatures at start-up and very low load; and cracking of blades. In general, the damage was reported to be made worse by cycling.

GE Frame 9FA and 9FA+e

Some utilities reported blade cracking, damage to blade coatings, and damage to combustion cans, etc, due to operational cycling. One utility with a plant that has been cycling since 2005 reported loss of TBC on the first stage blades, and damage to combustion cans, combustion can liner and transition pieces. Another utility reported that there had been cracking through the coatings on the leading edges of blades after 25,000 hours operation.

One plant operator reported various forms of damage: wear and hot spots on combustion components; cracks on ignitor/flame scanner holes; bulging/buckling, wear and burning of combustion liner components; damage to transition piece components (buckling, cracking, wear and loss of TBC); damage to compressor stage 15 stator hook fit, causing it to rock; and cracks at two locations in turbine stage 1 wheel cooling slot. However, the operator could not attribute any of the damage to cycling because the plant has been operating in a mixed base load and load-following (60%-100%) mode, with occasional start-stop cycles, since it started operating in 2003; there has been no period of sustained base load operation for comparison.

One utility with a fleet of Frame 9FA machines, which have been operating in load-following mode since commissioning in 2002-2005, reported no problems.

GE Frame 9FB

One utility has been operating Frame 9FB machines at three plants that were commissioned in 2006, 2007 and 2008. No cycling-related damage has been experienced to date.

Siemens V64.3

Some plant operators reported that stator and rotor blade cracking were exacerbated by cycling, and oxidation of blades/coatings was also accelerated. Cracking also occurred on the inner section of the combustion chamber at some plants. Such cracks have been repaired by welding.

One operator reported the onset of stator and rotor blade cracking after the plant had been running for about 12 years in load-following mode. One possible cause was the use of a coating with better oxidation resistance, which was adopted as coating-degradation had been a problem in earlier years. Unfortunately this new coating was somewhat less ductile than the older formulation.

Siemens V94.2

Hot section damage was reported by some utilities. One plant operator reported that cracking of the first stage stator blades occurred after the plant had started doing some cyclic operations, following 2-3 years base load operation.

Siemens V94.3, V94.3A

Some utilities reported cracking of stator blades and loss of TBCs due to cycling. One unit ran with stators in a cracked condition for six months. Disc cracking was also experienced by one user of the V94.3. Some plant operators stated that the rate of oxidation of coatings and damage to combustion can liners were also accelerated when their units started cyclic operations. One user reported IGV actuators were wearing out under the impact of repeated cycling and were not lasting the planned 25,000 hours.

Alstom GT 13e1

A plant that has been operating for over 12 years predominantly in base load mode, with overnight load reduction and limited cycling, experienced start reliability issues that were made worse by the operational cycles. These issues have been resolved. There had also been a start limitation due to the original blades, and this issue was also exacerbated by cyclic operation. A new design of blades was fitted to rectify matters.

MHI 701 F3

One plant operator reported that no problems have been experienced during two years of cyclic operation.

C.1.5 GT Areas of Increased Maintenance

The areas of GT experienced increased maintenance due to cyclic operation is summarized in the Table C-1.

Table C-1
GT areas of increased maintenance

Operating regime	Areas of increased maintenance / inspection
Two-shifting since 2001	- Gas turbine exhaust plenums and exhaust ducts - Borescope inspection of the gas turbine first stage nozzles in advance of scheduled Combustion Inspection - Monitoring of generator rotors slot liner condition - Maintenance of 3.3kV breakers
Cycling since 2005	- Large boiler valves and drain valves - Steam turbine stop and control valves - Increased HRSG inspections
Cycling since 2006	- Steam valves: blocking of stems and leaks
Two-shifting occasionally since 2000	- Boiler casing and internals - Exhaust diffuser, exhaust and load compartments
Load-following for 12 months	- Valves and actuators
Load-following since 2002	- Periodical borescope inspections between main inspections
New base load plant with limited cycling	- Increased inspection of P91 welds and P91 to P22 dissimilar metal welds (following issues with as-delivered components)
Two-shifting & load-following since 2001	- Modification of internal insulation in the duct between GT and boiler, following problems with cracking and deformation of duct
Mixed base load, load-following and occasional starts/stops since 2003	- Increased inspection of tubes, steam traps and valves - Inspection of tubes – visual, MT, creep assessment and metallurgical replication - Inspection of steam traps – by operator rounds, and thermography - Inspection of valves – stroke and function test during every shutdown

C.2 Damage to Heat Recovery Steam Generator (HRSG)

The damage mechanisms associated with cyclic operation during the life cycle of HRSGs are discussed in this section. The common damage mechanisms occurred in HRSGs at different time period over the full life cycle are presented in Table C-2.

Table C-2
Common mechanisms occurring in HRSGs

Component	Low cycle fatigue	Thermal shock	Creep	Flow assisted corrosion	Corrosion fatigue	Deposits/ corrosion	Oxidation/ Exfoliation	Gas-side corrosion	Gas-side erosion	Thermal expansion
Superheater headers	X	X	X							
Superheater tubes	X	X	X				X			
Reheater headers	X	X	X							
Reheater tubes	X	X	X				X			
Evaporator tubes	X			X	X	X		X		
Economizer headers	X	X		X	X	X				
Economizer tubes	X	X		X	X	X		X		
Drum	X	X		X	X	X				
Steam Piping	X	X	X							
Feed / connecting pipes	X			X	X					
Casing, liners, duct, etc	X	X	X					X	X	X

Flow assisted corrosion (FAC) and cycle chemistry issues related damages occur mainly at ‘Start-up Failure Period’ of the life cycle. Damages like short term overheat, long term overheat, thermal quenching, corrosion-fatigue, stress corrosion cracking, internal pitting, caustic gousing, acid phosphate corrosion, hydrogen damage, gas side corrosion, acid dew point corrosion etc. occurs during the ‘useful life cycle’ period. Most of the fatigue related damages such as thermal fatigue, creep fatigue, flow induced vibration fatigue etc. occurs at ‘wear our period’ of the life cycle. These damage mechanisms are discussed below.

C.2.1 Flow Assisted Corrosion

Flow assisted corrosion (FAC), or flow accelerated corrosion, mainly occurs at the 'Start-up Failure Period' of life cycle. As this damage progresses from the inside the first warning may be pinhole leakage to the surface, but internally the tube wall may have been corroded away leading to serious local thinning. If access to the internal surface is possible, a typical scalloped surface may be found, as shown in Figure C-1.

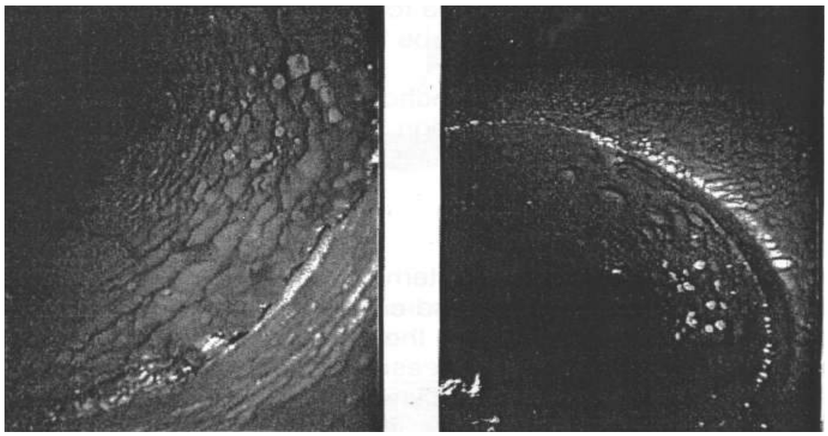


Figure C-1
Surface appearance of flow assisted corrosion

Damage is restricted to water-touched components and, as it is worse in the range 100-250ºC, is frequently restricted to the low-pressure circuits. Critical areas for FAC include the low-pressure economizer and evaporators including steam drum internals, vent and drain lines. Leaks may also develop at areas of high flow or turbulence such as tight bends, downcomer pipe bends, and steam drum separators. If access is available to the internal corroded surface, inspection is relatively simple using videoprobes, endoscopes and borescopes. An alternative is the use of ultrasonic thickness measurements from the outside, internal rotary inspection probe (IRIS) from the inside or radiography for small bore drain lines. Eddy current techniques are available for pipes and plain tubes but are not suitable for finned tubing.

Standard practice is to measure the rate of thinning over a period of time. Provided the predicted thickness at the next inspection outage is above the code minimum thickness, the component or tubing may be left in service. Beyond this, a fitness for purpose approach based on the calculation of allowable local thicknesses may be possible. Ultimately the tubing or component must be replaced, preferably with a chromium-containing alloy, though local weld overlay repairs, to increase the remaining thickness, may be used as a temporary repair. Such repairs can initiate thermal fatigue cracking. Improvement of water chemistry would address the root cause of the problem.

C.2.2 Cycle Chemistry Issues

While selection of the water chemistry is an integral part of plant design it should be carefully evaluated during commissioning and early commercial service. In essence, the requirements of the unit will change during the service life and the cycle chemistry should be periodically reassessed. Failure to recognize that the original cycle chemistry is no longer optimal may result in damage and failures. One example is the current trend to cease or avoid the use of oxygen scavengers to stop or prevent FAC in HRSGs.

For HRSGs in which the economizers and evaporators are constructed of carbon steel, protection from corrosion is achieved by controlling the purity and pH of the feedwater and boiler water and also by controlling the dissolved oxygen concentration in the feedwater. The aim is always to maintain conditions in which the solubility of metal oxides is reduced to a minimum so that corrosion and oxide transport processes are effectively suppressed.

While oxygenated treatment is not presently in common usage with HRSG units, it has found widespread acceptance in conventional units where it is used in both once through and drum units.

C.2.3 Short Term Overheat

Found on *steam tubes*, this result from an interruption to the flow of steam through the tube. With insufficient or no internal working fluid, the tube temperature increases rapidly and remains for some period, ranging from minutes to months, at elevated temperature well above the normal operating temperature. This elevated temperature reduces the tensile properties but most of the damage found normally results from accelerated creep deformation at the higher temperature. Under such conditions, the material is highly ductile and typically swells at the hottest area before developing a longitudinal split which then blows open. Such failures can occur on plain tubes downstream of the supplemental burners giving a thin edged failure with a typical 'fish-mouth' appearance. With finned tubing, swelling and/or bulging may also be present, but may not be visible due to the presence of the fining.

Internal blockage can be detected with videoprobes, endoscopes and borescopes. External examination is normally visual looking for bulging or signs of overheating on the tube and/or fins. Damage is normally quantified by tube sampling, and tubing which shows thinning and/or bulging outside tube diameter and wall thickness limits is replaced.

C.2.4 Long Term Overheat

Once more found on *steam tubes*, long term overheating is a result of operation at temperatures above the normal design/operating temperature for protracted periods of time, typically several years (Figure C-2). Temperatures are not as high as those found in short term overheating/steam starvation and the tube material does not show the same high ductility. However, creep damage still accumulates resulting in accelerated microstructural evolution eventually leading to failure.

Due to the lower temperature, swelling and bulging are not evident – unlike *Short Term Overheat* - but the oxides adjacent to the overheated area will be thicker, due to the higher operating temperature, and wall thinning may result. Damage is normally found in steam tubes adjacent to supplemental burners or on the low alloy side of material transitions. High temperature areas of the HRSG are most at risk, such as high-pressure superheaters and reheaters and tubing/pipe work at the outlet headers.

Detection of damage frequently requires the local removal of fining to allow access to the tube surface. This allows ultrasonic inspection for wall thinning and/or internal tube oxide measurement. The internal tube oxide thickness can be measured from the tube exterior using specially developed ultrasonic equipment. As the tube oxide thickness is dependent on the tube operating temperature, this can be directly converted into the tube temperature. Alternatively, measurements may be made on the bare tubes, adjacent to the header, and tube temperatures elsewhere in the tube bank calculated from this information. It is possible to scan the tubes using internal rotary inspection probe (IRIS) techniques in a similar manner to that in heat exchangers, however this will involve cutting tube stubs to allow access and chemical cleaning may also be required to ensure that the tubes are clean enough for inspection. These activities will be time consuming and therefore will require extended outages.

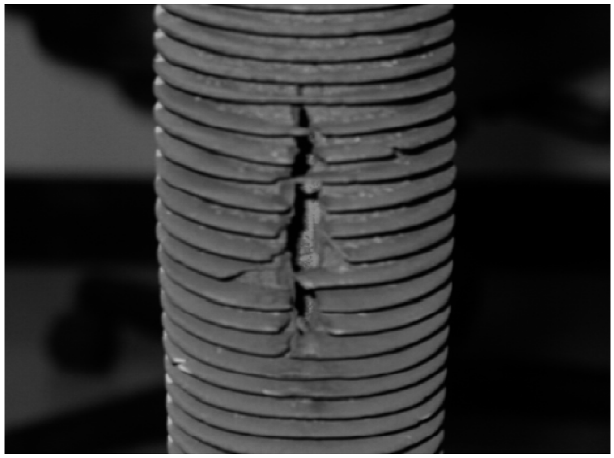


Figure C-2
Overheat failure of finned tube

C.2.5 Thermal Quenching

This is the damage caused by large thermal stresses arising from quenching of a hot surface with condensate or attemperator spray. The stresses developed are very high, typically above yield, and result in physical distortion, ovalization of tubes and cracking. Damage occurs on steam tubes in high temperature areas or where the tubing is restrained by other components. High-pressure superheater and reheater banks and tube to header connections are typical areas for damage. Damage may progress from the inside of the tube making it difficult to detect.

Visual inspection for distortion and surface inspection (DPI or MPI) for cracking are used. Quantification of damage depends on measuring the crack depth and ultrasonic inspection and exploratory grinding are commonly used. Potential drop measurements have also been applied. One operator has reported that the potential drop measurement technique is good for single cracks, but would give false readings for multiple cracks, as it sums up the impedance of all cracks and reports the crack depth for one single crack. It is quite common to have multiple cracks due to thermal shock, although some cracks are smaller than the major one. Repair involves the replacement of the tubing and examination of adjacent areas. The cause of the condensate or other quenching must be investigated and the thermal quenching prevented or failure will re-occur.

C.2.6 Corrosion Fatigue

This is cracking which develops under the influence of cyclic stress and a corrosive environment. It occurs at lower stresses than required for normal fatigue and may occur in fluid conditions, which do not result in corrosion in the absence of cyclic stress. Initiation frequently involves the repeated disruption of the protective oxide resulting in the formation of pits, which may act as sites for concentration of corrosive species. Cracking then initiates at these pits and oxide filled multiple cracking is typical. Corrosion fatigue develops on the inside as pitting which then develops into parallel cracks running either circumferentially or longitudinally, perpendicular to the driving stress. Typical cracking is shown in Figures C-3 and C-4. Again usually a 'family' of cracks initiates, one of which then propagates through wall.

Corrosion fatigue is most common on water tubes, but can also be found in steam tubes as a result of condensate-induced corrosion during outages. If not previously discovered, this will present as pinhole leaks on the outside of the tube. This indicates that damage has progressed through wall and extensive cracking may be present on the inner bore. Emergency repairs, by local overlay welding, will have limited life. Extensive bore cracking is present along appreciable lengths of tubing and replacement of the damaged section with a tube insert is required. It is usually associated with high stress regions such as tube connections, especially to headers and drums, tube bends and welded attachments. It is often a result of thermally induced stresses and moments.

If access to the internal surface is possible, at drums and headers, detection is relatively simple using videoprobes, endoscopes and borescopes. Ultrasonic inspection, using angled beams, can detect the cracking from the outer surface of the tube but the quantification of the extent of damage may require tube sampling in addition to ultrasonic NDE.

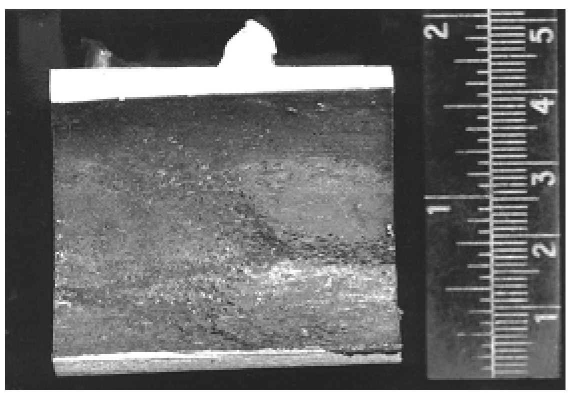


Figure C-3
Typical Corrosion fatigue

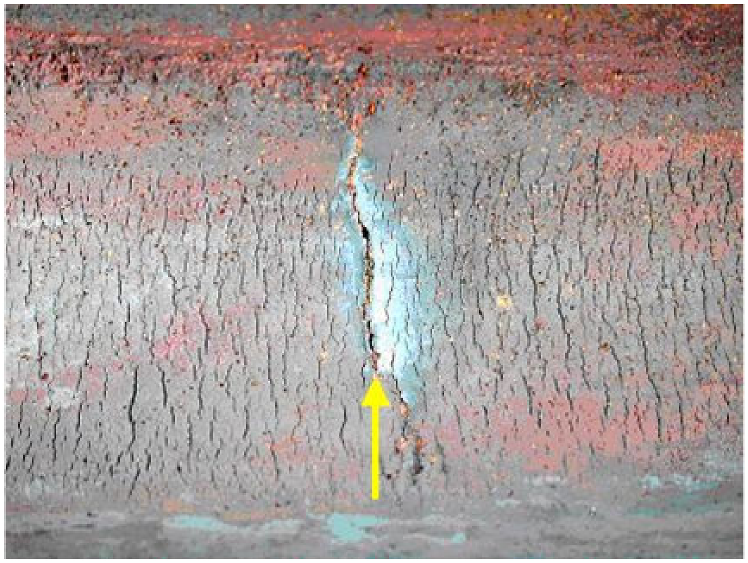


Figure C-4
Corrosion fatigue cracking

C.2.7 Stress Corrosion Cracking

This is a form of cracking which develops in the presence of particular material/ environment combinations and a static tensile stress either applied or residual. Stress corrosion cracking (SCC) is found in areas of high stress concentration such as bends, attachment welds and supports, and in areas where contaminant levels may be high, i.e. areas wetted with condensate. Stress corrosion cracking has been found in a range of components such as feedwater heater tubing, evaporator tubing, and steam tubing. It can also develop in a range of materials including carbon and low alloy steels, austenitic steels and brasses. Typically, there is virtually no attack over the surface and SCC results in fine cracking, orientated perpendicular to the dominant stress direction. The appearance of the cracking can vary with the particular material/environment combination (Figures C-5 and C-6).

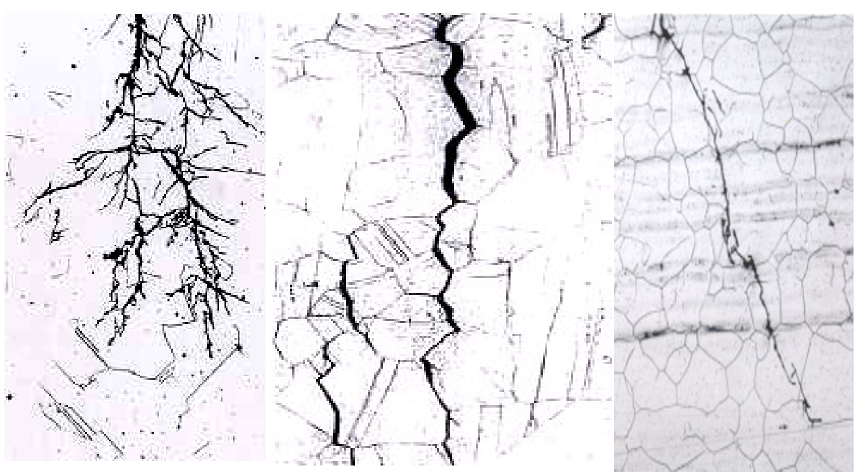


Figure C-5
Typical SCC cracking in Austenitic Stainless, Brass/Ammonia, Alloy Steel/Caustic Chloride environment

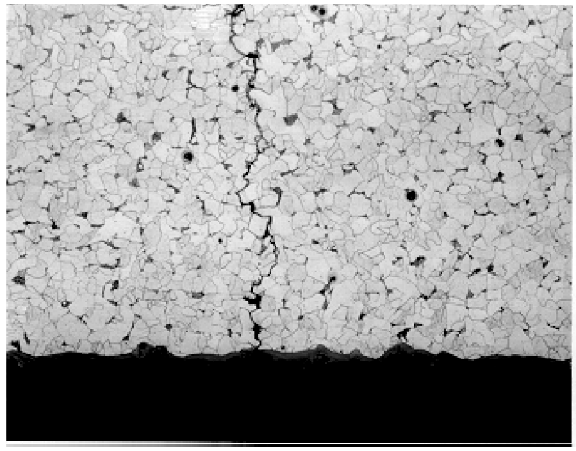


Figure C-6
Intergranular SCC in LP superheater tube bend caused by sodium hydroxide in water carried over from the drum

If not detected earlier, damage will present itself as a fine pinhole leak. However, with exploratory grinding the crack like nature of the defect will become apparent. SCC can be found by visual and surface inspection techniques only when it has penetrated through wall. Eddy current has also been employed successfully for detection. Cracked material must be replaced either with a different alloy, the chemistry altered or the source of the driving stress removed.

C.2.8 Internal Pitting

This is a form of localized internal corrosion, which results in the formation of deeply penetrating pits on the inner surface of the tubing. It is normally associated with air ingress during storage, resulting in high oxygen neutral pH water in contact with the tubing (Figure C-7). It has also been found because of acid attack during poorly controlled chemical cleaning. It can be found on any component in contact with an aqueous solution, including both steam and water tubing, as well as condensers, condensate pipe work, sparge tanks, deaerators and feedwater heaters.

In theory, damage can be identified by internal examination by videoprobe, endoscope and borescope. In practice, it has normally either been found by inspection of tube samples or tube failures from other causes. Pitting adjacent to seams in seam-welded pipe is also possible. By the time the tube has perforated, large areas of the internal surface may be severely pitted and widespread tube replacement may be required.

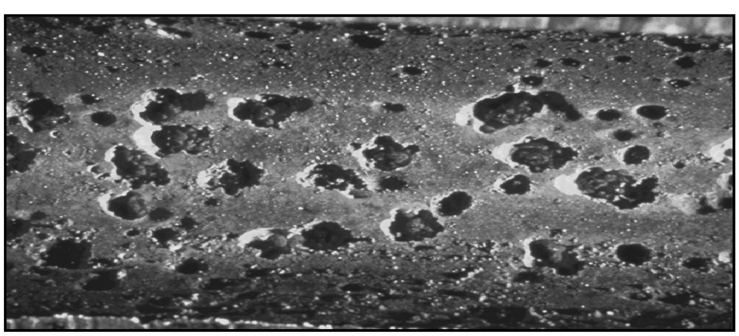


Figure C-7
Oxygen pitting of tube surface

The severity of damage has been quantified by the internal rotary inspection (IRIS) ultrasonic system. If the remaining tube thickness, after pitting, is greater than the specified tube thickness, the tubing may be left in service but conditions creating further pitting must be avoided. For greater pit depths, a fitness for purpose analysis may allow further service. The water chemistry should be reviewed and improved. If pitting cannot be arrested, the only repair option is replacement and the use of higher alloyed material should be considered.

C.2.9 Caustic Gouging

This is a waterside corrosion mechanism resulting from incorrect local water chemistry. The general chemistry may be too alkaline, typically due to ingress of contamination, or there may be a local concentration mechanism for the caustic (alkali) salts. Typically, a salt deposit builds up on the tube surface, at the point of highest heat flux, and a highly corrosive environment develops below it leading to the failure of the protective oxide layer and the rapid dissolution of the underlying metal (Figure C-8). The tube interior will contain hemispherical or elliptical depressions, which will eventually penetrate the tube causing pinhole leaks. The corrosion depressions may be filled with thick, layered deposits though these can be lost once a leak forms resulting in a high local pressure drop. This form of corrosion occurs mostly in the high and intermediate pressure evaporators where high heat flux occurs in combination with boiling. Damage is associated with areas of high heat flux and areas of deposition including gas inlet rows, horizontal tubing and module end tubing.

Damage can be seen from the tube interior if inspection with videoprobes, endoscopes or borescopes is possible. In severe cases, damage can be seen in radiographs and with ultrasonic thickness measurements either using an IRIS probe from the inside or from the outside. The *repair option* is to replace tubing when the remaining thickness reaches the specified minimum.

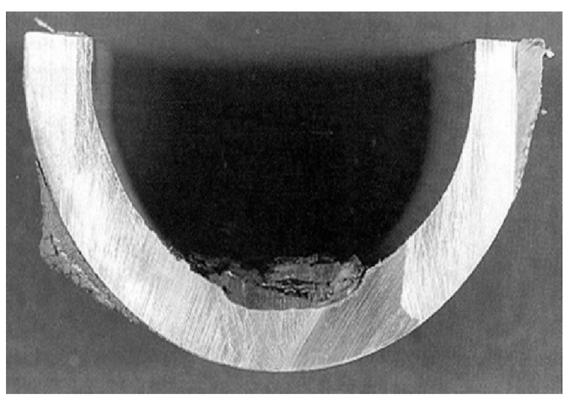


Figure C-8
Example of caustic gouging

C.2.10 Acid Phosphate Corrosion

This is similar to caustic corrosion in appearance but occurs when the boiler is operating a phosphate chemistry. Once more, it is a waterside corrosion mechanism, which develops due to local concentration of the chemicals in the boiler water producing deposits on the high heat flux surfaces. The local concentration of phosphate salts leads to the breakdown of the protective oxide film and rapid under deposit corrosion results. Internally the tube surfaces show a series of gouges often containing layered deposits with a distinct transition between areas of mild and severe corrosion. If not discovered earlier, it will eventually perforate the tube resulting in pinhole leakage. Damage occurs in the same high deposition, high heat flux areas as with caustic corrosion. Inspection and repair options are also the same as for caustic corrosion.

C.2.11 Hydrogen Damage

Hydrogen damage results from the development of acid, or possibly alkali, deposits on the internal tube surfaces as described above. While this can result in under-deposit corrosion and tube thinning, it can also liberate nascent hydrogen as a by-product of the corrosion reaction. Nascent or atomic hydrogen consists of a single Hydrogen atom, not the conventional H_2 molecule, and can lose its electron to any nearby metallic material leaving a free proton, which can rapidly diffuse through the metallic matrix. Nascent hydrogen is highly soluble in steel at elevated temperatures and high levels can be absorbed at under-deposit corrosion sites.

While dissolved in the steel it affects the steel ductility rendering the material more brittle. The hydrogen also accumulates in voids and at grain boundaries within the steel, reverting to molecular hydrogen or reacting with carbon, from the steel, to form methane gas molecules. In either form, it can generate internal pressures of several hundred atmospheres, literally blowing grain boundaries apart and leaving the steel full of grain boundary micro-fissures (Figure C-9). Hydrogen is attracted to areas with high residual stresses, which tend to be the weld HAZs where the structure is already strong and less ductile. Hydrogen damage can lead to relatively brittle tube failures with longitudinal splits in finned tubes and circumferential cracking at attachment welds.

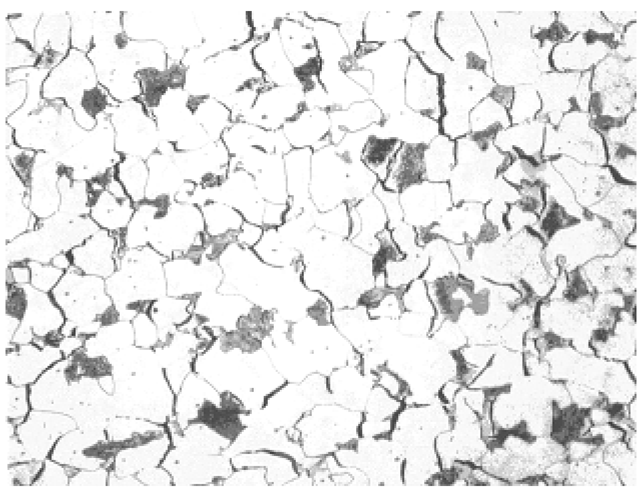


Figure C-9
Grain boundary hydrogen damage

Hydrogen damage can be detected with specialist ultrasonic equipment, typically by measuring a change in the ultrasonic attenuation caused by the micro-fissuring. In most cases, the associated corrosion and/or gouging can be detected by conventional ultrasonic thickness or IRIS inspection. Hydrogen damaged tubing must be assessed for its ability to remain in service and replacement of damaged tubing is recommended.

C.2.12 Gas Side Corrosion

This results from localized or general attack on the tube or fin material from corrosive species such as sulphur from fuel oils, catalytic reduction additives or chlorides from the atmosphere. Damage most frequently occurs at the cold end of the HRSG with surface deposits, pitting or loss of section in the fins and/or tubing. The severity of damage is quantified by ultrasonic thickness measurements on the tubing or by measuring remaining fin thickness.

In theory, damage is acceptable while the tube wall thickness is greater than the minimum specified. However, degradation of the fining can reduce the fin cross-section or corrosion can develop at the fin-tube interface. Both situations effectively reduce heat flow through the fining into the tube and can reduce HRSG efficiency. Repair is normally by replacement with new or upgraded tubing. Fin material can be upgraded or the alloy content of both fin and tube increased.

C.2.13 Acid Dew Point Corrosion

This is a form of gas-side corrosion found when the flue gas temperature drops to a point where sulphuric acid can condense from sulphur oxides and water vapor. As this is a form of low temperature corrosion, it is normally found on finned tubing at the rear of the HRSG or on the casing, ducts or outlet stack. If deposits are found on the finned tubing, these should be removed by water jetting or air blowing, as acid dew point corrosion can continue while the plant is off-line by the absorption of moisture, from the atmosphere, by these deposits (Figure C-10). The underlying metal may have a gouged or orange peel appearance and fins may be missing, corroded or easily broken or pulled off the tube.

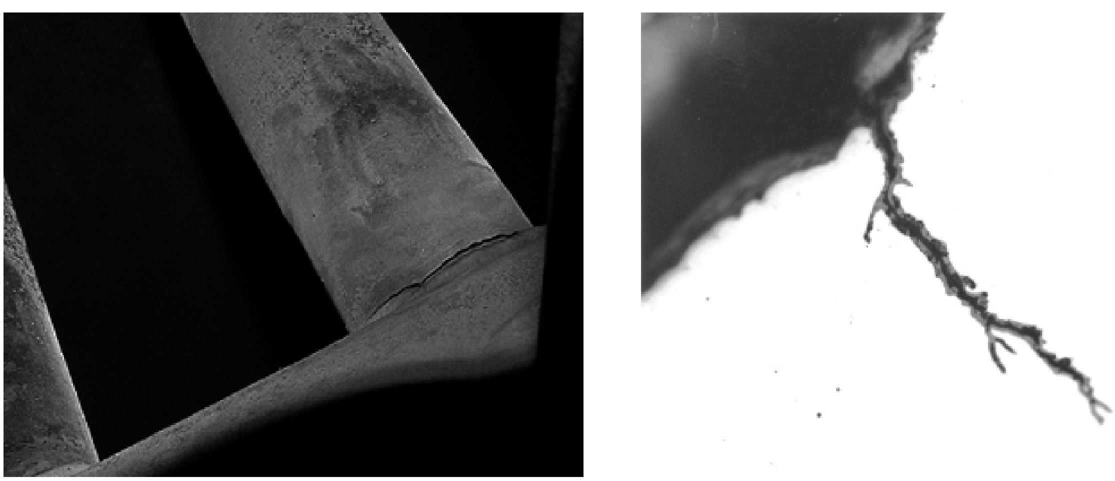
This problem is normally detected by simple visual inspection though ultrasonic thickness measurements may be used, after removal of an area of fining, to determine the extent of tube thinning. The tubing is normally acceptable for further use provided the minimum specified thickness remains, and further dew point corrosion in service can be avoided, but efficiency may be degraded if the fining is badly damaged. Upgrading to a higher alloyed material may offer some protection. At one plant significant sulphate deposition occurred, but metallographic sectioning of tube samples showed negligible corrosion of the tube metal. The preferred policy would be to ensure that the operating temperature is above the dew point, and the HRSG should be kept dry by blowing drying air and/or boxing up during off-load periods.

Figure C-10
Grain boundary hydrogen damage

C.2.14 Thermal Fatigue

This is a form of low cycle fatigue driven by large cyclic stresses, typically above the yield point of the material. These stresses arise from thermal expansions of the structure and produce cracking at the points where this expansion is restrained. Typically, the cracking is also located at stress concentrating features in these areas. While multiple cracks may initiate, one crack soon becomes dominant and grows through thickness leading to failure. Cracks grow perpendicular to the main cyclic driving stress and may be circumferential, longitudinal or bi-directional depending on the local stress field and stress concentration. As cracking is driven by high stresses, other forms of damage, such as corrosion fatigue from the tube bore, may also be present in failures.

Damage is most likely to occur where water or steam tubes are restrained from expansion, especially by stress concentrating weldments at headers, drums and tube attachments. It may occasionally also develop at bends or where there is an abrupt thickness change. As damage develops from the outside, it is easily found by *visual surface inspection* techniques (MPI, DPI). However, sometimes the crack may be filled with oxide, and hence it may not be detected using DPI, and even MPI could have difficulty finding the crack. The crack depth may be assessed by ultrasonic means but, since in many cases the cracking is simply ground out and weld repair applied to return the component to original thickness, this sizing is not always necessary. As the driving thermal stresses are a result of component design, reappearance of such cracking must be expected and further, regular inspection and repair will be required. In some cases minor design modifications, such as the removal or redesign of attachments, may prevent further cracking. Typical cracks are shown in Figure C-11 below. It should be noted that, due to the high operating temperature, the cracks are often rough and oxidized, and bi- or trifurcated crack tips are sometimes present.



Thermal fatigue cracking

Section through of a thermal fatigue crack

Figure C-11
Thermal fatigue cracking at tube to header weld

Smooth bore cracking of steam pipe welds is caused by thermal fatigue arising primarily from steam chilling events. It was first recognized in conventional power plant in the UK in 2001. Following the discovery of the first crack, further inspections showed the problem to be both widespread and potentially severe, with oxide dating showing that the defects typically initiated soon after the plant went into operation. The thermal gradient between the cooler inside surface and the hotter outside surface creates a transitory tensile stress on the inner surface at each chilling event. Cracking is fully circumferential. The cracks appear to grow independently of local microstructure, initiating either at weld roots or at internal changes of section at counterbores, and grow directly through-wall with little deviation of the crack path. They are not associated with creep cavitation. While it is believed that very severe thermal cycles are required to initiate the defects, they will grow as a result of the lower stress cycles that are common in power plant. Detection is difficult and the most reliable inspection is currently an initial conventional ultrasonic inspection, with particular attention to the bore area to detect the cracks, supplemented by Time of Flight Diffraction optimized to provide accurate sizing capability. Final failure will be by some other mechanism, such as fast fracture, creep or creep crack growth. Current effort is aimed at estimating growth rates and critical sizes to maintain cracked welds safely in service and minimize the need for repair.

C.2.15 Creep Fatigue

Creep and fatigue can interact to reduce the life of components. Creep-fatigue cracking is a form of damage that occurs under the influence of large cyclic stresses and operation at a high enough temperature to accumulate appreciable creep damage. Typically, the creep-fatigue cracking problem occurs as a result of cyclic thermal stresses due to rapid heating and/or cooling during start-up / shutdown, combined with operation at high temperature. The thermal fatigue stresses are highest in thick components and the creep-fatigue problem is well known in conventional power plant, where creep-fatigue cracking has been widely found in superheater outlet header boreholes - these remain an area that requires additional inspection in HRSGs.

This high temperature failure mechanism is normally found in steam tubes in high pressure and intermediate pressure superheaters and reheaters. The cracks form perpendicular to the applied tensile stress field but most often occur at stress raisers such as tube-to-header welds. Cracking can be circumferential, axial (in bends) or multi-dimensional at stress-raising features such as attachments, see Figure C-12. The initial section of the crack may be transgranular and dominated by fatigue, changing to intergranular later in growth, when there has been time for creep damage to accumulate, or it may be completely intergranular, see Figure C-13.

As cracking initiates on the external surface it can be detected visually and/or with the aid of surface inspection techniques. A range of other techniques including ultrasonic inspection, potential drop measurement and exploratory grinding can also be used. Replication may be employed to detect early signs. Heavily damaged material must be removed and replaced by weld repair, or complete replacement of the component.

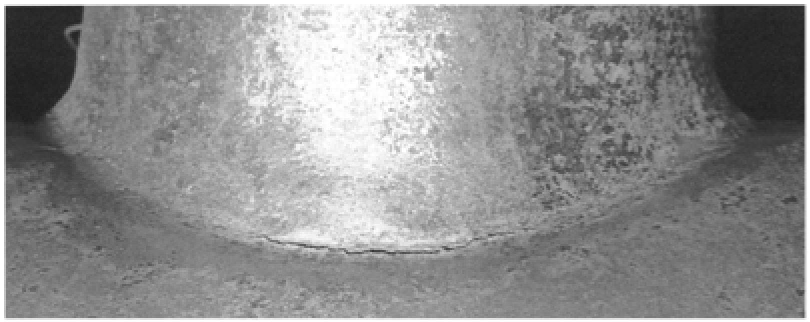


Figure C-12
Creep-fatigue cracking at tube stub

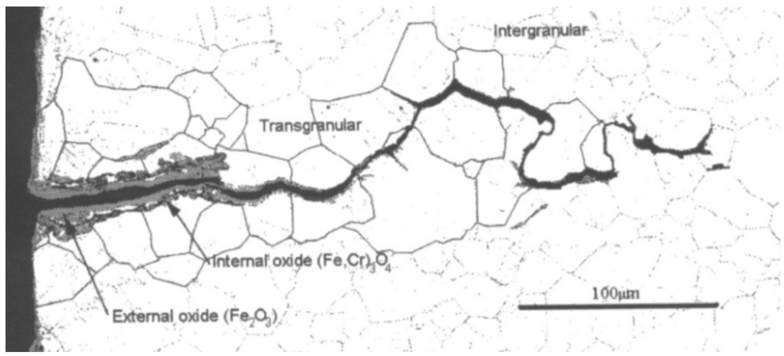


Figure C-13
Section through creep-fatigue crack

Ligament cracking of headers has become a recognized problem with relatively thick 2.25Cr1Mo (P22) headers in conventional power plant (Figure C-14). This is generally accepted to be due to creep-fatigue damage with the fatigue element accumulating as a result of thermal stresses generated by differences between tube bore and header ligament temperatures. The solution has been to fabricate new headers from the more creep-resistant modified 9Cr1Mo steel (P91). This material has higher creep strength and so the header walls can be made thinner, thus reducing the thermal stresses.

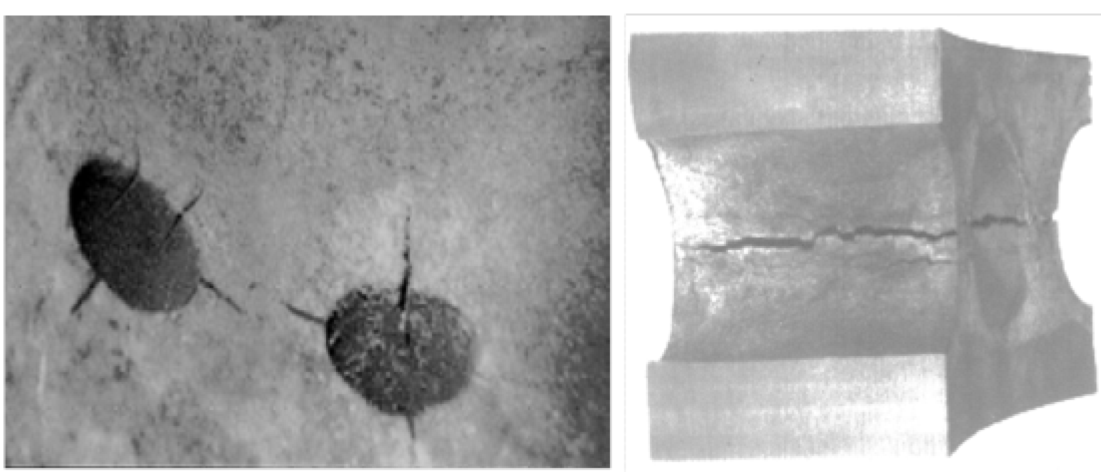


Figure C-14
Ligament cracking in header

C.2.16 Flow Induced Vibration Fatigue

This is a form of high cycle fatigue driven by high frequency but low amplitude stresses; typically well below the yield point of the material. Vibration can result from a number of flow related problems including vortex shedding, and flow eddy currents in the HRSG exhaust and resonance between pressure fluctuations within a tube array. Other sources include acoustically driven vibration and vibration resulting from flow turbulence within the tubes.

Cracking initiates on the outside of the tube perpendicular to the predominant tensile stress. While multiple cracking may initiate, a single crack soon becomes dominant and grows through thickness. These cracks may be axial but are typically circumferential near to tube/header welds. Separate cracks may be present on both sides of the tube. Where tube vibration is responsible, damage due to collision with neighboring tubes may be found on the fins. This kind of damage may be found throughout a HRSG but is most common in high-pressure superheaters and reheaters and adjacent to stress raising welded connections, attachments and at bends.

Damage is often indicated by cracking or impact damage to the fins on finned tubes. Once the fining has been removed, the damage can be confirmed by visual and surface inspection techniques. Ultrasonic, potential drop or exploratory grinding may quantify the depth of cracking, however, it is good practice to remove all cracking and replace with new tubing. The source of the vibration must be determined and avoided either by alterations to the flow or increased bracing/support of the tube bundles.

C.3 Potential Damage to Steam Turbines (ST)

In general, the likely damage for steam turbines in CCGT plant are similar to those for turbines in conventional steam plant. The principal damage mechanisms associated with cyclic operation of steam turbines are low cycle (thermal) fatigue and creep fatigue, erosion-corrosion and environmentally-assisted cracking (in the LP turbine). *Droplet erosion* can be a problem for the last stages of the LP turbines. Low load operation would be expected to increase the number of blade rows that are affected. Cyclic operation may also induce *erosion* of the first three or four blade rows of HP and IP turbines, as a result of enhanced *spalling of steam-side oxide* in superheater/reheater tubing of the HRSG. The effects of turbine blade erosion would be seen in reduced plant output, given correct steam inlet and condenser conditions.

In practice, steam turbines experience fewer problems than the GT and HRSG. Many of the ST problems are associated with the control valves and stop valves. In general, steam turbines do not experience any severe damage at early/ start-up period of the life cycle. Most of the fatigue related damage occurs mainly at 'wear out period'; differential expansion, erosion and environmentally-assisted cracking damages occur generally at 'useful life period' of the life cycle. These damage mechanisms are described below.

C.3.1 Thermal Fatigue and Creep-Fatigue Due to Thermal Cycling

The HP rotor can suffer significant thermal shocks during a start-up. In the initial stages it is important to match the incoming steam temperature to the rotor temperature. This, however, is not as simple as it looks. Measuring the temperature of the nozzle box (stationary blades) adjacent to the first row of blades can give a good indication of the rotor temperature. Software can now estimate the first row blade temperature from steam temperature measurements.

Although the boiler steam outlet temperature may match the temperature of the first row blades, the losses due to throttling through the steam control valves on the turbine can result in a lower steam admission temperature to the turbine cylinder than at the boiler outlet. The throttling losses can be reduced by operating at reduced boiler pressure but this can cause additional problems.

The start-up procedure must balance the requirements of the inlet stage and subsequent stages of turbine rotor. Excessive temperature imbalance and rapid fluctuations on start-up will cause high stress levels in the turbine rotor, blading and shrouding. This can result in various forms of damage to the components.

Thermal fatigue and creep-fatigue can lead to damage to thick-walled components, including governor and stop valves and HP and IP turbine inlet belts. Modern analysis techniques, utilizing finite element methods, are now widely available at reasonably low cost to permit modeling of components perceived to be at risk. Application of this type of modeling, whilst it may not be able to accurately predict the life of components, does provide a valuable understanding of the stress profiles within the component and identifies potential weaknesses and vulnerable areas. Armed with this knowledge, operational procedures can be optimized to minimize the effects of thermal fatigue and inspection procedures can be focused on selected locations at appropriate operating intervals. The addition of temperature and temperature differential instrumentation will enable the operator to minimize the intensity and duration of the adverse conditions. The scope for modifications to existing turbine plant is limited unless new rotors or casings are being fitted. Possible modifications to reduce thermal stresses include improvements to thermal insulation, pre-warming (especially of half joint flanges) and slotting of flanges to increase flexibility. Some operators are known to favour “skin peeling” of high temperature rotors where fatigue damage is of concern. This is basically skimming off a millimeter or so of material from the surface of potentially critical regions of the rotor, e.g. radii, etc. This effectively removes fatigue-related damage and presents what is essentially an “as-new” surface. This process is typically carried out at mid-life.

C.3.2 Mechanical Fatigue Due to Load and Speed Variations

The mechanical fatigue issues arise from two sources. Firstly, during turbine run-up, the rotor passes through a series of critical speeds where vibration levels increase significantly. This is a well-understood phenomenon and the critical speeds are well defined for most machines. Clearly it is important to pass through these speeds as quickly as possible. Over a number of starts, the number of cycles at the critical speeds can accumulate to significant values and subject components such as turbine blades to unacceptable *high cycle fatigue* levels. The most vulnerable area for mechanical fatigue is generally regarded to be the LP blading. Obviously the length of the blades subjects the root area to very high centrifugal stresses. Any defects within this high stress area will cause a significant reduction of blade integrity.

In conventional steam plant, the inspection of disc slots on some older units with large numbers of starts has identified the formation of fatigue cracks in the root serrations where stress levels are concentrated. These cracks are believed to propagate slowly and have not resulted in any major problems, although blade replacements have been required. More modern units have employed advanced design and analysis methods to improve the blade-root detail which should eliminate this potential problem. Where mechanical fatigue problems are encountered with LP blading (or indeed any blading) it is usually possible to re-blade the rotor with a modern design of blade to reduce or eliminate the problem. In some instances, reblading may be combined with improvements in efficiency. Where cracking of the LP blades is present, the cracking may be exacerbated by the onset of *corrosion-fatigue*.

C.3.3 Differential Expansion of Turbine Rotors and Casings

Expansion and differential expansion of the turbine rotor and casing are not usually a problem under cyclic operation, although it is essential to have good turbovisory indication of turbine movement and clearances. Relative movement between the rotor and casing during turbine run-up is always a potentially critical period when rubs may occur on both turbine blade tips and on shaft seals. Where cycling is introduced, it is important to understand what is happening within the turbine from the turbovisory equipment. Each case has to be evaluated on its own merits. Where problems arise, the solution may require changes to operating procedures or increased clearances, albeit at the expense of efficiency.

When monitoring LP turbine differential expansion it needs to be remembered that many LP turbine rotors get longer as they run down (and shorter as they run up) due to the Poisson effect. The centrifugal force generated by the rotation of the blades causes strain in the rotor body perpendicular to the longitudinal axis of the rotor which then results in contraction along this axis which increases with speed. This magnitude effect will vary from design to design depending on many factors. It is best observed in practice. The magnitude needs to be noted. If a turbine is run down with inadequate clearance, then this effect can cause a rub between rotor and stator as the speed decreases. It is important when moving from a base-load situation, where efficiency is of particular concern (and hence the need to minimize blade tip and gland clearances), to review the clearances and make adjustments appropriate to a cyclic operating regime where reliability may be increased at the expense of some efficiency.

C.3.4 Erosion of Turbine Blades

Steam turbine blade erosion may be of two types. Erosion by particulates is due to oxide scales and is found at the front end of turbines. Of greater significance is the risk of erosion corrosion due to water droplets at the back-end of the LP turbine. Off-design operation can lead to increases in steam wetness, which results in more rows of blades being affected. This can cause erosion of the blades towards the tips due to the impact of water droplets on the leading edge of the blades. Stellite shields have been welded onto the leading edges of LP turbine blades to increase the resistance to this erosion. Unfortunately, while the erosion resistance is improved, these shields have sometimes become detached from the blades. An alternative solution would be to employ titanium alloy blades.

Excessive growth of steam-side oxide scales in superheater/reheater tubes, and consequently spallation of the oxide, can be a major issue in conventional plant because the metal temperature can increase significantly due to very high furnace gas temperature and the insulating effect of the steam-side oxide. In CCGT plant, the GT exhaust temperature entering the HRSG is limited to ~600°C maximum (depending on GT type) and hence erosion due to spalled oxide may not be a significant issue because of the relatively slow rates of steam-side oxidation at typical HRSG operating temperatures. However, steam temperatures of modern HRSGs are now as high as 580°C, and even 600°C. It should be noted that *steam oxidation of T91 material could be an issue if superheater/reheater tube temperatures are approaching 600°C.*

C.3.5 Environmentally-Assisted Cracking in LP Turbine

Stress corrosion of turbine blades and rotors is likely to increase with cycling due to greater potential for carry-over from the boiler drum and steam contamination (Figure C-15). Sections of the turbine, which are not normally susceptible to attack since they are operating above the dry-out line, will be operating in hot-wet conditions. Much will depend on the turbine construction. Older designs of rotor using shrunk-on discs would appear to be more at risk than rotors of the monoblock or welded construction type. Accordingly, there could be a plant-to-plant variation, although even with modern equipment some designs may be more susceptible than others. A major factor in governing resistance to stress corrosion is the yield strength of the rotor material.

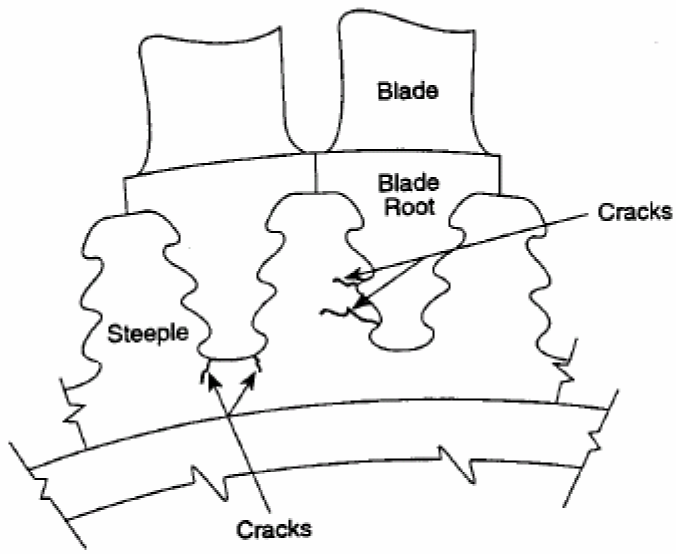


Figure C-15
Axial entry blade attachment showing crack initiation locations

C.3.6 Valves

Clearly, operating in a cycling mode will require increased operation of turbine governor valves and stop valves. Inevitably there will be additional wear and tear on the valve seats and valve stems, especially under throttling conditions when flow-induced vibration can lead to mechanical fatigue and wear. This can, in the main, be contained by redesign of the valve head, modification to the steam flow path and the use of Stellite or similar hard-facing materials on wear surfaces.

During the start-up phase when the throttle valves really are working hard to control the admission of steam to the turbine the valve sealing surfaces are subject to erosion and the steam flow can put significant forces on the valve heads along with the linkages and valve actuators. Wear internal to the valves or in the linkages / actuators can result in lack of precision in positioning of the valve and hence fluctuations in steam flow. These fluctuations can occur either with low or moderate steam flow. Some turbines have two pairs of throttle valves which operate in sequence with one pair opening almost fully before the second pair starts to move. If both pairs suffer from instability then the effect can manifest itself at two points during the start-up phase – one occasion for each pair of throttle valves.

The technique of using two pairs of throttle valves allows better control of the steam flow to the turbine but the second pair of valves to open can suffer rapid changes in steam flow and temperature. Having said this, they may also avoid some of the chilling due to the throttling losses on the first pair of valves during the early stages of start-up. It is not clear which of the sets of valves will accrue the greater damage over the operating cycles. It will be necessary to inspect the valves periodically to determine what damage is occurring.

The sealing glands on the throttle valves will suffer from increased wear with the increased movement of the valve spindles, which can lead to increased leakage or maintenance requirements.

C.3.7 Bearings

Whilst there may be issues with turbine bearings under steady load conditions, cycling puts a lot more stress on them and can raise a number of issues. Whilst the turbine is on barring (slow rotation on turning gear 30 to 100 rpm) during the shutdown period, the oil flow to the bearings will be reduced. There will still be, however, a considerable heat flow along the shaft from the HP turbine, in particular. On barring there is usually only the jacking oil flow. This should be adequate to keep the bearings cool as well as keeping the shaft lifted in the bearings but the bearing temperatures should be monitored to ensure that there is in reality enough oil flow. If the load on the bearings changes as the machine contracts during the shutdown, this can reduce the oil flow to a particular bearing and cause overheating.

C.4 Potential Damage to Electrical Equipment

The key electrical components that suffer from cyclic related issues are alternator, generator, rotor, transformer etc. Flexible operation puts alternators through thermal cycling in a similar manner to boilers and turbines, although clearly the temperatures involved are much lower in alternators than in many other areas of plant. The potential damages experienced by key electrical components under cyclic operation are summarized below.

C.4.1 Generator Windings

Movement of the rotor windings may occur under the combined centrifugal and thermal stresses. There is a ratcheting effect of differential expansion between the copper windings and the rotor end rings, and similarly between the copper windings and stator core, leading to breakdown of insulation and movement of windings. Differential winding movement or blockages of cooling ducts can then cause the rotors to take up a slightly bent condition that is dependent on the rotor winding current. The electrical windings suffer increased damage due to cyclic operation, thereby cracking the insulation and loosening the wedges holding the coils. This is a long-term effect.

C.4.2 Rotor

There is potential for thermal fatigue of rotor forgings, especially at section changes, e.g. winding slots. With periods of prolonged steady load the rotor windings and the retaining bars which hold them in the slots can become stuck in one position. When reducing load they can take some time to free off and move as the thermal differential expansion between them and the rotor body would determine. Equally after a period of shutdown when returning to full load they can stick as they move to the fully warmed-up position. This phenomenon is often referred to as "stick/slip" because the component initially sticks but will ultimately slip. When the bars / windings are stuck and under tension trying to expand or contract, if this does not occur equally in both halves of the winding (assuming a two-pole rotor), then the rotor behaves somewhat like a bi-metallic strip and tends to bend. This will produce an imbalance in the rotor leading to vibration.

Even if there is no excessive "stick" during the expansion and contraction, the cycling will inevitably result in some wear on the components in the alternator rotor. There is a risk if components such as packers gradually migrate due to the repeated expansion and contraction that eventually a cooling slot make suffer some blocking. If the blocking becomes significant enough it may result in uneven cooling of the rotor body. This can then cause a thermal bend of the alternator rotor, resulting in higher than normal vibration. The magnitude of the vibration will vary with the excitation current being carried through the rotor windings. The effect can be a little confusing because as the current increases the bending may, initially, reduce the overall vibration level experienced. This can then be followed by an increase in vibration as the current rises further. The exact behavior depends on the phase (angular position) of the natural imbalance of the rotor compared with the phase of the thermal bending.

To fully diagnose this type of problem, a computerized vibration analysis system should be used as this will be able to determine the magnitude and phase of the alternator vibrations and hence calculate the vector change.

During the run-up and run-down, the alternator rotor will pass through its critical speeds at which vibration levels will increase significantly. Most alternator rotors have asymmetrical characteristics due to the slots for the windings, and have two critical speeds. It is important that these are known to the operators and that these speeds are passed through quickly on run-up and run-down. Dwelling in the critical speed ranges will subject both the rotor and stator to unnecessary vibration-driven damage.

The material properties and machining quality of modern generator rotor shaft and end rings should ensure that they are not adversely affected by cyclic operation. Some rotor designs have suffered from copper dusting whilst the machine is barring. This phenomenon is caused by the individual turns moving radially in the slot under the influence of gravity. The copper rubs against the slot insulation leaving deposits and possibly leading to an inter-turn fault.

C.4.3 End Rings

There is potential for stress corrosion cracking of the rotor end rings due to moisture when the unit is cold. Concern has grown about end-ring integrity, despite the introduction of the Fe-18Mn-18Cr end-ring alloys. If the end rings are of the older Fe-18Mn-5Cr type alloy, then these should be inspected prior to cyclic operation, if there has been any suspicion of moisture contamination, because of the threat of stress corrosion cracking. The newer Fe-18Mn-18Cr alloy has much better resistance to SCC but it may have poorer corrosion-fatigue performance. Hence the rings should be kept dry at all times. How practical this is with modern generators is open to question since many of these are air-cooled rather than hydrogen-cooled. In order to minimize the risk of stress corrosion cracking it is vital to keep the dew point of the coolant gas within the alternator as low as possible.

The stress variations which result from cycling are important, although variations in the electrical output can also induce fatigue, even though there is no change in the rotor speed. This is due to eddy current induced heating in the end ring. Older designs of end ring were more susceptible to damage, since they were prone to high cycle fatigue. One of the older variants used retaining rings that were shrunk onto the end disc, which itself was shrunk onto the shaft. Here relative movement between the end ring and the end windings occurs with each revolution. The other design can induce cracking in the shaft, in the rotor tooth region and in the end ring itself. Here the ring is shrunk on to the end windings and the shaft. The more modern design, which is less susceptible to high cycle fatigue, is of the so-called "cantilever type". Here the end ring simply grips the alternator rotor, and does not rely on support from the shaft. Another shortcoming of older designs was that the fixing of the end ring to the rotor was of a relatively simple form. Modified designs are available to reduce stress concentrations in this area.

C.4.4 Stator

Leaks may occur in stator water cooling connections inside the alternator due to thermal cycling and high vibration levels on run-down / run-up. The repeated thermal expansion and contraction caused by cycling can cause damage over a long period of time in an alternator stator. Damage to the windings (particularly the end windings which overhang the end of the stator core), looseness of wedges, and fretting of the laminations in the alternator core are often caused by long-term operation at high vibration levels. It is generally the number of cycles accumulated over time which results in damage. However, it is possible for the thermal expansion and contraction of cycling to initiate or exacerbate problems.

Thermal cycling of the generator stator can lead to the development of partial discharge phenomena in the slot region. This is caused by expansion/ contraction of the conductor bars relative to the core, degrading the semi-conducting coating of the bar. This is a long-term issue and it is possible to detect this type of degradation by the use of on-line discharge monitoring equipment.

Another potential effect of thermal cycling is that stator slot wedges may become loose. Again, this is a long-term issue. It would not be expected to become a problem before the first scheduled removal of a machine's rotor and so the wedge tightness can normally be checked then and a suitable strategy formulated. Systems are available to perform wedge tightness checks with the rotor installed.

Fretting within the stator core can result in breakdown of the insulation between the laminations which can produce hot spots within the core. High vibration or stator windings not sliding correctly within the slots can result in damage to the insulation of the winding or weakening of the mechanical strength of brazing or other joints within the windings which can result in leaks.

The stator windings are water-cooled in a lot of machines. This usually involves circular manifolds within the alternator with PTFE (Poly TetraFluoro Ethylene) hoses feeding the water from the manifolds into the individual winding. There are, therefore, seals at each end of every hose sealing the PTFE to the metal. The different thermal expansion characteristics of the PTFE and metal mean that repeated load cycling will stress the seals. Usually the seals contain 'Viton' o-rings which help to absorb the movement. If the rate of operation of this system, or venting as it is often called, increases significantly, then it is an indication of a stator water leak which can often come from a cooling hose seal.

C.4.5 Transformers

Increased wear and tear would be expected due to stop/start operation. During steady state operation, transformers stay warm and at a relatively constant temperature. The move to flexible operation will completely change this. Depending on the location of the transformers (many are located outside to improve cooling in hot weather and due to their sheer size) and the ambient temperatures, transformers can cool significantly when off load. This cooling will cause the transformer internal oil and air volume to decrease thus sucking in air from outside. It is important that the breather systems are functioning correctly. If blocked, then the transformer tank can be put under suction and air may leak in other than via the breather. If the breather is not blocked but its desiccant crystals are exhausted, then damp air can be drawn into the transformer thus degrading the performance of the transformer oil. Regular monitoring of the breather and oil condition will ensure that this is not happening.

Transformers can suffer similar winding problems to alternator rotors and stators due to thermal expansion and contraction.

Transformers are often equipped with tap changers. These are switches to change the turns-ratio between the two windings of the transformer. During base load operation these may operate over a limited range, for example to adjust the reactive generation from the alternator to the grid system. On shut-down and start-up the tap changer may be operated over a much bigger range. It may be that the mechanism is in poor condition at the extremes of its range. The mechanism may be stiff or the electrical contacts in a poor condition. As a result, it may be necessary to carry out more inspections. Monitoring the condition of the oil in the tap changer compartment, which is often separate to the main transformer oil, can help to identify arcing from the electrical contacts.

C.4.6 Electric Motors

The number of starts per year imposed by cyclic operation should not normally adversely affect motors. However, the problem of rotor bar cracking on large motors has been encountered worldwide following a move to this type of running regime. Abrasion of the stator coil insulation at the slot emergence has also been experienced. This has been sometimes linked to the number of starts.

The repeated high starting current and torques put stresses on motor stators and rotors. This can cause rotor bar cracking. Often it is the connection (often brazing) of the rotor bar to the shorting ring at the end of the rotor bar which cracks rather than the rotor bar itself.

It is possible to analyze the waveform of the supply current to the motor to identify if there are any cracked rotor bars. This technique is known as *Motor Current Signature Analysis* (MCSA) or *Motor Current Analysis* (MCA). Cracks in the rotor bars result in harmonics in the current waveform which can be detected and measured.

C.4.7 Coolers

The coolers (hydrogen or air) also suffer from increased long-term failure. Cracking of the cooling tubes, near the tube plates of the usual shell and tube type of heat exchanger, results from the heating and cooling of the generator. The temperature control valves also tend to stick in one position, generally overcooling the generator in the off-load condition, causing condensation to take place in the windings (air-cooled).

C.4.8 Switchgear

Increased switching operations give rise to increased wear and tear. Switchgear can suffer from mechanical breakage of the operating mechanism. This has serious consequences for the generator circuit-breaker in particular, and for essential auxiliaries such as emergency lube oil pumps, etc. The mechanical linkages and switchgear might not be designed for frequent operations as the plant may have been designed for base load operation. These breakages are usually due to fatigue of the metal part. Moisture and dust ingress can occur during shutdown periods.

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3420 Hillview Avenue, Palo Alto, California 94304-1338 • PO Box 10412, Palo Alto, California 94303-0813 USA
800.313.3774 • 650.855.2121 • askepri@epri.com • www.epri.com