Single Electricity Market

Market Monitoring Unit

Power Plant Cycling

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Index

1 Introduction

This report is prepared by the Market Monitoring Unit (MMU) and is aimed at informing industry on issues of power plant cycling, one of the foremost concerns of thermal power generation owners operating in the SEM. Much of the work presented in this report is derived from a recent report produced by European Technology Development Ltd (ETD) for the MMU.

The MMU is one of the Joint Management Units of the Regulatory Authorities (CER and NIAUR).

Disclaimer:
Information provided in this report (re: Power Plant Cycling) and the accompanying Appendices are presented for the benefit of power plant operators and other interested parties. While the Regulatory Authorities (RAs) feel this information is of benefit to market participants, the advice and views contained in these sections are not necessarily those held by the RAs and should not be treated as such.

2 Power Plant Cycling

Information in this section is largely derived from a recent report produced by European Technology Development Ltd (ETD) for the SEM Committee. Significant portions of this report are not covered in this document due to the confidential nature of its content.

2.1 Background

Because of changes in demand and competition in power generation markets, many generation plants around the world are now subject to cycling operation (described below). Since the introduction of the SEM, cycling has become a prominent issue on the island of Ireland, with some generation plant experiencing more intense cyclic operation than before. A major SEM Committee Inquiry into the bidding activity of some Participants looked at how the costs associated with cycling should be accounted for within Commercial Offer Data (amongst other issues). The final decision on this Inquiry was published in June 2008 (SEM-08-069).

Going forward, the anticipated growth of intermittent renewable generation is expected to have an impact on the amount of cycling experienced by thermal plant in the SEM.

2.1.1 Technical Definitions

In this report ‘cyclic operation’ or ‘cycling’ are wide-ranging terms that cover the following:

- **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
- **Load-following** in which the plant is on for more than 48 hours at a time but varies its output to follow the daily pattern of electricity demand
- **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
- **Weekend shutdown** in which the plant shuts down at weekends. This is often combined with load-following and two-shifting
- **Sporadic operation** for periods of less than two weeks followed by shutdown for more than several days.

There are several different types of thermal plant operating in the SEM. The following gives an overview of some of these.

- **Conventional Steam Power Plant (Rankine Cycle)**
  These consist of a boiler in which water is evaporated to form steam, that is then superheated. This superheated steam is passed through a steam turbine, which turns an alternator that generates the electricity. The boiler may be coal, oil or gas fired, or it may use a combination of fuels.

The majority of coal, oil and non-CGT gas fired generators in the SEM use a Rankine cycle, these include Moneypoint 1, 2 & 3 (coal), Aghada Unit 1 (gas), Tarbert (oil), Poolbeg 1 & 2 (gas), Ballylumford 4, 5 & 6 (gas) and Kilroot 1 & 2 (coal/oil).

- **Open Cycle Gas Turbine (OCGT)**
  In an OCGT, air is compressed in a compressor, fuel is burnt in a combustion chamber and the hot product gases are passed though a turbine which is used to drive an alternator. OCGTs are relatively inefficient but are very flexible and designed for cyclic operation.

  OCGT units in the SEM include Kilroot GT1 & 2, Rhode 1 & 2, Ballylumford GT1 & 2, Tawnaghmore 1 & 3 (all distillate) and Aghada CT1, 2 & 4 (gas).

- **Combined Cycle Gas Turbine (CCGT)**
  A CCGT consists of at least one gas turbine which drives an alternator (similar to the above). The high temperature exhaust gases from the gas turbine are passed through an HRSG (Heat Recovery Steam Generator) which uses the waste heat to generate steam. This steam is used to drive a steam turbine which powers a second alternator. The use of the waste heat from the gas turbine to generate additional electricity means this type of plant has a relatively high efficiency.

  CCGTs in the SEM include Ballylumford 10, 31 & 32, Poolbeg 4, Huntstown 1 & 2, Coolkeeragh, Dublin Bay and Tynagh. The new Aghada and Whitegate units due to enter the SEM in 2010 are also CCGTs.

2.2 Damage Caused by Cyclic Operation

When a generator unit is cycled, the boiler, steam lines, turbines and auxiliary components undergo large thermal and pressure changes. These cause stress damage. This section covers some of the issues that cycling operation presents to thermal power generators.

2.2.1 Creep

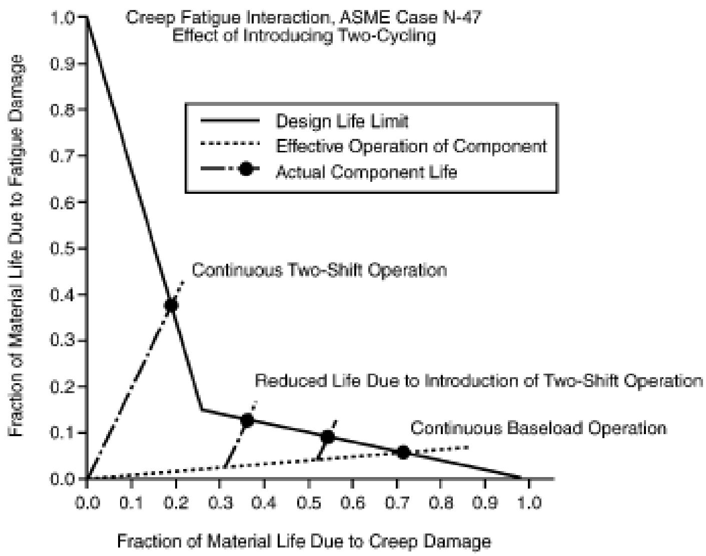
If a unit was designed to operate in base load conditions (where the unit generates at or near its maximum output for sustained periods), creep conditions can arise. Creep is the time-dependent change in the size or shape of a material due to constant stress, or force. In thermal power plant, creep is caused by continuous stress that results from constant high temperature and pressure in a component. Units that are designed for base load operation are constructed of materials and components capable of withstanding creep. Because creep is both time and temperature dependent, two-shifting and low-load operation would be expected to reduce damage caused by long-term creep, although this may not always be the case.

2.2.2 Creep-Fatigue

Fatigue is a phenomenon which leads to failure in a material when it is subjected to repeated, varying levels of stress. In thermal power plant, such stresses result from changes in pressure and temperature, which typically occur during cycling operation. Resulting damage is often exacerbated by creep-fatigue interaction, in which creep and fatigue act together to cause premature failure.

Figure 7 below illustrates the interaction between creep and fatigue. The limit line (solid line) represents the design life limit, expressed in fraction of material creep life and fatigue life for a particular grade of steel (2.25Cr1Mo). This limit line is used to establish the effect of combining creep and fatigue. Although the line is only a representation of the phenomenon, it demonstrates the effects. Different materials will have different creep-fatigue curves.

Figure 7: Creep-Fatigue Interaction



The length of the dotted lines indicates the relative life of components that are exposed to a combination of both creep and fatigue. Components that are exposed to just one (or predominantly one) type of stress damage tend to last longer. Those exposed to a combination of both tend to fail sooner.

Base load components experience dominantly more creep than fatigue, and this is illustrated in the graph. When a base load component is exposed to increased fatigue part way through its 'life' via a change to cycling dispatch, the expected time to component failure can be rapidly shortened.

Creep-fatigue interaction is of particular interest for a number of components, including cracking of thick-wall components; superheater and reheater header ligament cracking; evaporator header stub cracking; and damage to economiser headers, feedwater heaters and tube ties.

2.2.3 Expansion Related Issues

As a material heats up it expands and as it cools down it contracts. This thermal movement is considerable in power generator components as the temperatures involved are significant. A number of components are adversely affected by these thermal movements, including, boiler structures, pipework systems, and turbine rotors and castings.

2.2.4 Other Potential Issues

Other potential issues associated with two shifting include corrosion and fouling of components, and wear and tear of pumps and other auxiliary equipment. More detail on these and the issues highlighted above is included in Appendix A.

3 Cycling in the SEM

Since the introduction of the SEM in November 2007, operators at many power stations realised that the plant's mode of operation was likely to change due to increased competition with the consequent introduction of high efficiency CCGT plant, new market structures, and increased penetration of wind farm generation on the island.

Figure 8 show the number of starts for SEM registered thermal units during the period 2007 to 2009. Note that for the first ten months of 2007 (until the SEM came into operation), these units were dispatched under different rules and not on an all-island market basis. This graph shows that the quantity of starts post-SEM has increased in some quarters and decreased in others. In Quarter 4 2007 there was a sharp increase, as the new market led to a number of participants being heavily cycled. After this 'bedding–in' period the total number of starts appears to have decreased, possibly influenced by higher start costs and lower demand. In winter 08/09, the total number of starts increased, this is likely to have been driven by a higher demand over the winter period and also a relatively high level of wind generation over this period.

Fig. 8: *Total Starts of SEM Registered Thermal Units (2007-2009)*

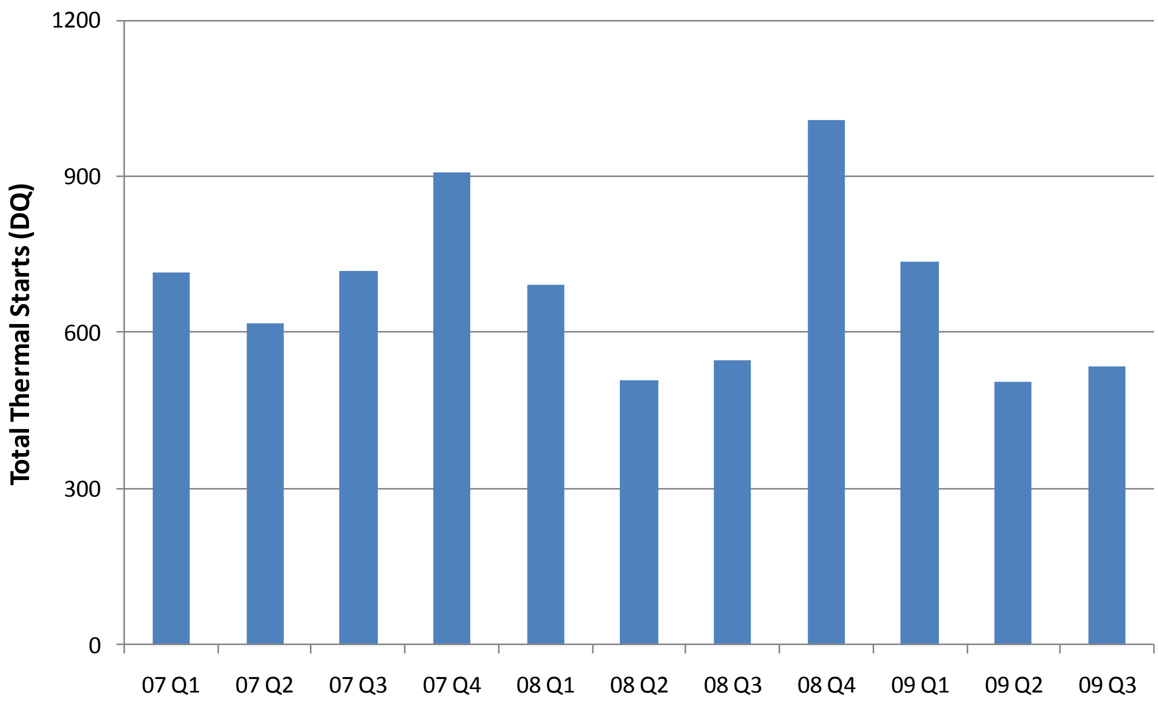
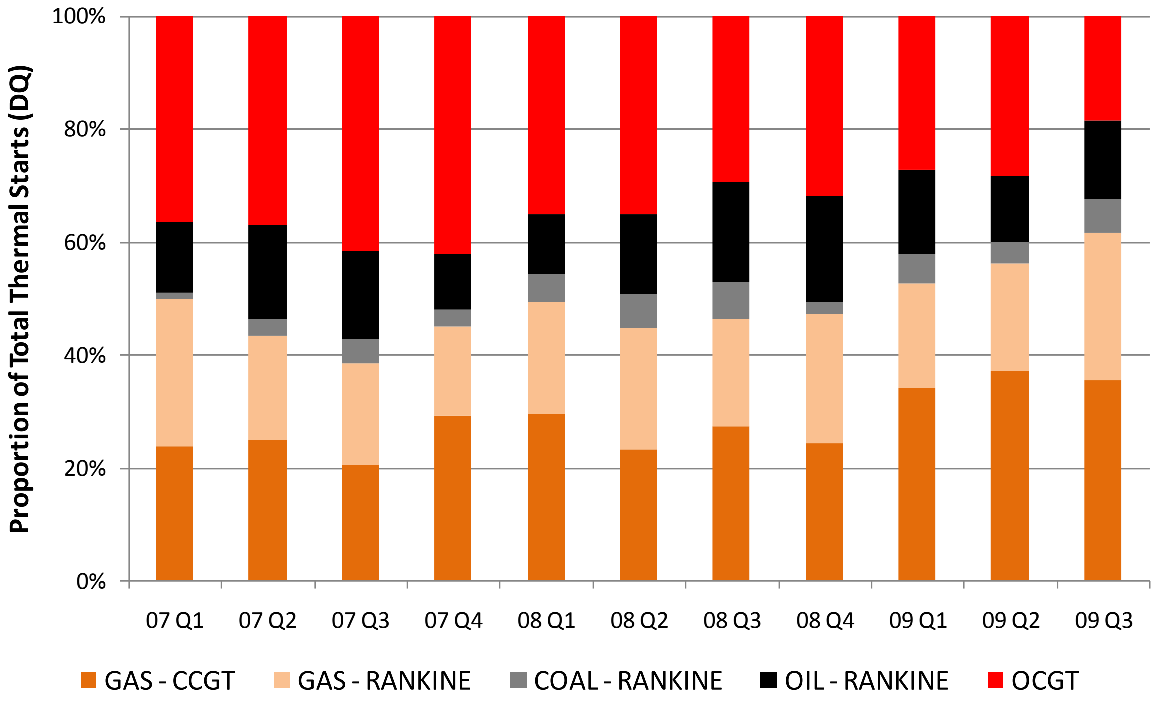


Figure 9 shows the total number of thermal starts proportioned into fuel/technology type. As can be seen, a decreasing proportion of starts was incurred by OCGT units (mostly distillate fuelled), while CCGT and Rankine cycle technology saw a general increasing proportion of starts.

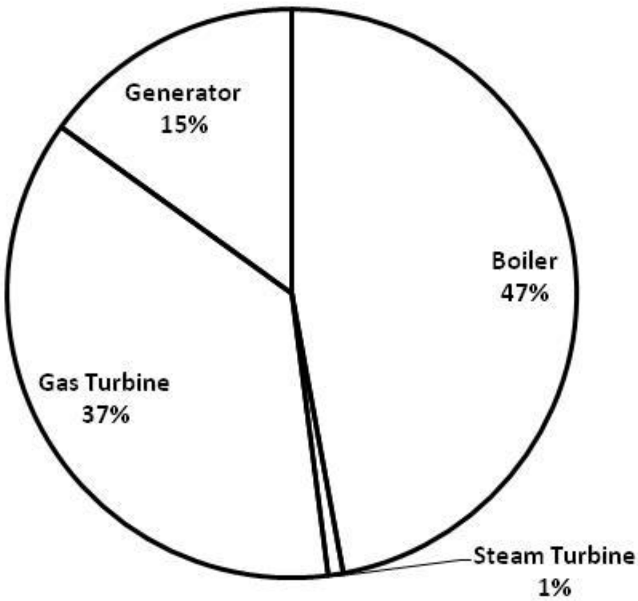
***Fig. 9: Proportion of Total Thermal** Starts by Fuel/Technology (2007-2009)*



3.1 Modification costs of the SEM plants

Some SEM units which were originally designed for base load have changed to two-shifting operation. In some cases, operators have incurred modification costs upgrading their plant in order to accommodate this change of running regime. As part of the ETD study, questionnaires were sent to market participants requesting information on individual units. Although this data is used to inform this report, the majority of this information is confidential. The makeup of modification costs for the majority of SEM registered Steam Rankine and CCGT units for the period 2005 to 2008 are shown in Figure 10.

Fig. 10: *Makeup of modification costs for Steam Rankine and CCGT units*



From data submitted in the questionnaire responses, the auxiliary systems and staff costs were the principal areas of investment. However, more detailed analysis shows that these two investment categories are principally accounted for by cost data submitted for a small number of units. If staff and auxiliary system costs are regarded as 'outliers' from the general investment trend, then from the modification cost viewpoint, the two critical parts of a power plant are the boiler (conventional and HRSG) and the gas turbine. This is illustrated in Figure 10. Perhaps the most striking aspect of this analysis is the low investment in steam turbine modifications. The reason for the lack of investment in modifications may be because the Steam Turbine is a relatively well understood item of equipment and remaining life assessment has reached a high stage of development, thus permitting replacement of a component practically at the end of its life. This lack of initial investment in modifications may be also due to the high costs of the steam turbine components, which means that earlier replacement/modification is an expensive option that might involve significant cost.

A change from base load to cyclic operating conditions can potentially result in:

- Increased capital spend for component replacement
- Increased routine O&M costs as a result of increased wear and tear
- Lower availability due to an increase in failure rate and increased outage time (with associated replacement costs)
- Increased fuel cost from reduced efficiency and non-optimum heat rate.

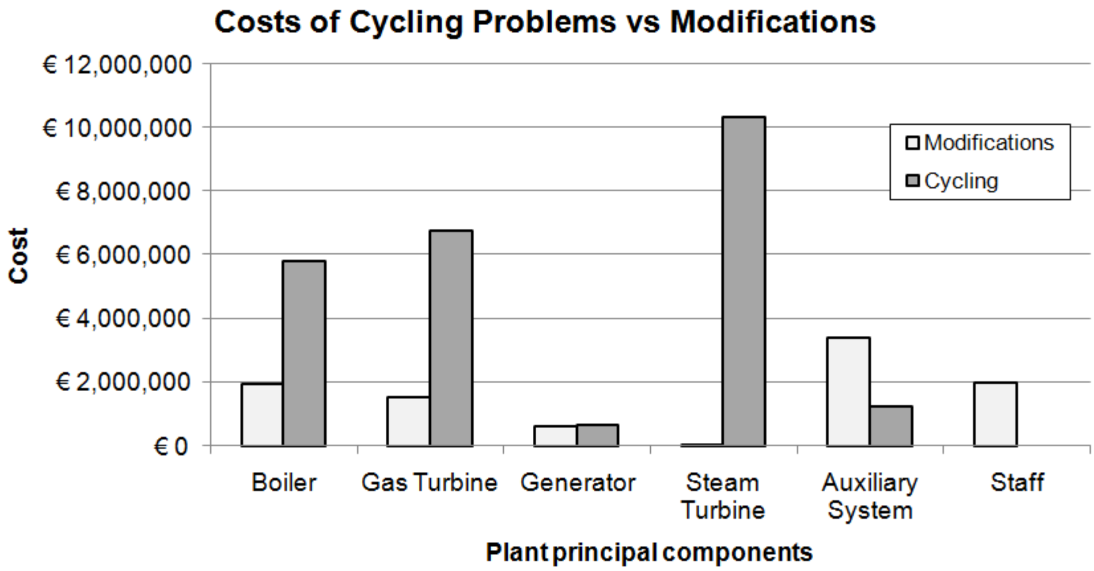
The difficulty lies in establishing the degree to which each of these factors will be applicable to any specific plant. The plant specific factors that can be influential include:

- The overall design of the plant - drum type or once-through boiler and the layout, e.g. bypasses, etc.
- Individual design of major components – material, thickness, etc.
- The way the plant is operated
- The quality of the water chemistry
- The size and age of the plant
- Previous maintenance quality and philosophy.

Each of these will influence or even determine the degree to which cycling will impact on the overall O&M and capital costs.

The ETD study analysed and compared the costs of modifications implemented in order to reduce the impact of cycling with the costs of cycling-related problems. This comparison of the costs for the majority of Steam Rankine and CCGT plants in the SEM is shown in Figure 11. For the principal plant components (boiler, GT, ST), it can be seen that the total of the reported cycling-related costs is higher than the financial investment made in modifications. The on-going costs of cycling for the conventional boiler/HRSG are related to damage and failures of the casing, baffles, tubes, etc, as a result of differential thermal expansion and contraction during operational cycles. The GT also experiences cycling-related costs due to failure of blades in the compressor and turbine sections, and other failures in the combustion chamber. In the case of the Steam Turbine (ST), the cycling related costs are much greater than the initial investment in modifications, which was very low compared to the other equipment of the power plant.

Fig. 11: *Comparison of costs of modifications with the costs of cycling-related problems*



In general, a relatively small amount is spent on cycling related modifications than is incurred though cycling related problems. This is particularly stark for the ST.

3.2 Overview of Plant Problems and Costs

HRSG tube failures

HRSG tubes provide the means for extraction of useful energy from the waste heat in gas turbine exhaust at combined cycle power plants. HRSG tube failures are an infrequent occurrence at most units. Typically, there is less than one failure per year. However, the cost associated with unexpected outages encourages efforts to anticipate problems. Like boiler tubes in radiant boilers, HRSG tubes are subjected to a variety of service-related damage. Damage mechanisms vary in severity based on materials used, local operating stresses and the interaction between vibration and corrosion. Because HRSGs have operated at generally lower metal temperatures than older radiant boilers, creep and creep/fatigue have not been common failure causes. Instead, corrosion fatigue, flow accelerated corrosion and damage at locations with weld defects have occurred in many units. Other corrosion mechanisms such as pitting and acid dew point corrosion of cold end tubes have also contributed to premature HRSG tube failures.

Gas Turbine Modifications

Combustion/Gas Turbine (CT) units on early CCGT plants used uncooled 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. For the past ten years 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. Developments in blade material to cope with CT inlet temperatures of around 1400ºC improved the components' ability to cope with cycling and the associated thermal fatigue.

Generator Modifications

The main change in this has been a move towards utilising air cooled alternators in CCGT plant where there is a strong drive to reduce manpower and ancillary equipment. Alongside this have come the usual financial pressures to reduce capital costs. Consequently, these designs are being pushed to their limit. Some manufacturers are making full use of modern insulating materials, which permit hot spot temperatures to reach 155°C as against 140°C for older materials. This increases the thermal cycling of the generator assembly as a whole compared to earlier designs. The materials of construction comprise of mica-strengthened polymers, copper and high strength steel, all of which differ from each other in expansion coefficient, thermal conductivity and resistance to creep. The effect is to reduce insulation life, under cycling conditions. A major threat to the generator life also comes with the introduction of compressor air cooling to boost output during hot weather. Although the generator output may be within specification, cooling will be poorer than the designers will have anticipated and with some models, deterioration of polymeric materials and thermal fatigue of metallic components will be more rapid.

Steam Turbine Issues

Temperature differentials during start-up and bowing of the rotor can lead to blade rubbing, but this can usually be prevented by appropriate start-up procedures. The main problems which will occur, even with careful operations are those caused by thermal fatigue and simple cyclic fatigue. In general, cracking caused by thermal fatigue should be monitored rather than repaired. Cyclic fatigue will occur due to the centrifugal stress build-up during start up and also because rotors and blades will need to go through a series of critical speeds in which stresses are high.

> Appendix A includes some more detail provided by ETD on the potential damage associated cycling operation.
>
> Appendix B contains a best practice guide for cycling, produced by ETD.

APPENDIX A: Potential Damage Caused by Cycling

A1. Cracking of Thick-Wall Components

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 as a result of through-wall temperature differences during start-up and shut-down. These heavy section components are often produced as castings and tend to be thicker than the forged equivalents. Despite this, these components are generally regarded as being more tolerant of thermal transients.

Thermal fatigue cracking tends to be focused at stress concentrations, such as at changes in geometry or section (which result from the casting or forging process) or where subsequent machining has led to poor geometry (for example, grooves for valve seat placement). Fortunately, this type of cracking is fairly innocuous. It generally propagates to less than 10 mm depth and then stabilizes with little further growth. In new components, the tendency for cracking can be greatly reduced by avoiding sharp corners and using high-quality castings.

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 these cases, it is probably best to leave the defects in-situ and to monitor their growth because continued machining and repair welding may be more damaging in the longer term. The problem needs to be managed through a programme of routine inspections and planned replacement and repairs at scheduled outages.

Many boiler stop valves fitted to older boilers were not designed with regular thermal cycling in mind and suffer the effects of thermal cycling, resulting in cracking of valve bodies, seats, and disks. 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 steel, which significantly reduces the overall wall thickness and has thus been expected to reduce tendency to cracking. Where the cost of these valves is unjustified, an alternative is to have a set of spares that can be replaced during a planned outage and subsequently refurbished on-site for use at the next outage.

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.

Considerable header ligament cracking was first observed in the United Kingdom in the mid-1980s and has now been observed worldwide. Extensive investigations were performed by the then-CEGB to gather data on operational stresses and temperatures, followed by finite element computer analysis. The problem was attributed to poor design and manufacturing detail combined with poor header temperature control when two-shifting. Cracks had initiated and propagated to more than 50% wall depth in as few as 300–500 starts. It was possible to make a safety case to continue operating the units until a more permanent repair of the ligaments or header replacement could be made. The case assumed that cracking would not have initiated until some time into the operating life of the component. It was also assumed that the crack growth rate was reasonably constant, so that the component could survive a limited number of cycles before the crack reached a critical depth. This depth was accepted as 70% of wall thickness for circumferential cracks if the longitudinal stresses were relatively low.

In the United Kingdom, a number of headers have been replaced since the late 1980s when the problem first became apparent. In many cases, header replacement was associated with upgrading of materials, for example, from 2.25Cr1Mo to P91 steel.

The susceptibility of any header to ligament cracking is a function of its wall thickness and the spacing of stub holes, material of construction, and operating conditions. The superheater outlet headers are at greatest risk, 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. Installation of thermocouples has shown that these headers can be subject to high temperature swings at start-up. Inner wall temperatures are directly influenced by steam temperatures. Thus, a superheater header that normally operates at HT (>500°C) and is well-insulated may retain HTs on shutdown for a short period. On start-up, if condensate forms in the tube sections, or if saturated steam (usually at about 360°C or even lower if operating on a sliding pressure regime) is admitted to the header, rapid cooling will occur and could result in a temperature difference of up to 200°C.

Because of their relatively thin wall construction, reheater headers are at low risk of ligament cracking. Crack initiation and growth are driven by temperature transients, which may occur as follows:

- During hot starts, when condensate (formed in platen elements during the shutdown) is passed into hot headers, resulting in a rapid quench
- During start-up, when hot steam is admitted to relatively cold headers, causing a rapid change in temperature
- When attemperator sprays are used without adequate control, resulting in excessive water carry-over and quenching of hot surfaces downstream of the attemperator
- During rapid de-loading associated with forced cooling for tube repairs, when saturated steam can be carried over from the drum into the final superheater sections
- During block loading, this may impose sudden changes in air-flow through the boiler and changes in steam flow.
- If cracking is present, continued operation may be justified based on the following:
- The extent of cracking
- Any previous inspection history, enabling estimation of crack growth rates
- The likelihood of the cracked ligament resulting in a leak rather than rupture
- The consequences of failure and hazards arising
- A programme of inspection based on predicted crack growth rate
- Monitoring of temperatures, including through-wall temperatures
- Structural assessment
- Changes to operating practices to reduce thermal stresses.
- The outcome of this assessment will determine the actions to be taken. Options include the following:
- Continued operation in the presence of cracks with an associated programme of monitoring and inspection
- Local repair of the cracks in the header (suitable only for isolated cracks and 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.

Ligament cracking is now well-understood and has been greatly reduced by improved header design to reduce stress concentrations and improve ligament efficiency (see Figure 4-3). Improved efficiency includes rearrangement of stubs in a diagonal format, application of manufacturing controls to ensure correct tolerances and alignment of apertures, and eliminating sharp edges and radii.

Improved Header Ligament Design

Cracking can be avoided or reduced by implementing the following:

- Improving boiler operation to avoid severe temperature excursions and to drive to pre-set criteria identified previously. This is achieved by monitoring temperatures at inner and outer walls and header inlet and outlet stubs to establish actual rates of temperature rise (or quench) and variations in both through-wall and diametric temperatures. Analysis of these variations, usually using finite element (FE) modelling, can be used to estimate the stress ranges generated and define acceptable work limits.
- Using stronger materials such as P91 (9% Cr martensitic steel), resulting in the use of thinner sections. However, research findings show that P91 may be problematic in terms of its vulnerability to accelerated cracking under creep/fatigue interaction.
- Applying routine inspection techniques using a combination of internal closed circuit television (CCTV), intrascopic visual examination, or external ultrasonic examination. Because on-going plant operation may lead to cracking, an inspection strategy must be adopted to monitor the situation. Typically, this strategy would comprise CCTV inspections of the bore of the header with some external non-destructive testing (NDT) using ultrasonic and magnetic particle methods.

Evaporator Header Stub Cracking

During boiler light up, the expansion of furnace wall tubes may not be uniform across the boiler. The centre tubes are more exposed to firing and expand more rapidly than the wing tubes. If the tubes are connected to rigid headers at the top and bottom, the differential expansion must 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 generally concentrates 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 waterwall headers and economizer headers.

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

- Performing regular inspection and local repair during planned outages. This is a low-cost option but does not guarantee the elimination of tube failures.
- Replacing stubs with stronger (thicker) stubs able to accommodate the higher stresses. This may provide a short-term palliative solution but does not necessarily guarantee long-term integrity.

- Modifying the tubes to incorporate greater flexibility (for example, by installing expansion loops). This is the preferred option.
- Replacing long headers with shorter interconnected box headers. This is the most expensive solution.

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

Economiser 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 feedwater are injected into the boiler as flow is established. Although thermal fatigue is encountered during baseload operation, the magnitude of cracking can increase dramatically. Options include replacing or repairing the header or recirculating some of the water from the steam drum into the feedwater line to raise the inlet feed temperature.

Feedwater Heaters

The temperature range experienced by feedwater 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 two-shift 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. The integrity and life usage of these bolts should be monitored with a clear policy on bolt replacement. Other potential problem areas include thermal fatigue cracking of baffle and division plates. Modern feedwater heaters use integral distribution headers, which are inherently thinner section and less vulnerable to thermal fatigue problems.

Tube Ties

Tube failure attributed to attachment weld failure is one of the main areas of reduced availability when boilers are two-shifted. Boiler tubes are held in position by attachment either to adjacent tubes by means of slip ties on platens or by brackets to cold steelwork on furnace walls. Under two-shift operation, these attachments are subject 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. Because of their bulk, some attachments also generate local hot spots and can enhance the creep and fatigue effects. Little 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 by a flexible link. Replacement of furnace wall tube ties with cranked ties gives greater flexibility. Platen alignment can be effectively maintained using 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 replacing boiler casings and thermal insulation.

A.2 Expansion-Related Issues

Boiler Structures

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

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

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

Expansion joints are subject to increased 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 are required to accommodate the thermal movement between the hot pressure parts and cold support steelwork. These supports are usually clevis pin or rocker connections and do not generally pose a problem under two-shifting. However, if these connections seize, the sling rods are 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. Another potential problem is load migration where the support load is transferred across the boiler either by relaxation on or failure of highly-loaded supports; this may lead to overloading and subsequent failure of individual supports. Periodic inspection is advisable, possibly with check weighing of the load in supports to identify locations where this problem might be occurring and to confirm the integrity of the support structure.

Pipework Systems

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

Most pipework 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 variations in load caused by physical changes in insulation weight or valve and actuator weights. These changes result from deterioration of the support mechanism, which is caused by increased friction (seizure) and build-up of dust on the pipework.

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

Under two-shift operation, ensure that the pipework is free to move throughout its full design range. A full visual survey of the pipework in both the hot and cold positions is required, and the movements of individual supports should be identified and compared with design. Where significant differences appear (for example, >20% variation on movement), further analysis should be performed 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 be beneficial.

Differential Expansion of Turbine Rotors and Casings

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

Some older machines that have previously been on base load may exhibit “sticking” if the keyways have not been adequately maintained and lubricated. This problem can usually be overcome by renewal of the keyways during a planned outage. Note that this work may require removal of the turbine cylinders.

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 two-shifting is introduced, it is important to understand what is happening within the turbine from the turbovisory equipment. Each case must be evaluated on its own merits. Where problems arise, the solution may require changing the operating procedure or increasing clearances, albeit at the expense of efficiency.

There have been instances (especially on older machines) of thermal distortion as a result of turbine support pillars "flexing" under the influence of temperature changes. The pillars tend to bow toward the hotter side. Although such movement is small - probably less than 1 mm - the movement combined with the tilting effect may be sufficient to disrupt turbine alignment and clearances. The problem can usually be overcome by thermal insulation of the pillars.

Low-pressure (LP) rotors have posed problems because of their construction with shrink-fit diaphragms. Cyclic loading may jeopardize their integrity. To combat this, some LP rotors have been replaced with mono-block forgings.

Alternator Components

Concern has grown about generator ring integrity, despite the introduction of the Fe-18Mn-18Cr end ring alloys. It is recommended that before two-shifting commences, end rings should be ultrasonically tested to ensure their ability to supply an automated technique. If the end rings are of the older Fe-18Mn-5Cr type alloy and there has been any suspicion of moisture contamination, they should be inspected because of the threat of stress corrosion. The stress variations that result from two-shifting (that is, starting and stopping) are more likely to cause damage, but 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 end ring designs are likely to be more susceptible to damage because they are prone to high cycle fatigue. One of the older variants uses retaining rings that are shrunk onto the end disk, which itself is shrunk onto the shaft. In this case, a relative movement between the end ring and the end windings occurs with each revolution. More recent designs can induce cracking in the shaft, both in the rotor tooth region and in the end ring itself. With this design, the ring is shrunk onto 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. 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 relatively simple. Modified designs that reduce stress concentrations in the end ring have overcome some of these shortcomings.

Two-Shifting Problems Associated with Steam Turbines

From about the mid-1970s, 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 suggests that, in general, most turbine plants will 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 two-shift regime if due care is taken.

Most large turbines 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 single flow with double-shell construction, and IP and LP turbines are usually double flow, single-shell construction. The majority of rotors are mono-block 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 disk and diaphragm construction.

Operation in a two-shift regime has two main effects:

- Thermal fatigue and associated creep-fatigue
- Mechanical fatigue as a result of load and speed variations

Creep-fatigue associated with thick-walled components, including governor and stop valves and HP and IP turbine inlet belts, is described above. FE analysis methods are now widely available at a reasonably low cost to permit modelling of components perceived to be at risk. Although it may not be able to accurately predict the life of components, application of this type of modelling provides a valuable understanding of the stress profiles within the component and identifies potential weaknesses and vulnerable areas. This knowledge can be used to optimise operational procedures in order to minimize the effects of thermal fatigue as well as inspection procedures focused on selected locations at appropriate operating intervals. The addition of temperature and temperature differential instrumentation enables the operator to minimize the intensity and duration of adverse conditions, information that can then be incorporated into autostart sequences. The scope for modification to existing turbine plants 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 plant operators favour “skin peeling” of HT rotors where fatigue damage is a concern. Skin peeling involves skimming off approximately 1mm of material from the surface of potentially critical regions of the rotor (for example, radii). This process is typically carried out at midlife and effectively removes fatigue-related damage, resulting in what is essentially an “as-new” surface.

Another area of concern has been the effect of embrittlement and fatigue on the critical crack size of HT rotors. Major rotor failures in the 1980s (for example, the Gallatin plant in the United States and Irsching in Germany) led to the development of inspection methods (for example, borosonics) and assessment procedures such as EPRI's SAFER code. The potential for embrittlement is largely a function of residual or tramp elements, which are strongly influenced by the steelmaking process. In general, only older rotors (that is, 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. Although severe embrittlement is rare, it should be assessed in older rotors under cyclic operation.

Mechanical fatigue issues arise from two sources. First, 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. It is important to pass through these speeds as quickly as possible. Over a number of starts, the number of cycles at critical speeds can accumulate to significant values and subject components such as turbine blades to unacceptable high cycle fatigue levels.

The area most vulnerable to mechanical fatigue is LP blading. Blade length subjects the root area to very high centrifugal stresses, and any defects within this area significantly reduce blade integrity. The inspection of disk 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 design and analysis methods to improve the blade-root detail, which should eliminate this potential problem. Where mechanical fatigue problems are encountered with LP (or any) blading, it is usually possible to re-blade the rotor with a modern blade design to reduce or eliminate the problem. In some instances, re-blading may be combined with other improvements in efficiency. Where the LP blades are cracked, the cracking may be exacerbated by the onset of corrosion-fatigue.

Four minor potential problems have been identified:

- Increased wear and tear on turbine valve gear
- Overheating of turbines as a result of windage
- Turbine differential expansion
- Erosion as a result of oxide (scale) impact on HP and IP blades.

Cyclic operation requires increased operation of turbine governor valves and stop valves. Additional wear and tear will occur on the valve seats and valve stems, especially under throttling conditions when flow-induced vibration can lead to mechanical fatigue and wear. This wear and tear can usually 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 is reduced, conditions could arise where the turbine is actually driving the steam, which can lead to a degree of overheating. This occurs on HP cylinders where a bypass 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.

The section above describes turbine expansion problems relating to rotor and casing differentials, gland sealing, and blade tip clearances. When moving from a base load situation where efficiency is of particular concern (and hence the need to minimize blade tip and gland clearances), it is necessary to review the clearances and make adjustments appropriate to a two-shift operating regime where reliability may be increased at the expense of 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 United Kingdom but more prevalent in the United States and may be the result of excessive thermal shock under abnormal conditions or of rapid load shedding under fault conditions. When these oxide particles are small, they are carried through the filters and enter the turbine. The high velocity impact on the nozzles and blades results in increased wear, which (over time) can lead to significant levels of erosion. In one instance, the build-up of scale deposits on the filter resulted in increased pressure drop across the turbine and, therefore, 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 or use of bypass systems (which are relatively rare in the United States) appears to minimize solid particle erosion.

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

- A fully integrated solid forging construction to reduce rotor manufacturing time and decrease the likelihood of intergranular stress corrosion cracking
- 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 modelling of the new turbine to optimise thermal transient effects

- A bypass system.

A.3 Corrosion- and Fouling-Related Issues

Waterside Corrosion in Economizers, Feedwater Heaters, and Evaporators

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

Two-shifting creates a need for increased supplies of feedwater. This need is the result of both the necessity to drain off condensate from pipework, manifolds, and turbine casings and the problems in maintaining boiler contaminants below specified levels. The more thorough the efforts to keep the boiler water within specification, the more likely the requirement for an increased supply of feedwater.

Feedwater of a “peaty” character can result in additional problems with erosion-corrosion of the feedwater heaters because organic acids derived from the peat decompose at pressure, liberating CO2.

Differences between once-through and drum boilers are likely because the former are normally operated with minimal levels of inhibitors to avoid deposition of solids at the steam-water interface. Therefore, during steady-state operation, the risks of contaminant-induced failures in once-through systems are much less than the risk in drum boilers. Conversely, it would seem that once-through boilers could experience significant problems during two-shifting when the chances of contamination will greatly increase. Drum systems can blow down the boilers to reduce contaminant levels.

Many plants experience problems with condensers. Leakage of air and cooling water continues during shutdown, leading to contamination. Unless ancillary steam plants are available, there is likely to be a period (during start-up) when oxygen gets into the system - resulting in corrosion-fatigue in the evaporator sections of boilers. This can be one of the most serious problems with two-shifting. Straight hydrazine injection can often not be used to eliminate oxygen during start-up because the water temperature is too low for the necessary reactions to take place. Leakage of air into deaerators is an additional risk.

The view that high oxygen levels during start-up strongly influence the propensity to corrosion-fatigue is generally accepted by the power plant industry. However, Dooley (EPRI, United States) has asserted that during steady-state operation, corrosion-fatigue is not definitely related to high oxygen levels. In his experience, the most important factor is the cracking of the protective magnetite film, which has a critical level of strain of 0.2%. Such strains are likely where there is some kind of restraint or where temperature differences lead to excessive thermal gradients. There may be an environmental aspect to corrosion-fatigue, but the environmental factor does not include oxygen. If this is the case, the results of autoclave trials may be misleading if not read carefully. The results from these showed that at lower temperatures typical of feedwater heaters and economizers, the number of cycles needed to initiate cracking was only marginally dependent on oxygen levels, and crack initiation occurred within about 1,000 cycles.

At higher temperatures typical of evaporators and HT feedwater 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 10,000 parts per billion (ppb). However, the risk of encountering this level during normal operation is quite low, because the oxygen level would have fallen away by the time that the plant got up to temperature.
Alternatively, it is possible to envisage cold slugs of oxygenated feedwater being admitted to the plant as it is going on-load. The combination of high oxygen levels and the thermal stress engendered by the “cold slug” sweeping its way through the system could be very deleterious. Work is needed to specifically address this area, although in principle it should be possible to eradicate the cold slug problem by changing operating practices. Nevertheless, it is difficult to see how thermal stress can be eliminated during start-up, which leads to the central question: what are the corrodents, other than oxygen, that contribute to corrosion-fatigue?

One of the most serious problems with demineralizing water production in two-shifting operation may be the result of water treatment and condensate polishing equipment being put on standby during weekends. During periods of zero flow, migration of ions between the different layers of resin could result in a build-up of sodium and sulphate in the water contained inside of the water treatment vessels. This water is carried forward into the equipment. Detection of leakage is simple and can be performed using a conductivity meter; however, disposal of contaminated water carries an economic cost.

A.4 Steam Turbine Erosion, Corrosion, and Fouling Aspects

Steam Turbine Erosion

There are two types of steam turbine blade erosion: erosion by particulates is caused by oxide scales and is found at the front end of turbines; and erosion by water droplets is essentially a back end problem.

Oxide scale erosion is normally the result of the exfoliation of magnetite scales from the superheater and reheater. During two-shifting, peak steam temperatures can be up to 30°C over design. This 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 two-shift operation will cause some initial difficulties with erosion, the problem will disappear after loose oxide has been shed.

Of greater significance is the risk of erosion-corrosion as a result of water droplets at the back end of the LP turbines. Off-design operation can lead to increases in steam wetness, which result 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.

Steam Turbine Fouling and Stress Corrosion

The issues of steam turbine fouling and stress corrosion are linked because they essentially result from the same phenomenon: the carryover of boiler water salts and impurities that are deposited on turbine blades and rotors. The problem of carryover is more likely with drum boilers that have continued to use the congruent phosphate treatment.

Some would say that in two-shift operation, the difficulty in controlling feedwater and condensate quality increases the risk of carryover. Nevertheless, the risks of turbine fouling appear to be lower in two-shift operation than in normal operation because 1) steam temperatures and pressures are below normal levels much of the time and 2) the Wilson line in the LP section of the turbines moves during start-up and shut-down, which tends to wash away deposits. (The Wilson line is the point in the LP turbine at which steam begins to condense into water.) The need to flush through the turbines in order to control temperatures may also help to wash away deposits.

Stress corrosion of turbine blades and rotors is likely to increase with two-shifting as a result of greater steam contamination. Sections of the turbine not normally susceptible to attack because they operate above the dry-out line will operate in a hot/wet condition.

Much of the likelihood of steam turbine fouling and stress corrosion depends on the turbine construction. Older rotor designs using shrunk-on disks appear to be more at risk than mono-block or welded construction rotors. 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.

A. 5 Fireside Corrosion

Superheaters and Reheaters

Serious fireside corrosion has been confined to UK plants and a limited number of units in the United States, where steam temperatures were at or in excess of 565°C. With these temperatures, 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. Therefore, this compound is corrosive only over a 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 two-shift operation is likely to alter the furnace environment so as to change the attack rate. Earlier views suggested that the level of SO3 in the furnace atmosphere was critical. It is now considered that the bulk of the SO3 forms within the deposit as a result of catalytic reactions within the ash layers, stabilizing the alkali-iron trisulphates. The sulphur content of the furnace gas has only a minor effect.

The effect of furnace (that is, combustion product) temperature is likely to be more marked. High furnace temperatures are known to increase attack rates. Furnace temperature has an indirect effect, involving the decomposition of molten trisulphate near the outer surface of the deposit, thereby increasing the concentration of SO3. This, in turn, increases 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 dramatically affect corrosion rates. Furnace and metal temperatures are low for much of the start-up, and these parameters are likely to be more significant. Some acceleration occurs if the fuel oil contains significant quantities of vanadium. Some of these effects may have been masked in the United Kingdom by the switch to low chlorine and, in some cases, low sulphur coal.

If the fuel oil is cut off prematurely, some of the pulverized fuel (PF) burners will cease operation. A layer of pulverized coal is then spread over the superheater and reheater tubes, resulting in carburisation. This increases corrosion rates of austenitic alloys by a factor of two.

Frequent changes in furnace temperature associated with two-shifting can lead to "protective" ash layers spalling off. This could be the result of simple differential expansion between the deposit and metal breaking the bond. However, it is also suggested that the deposits drop off because of dew point corrosion over a weekend shutdown, as a result of preferential attack at the deposit-to-metal interface.

Although deposit spalling does occur, there is as yet no clear evidence of an increase in the rate of attack. On one hand, the metal surface is 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 build-up of ash, the activity of SO3 cannot reach a critical value.

Furnace Wall Corrosion

Furnace wall corrosion is caused by a combination of oxidation and sulphidation, the latter giving a marked increase in the rate of attack. In an attack involving sulphur, a layer of sulphide is commonly found beneath the oxide. The sulphide layer can be regarded as the "shock troops" of the corrosion mechanism because sulphidation attack is much faster than simple oxidation. The issue, however, is more than one of simple metal wastage. Thermal stress and fatigue in boilers leads to craze or "elephant skin" cracking of the tubing or waterwalls. Cracking of this type can be difficult to identify because of the layers of ash and slag that cover the affected parts.

With the development of low NOx burners, furnace wall corrosion has become a serious issue in some situations because many of the designs increase H2S levels. These problems have prompted some utilities to use austenitic or nickel-based welded overlay coatings. Although these coatings have given protection, there is concern about distortion following welding and the levels of residual stress. In contrast, research gives a favourable account of the introduction of low NOx burners in the United Kingdom. Good results, however, stemmed from the fact that the burner flames were surrounded by an "oxygen excess blanket," which prevented reducing conditions occurring at furnace walls.

The effect of chlorine on furnace wall corrosion, which is considered a secondary factor by most workers outside of the United Kingdom, has been analysed. The chlorine effect is seen only under conditions of high heat flux and under reducing conditions. In these circumstances, the attack begins to mimic that seen in waste incineration environments with a layer of iron chloride developing at the scale-metal interface.

Two-shifting seems certain to exacerbate the furnace wall problem. Its short time of operation implies that the furnace structure will be cold, making it difficult to ensure that the pulverized coal will burn properly. These factors 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 unburned pulverized coal impacting on the furnace wall, increasing corrosion rates in that area of the furnace.

All of this must be viewed 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 stressed areas. Differences in expansion coefficient between the ferritic and austenitic alloys will result in thermal/corrosion fatigue, both on the fireside and waterside of the tubing. In this case, the use of low-chlorine coal in two-shifting operation would be advantageous.

Dust Removal

In general, electrostatic precipitators perform better at low loads because the reduced proportion of unburned carbon in ash and increased residence time of the gases in the precipitator allow 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 because any moisture can result in a build-up of dust, which, if pozzolanic, can be difficult to remove. The acid gases also increase corrosion at lower temperatures. If low temperatures do become a problem, it may be necessary to install a warming system (for example, a gas burner system) to pre-heat the precipitators when bringing the unit back on load.

If SO3 injection is used (as with low sulphur coals), it may be necessary to modify the injection rate to match the gas flow.

During boiler start-up, the precipitators are not usually energized until stable combustion has been established. This may give rise to emission problems into the local environment. Where this is not acceptable, the operating procedure must be reviewed with regard to energizing the precipitators earlier in the start-up procedure.

Electrostatic precipitators need special consideration because of the potential risk of moisture causing dust to adhere to the electrodes, which subsequently become baked. Maintaining temperatures above 90°C reduces this problem. A similar problem exists with bag filters, in which the main problem is to avoid temperatures dropping below the dew point of the flue gas.

A.6 Issues

Pumps and Auxiliaries

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

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

Most utilities engaged in two-shifting have moved to a more proactive maintenance regime to anticipate wear and tear.

Electrical Equipment

General

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

Motors

The number of starts per year imposed by two-shift 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

The material properties and machining quality of modern generator rotor shaft and end rings should ensure that they are not adversely affected by two-shifting. Some rotor designs have suffered from copper dusting while 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 two-shift operation can cause abrasion of the inter-turn insulation. Another common problem is stick or 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. These phenomena are caused by expansion and contraction of the conductor bars relative to the core, degrading the semiconducting coating of the bar. This is a long-term issue, and this type of degradation may be detected 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. Because it is not be expected to become a problem before the first scheduled removal of a machine's rotor, the wedge tightness can normally be checked at that time and a suitable strategy formulated. Systems are available to perform wedge tightness checks with the rotor installed.
Switchgear Because of the increased number of operations involved in two-shift operation, extra wear and tear is imposed on switchgear.

APPENDIX B: ETD Best Practice Guide for Cycling

B.1 Improved Hot Start Procedure for Rankine Plants

Guidance from the original equipment manufacturers (OEM's) for traditional base load units is likely to have been very conservative with a typical cold start time of 12 to 15 hours and 3 hours for a hot start for a large machine. Economic two-shift operation requires that units are brought back on load and taken off load as quickly as possible to minimise off-load heat costs. This action has to be balanced against the obvious effects of induced thermal stresses, resulting in costly plant failures.

Years of experience in two-shift operation of large units, predominantly in coal-fired regime, have convincingly affirmed (proved in studies) that large coal-fired generating units can perform extensive two-shift operation without requiring replacement of major components of boilers or turbines. This achievement required a focused attention to thermal and operational flexibility during the design stage, and a sound understanding of the influence of methods of operation on the conditions developed in boiler and turbine during start-ups and shut-downs.

To accomplish this, the more damaging thermal transients that can occur in boiler, turbine and key balance of plant equipment must be avoided through the establishment of sound procedures for shut-downs and start-ups.

Accelerated start-up is typically achieved through a combination of changes in procedure and plant modifications. Whilst each unit will need to be assessed on its own merits and limitations, the following is typical of units currently operating successfully on a two-shift regime.

On boiler shut-down:

- De-load the unit to 50% load (using sliding pressure control if available) and follow with rapid shut-down whilst maintaining maximum superheat and reheat temperatures.
- Top up boiler level (drum type boilers) before burners removed.
- Box up the boiler to maximise heat retention. Avoid air purge of boiler.

On boiler start-up:

- Commence boiler light-up sequence fans in service (purge boiler), light up burners in service.
- Begin to raise condenser vacuum (air pumps in service).
- Open boiler stop valve or by-pass early to facilitate turbine gland steam sealing.

- Open turbine by-pass if fitted.
- Initial firing with boiler drains closed to raise the temperature as quickly as possible to match metal temperatures and to begin to raise steam.
- Fire in a balanced pattern to promote boiler circulation (natural circulation boilers).
- Increase firing rate to raise boiler gas temperature to match final steam temperature as soon as possible.
- Open superheater bypass if fitted.
- Open drains progressively starting from the primary superheater and moving towards the final superheater (fit inter-stage drains) so that flow is established in final superheater when gas temperatures are matched.
- Delay opening of turbine steam drains to avoid cooling of main steam pipework.
- Fire on coal as soon as possible (just prior to steam to set if bypass fitted or just after if not fitted).
- Steam to set and run up to speed.
- Synchronise unit and raise load at prescribed rates (typically 3% MCR/min up to 50%MCR and then 5%MCR/min up to full load)
- Close turbine bypass and close superheater bypass (where fitted)
- Commission mill burner groups as required.

Figure B-1 shows graphically the main actions during start-up along with temperatures of critical equipment items.

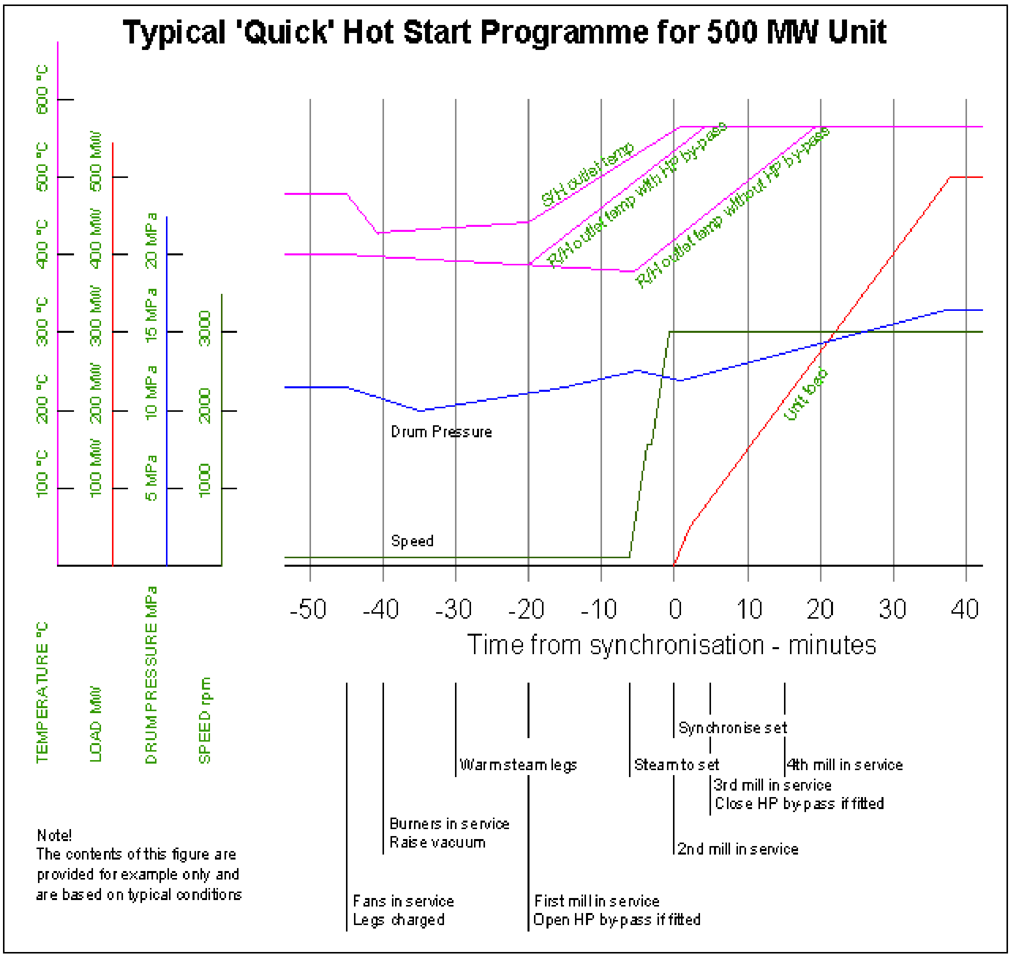


Fig. B-1: *Start-up Parameters*

Rapid load change, especially during load shedding can also upset flow through the boiler with similar localised overheating. This does though tend to be short lived as the firing rate is reduced to match the load.

Flexibility of Ramp Rates and Start Times for Cycling HRSGs

Start-up times are obviously critical to the stressing of HRSGs and this is basically determined by the need to get the water and steam up to temperature as quickly as possible. For a Foster Wheeler HRSG at Fort Myers, Florida, USA, a hot start is scheduled to take 120 minutes, a warm start 150 minutes, and a cold start 250 minutes. In other units a hot start, after a trip, has taken just 75 minutes. A cold start is required when pressure and temperature have dropped below certain limits. Stroman has also indicated that for a typical Foster Wheeler unit a cold start is specified whenever the pressure drops below 28 bar (400 psi) or if it drops below 31 bar (450 psi) for an hour or longer. To improve heat up rates and reduce the thermal stress, some operators use sparging steam to pre-warm the HRSG prior to firing the GT.

It is important to note that the start up times may also be determined by the constraints on temperature differential in the steam turbine. Carefully matching steam temperature to turbine metal temperatures is very important. It is possible to damage a turbine by having steam that is too hot or too cold. Stroman suggests that a rule of thumb for “temperature matching” is to have 50°C (90°F) of superheat (presumably to prevent condensation), but this should not be more than 50°C (90°F) above the turbine metal temperature (to prevent thermal shock on the rotor). This is only a rule of thumb which should not be used in place of the manufacturers’ recommendation, and should only be used when there is no other guidance. Pearson and Lefton consider that the temperature difference for matching should be much lower. In the view of the authors of this report, it would seem more sensible to take this more conservative approach with modern HRSG designs which operate at higher temperatures.

B.2 Condition Monitoring

Traditionally, many utilities have relied upon experience and inspection to verify the condition of the plant. However, modern technology using computers to log information and subsequent sophisticated data analysis techniques can yield useful information at a much earlier stage. This information can be used to modify plant operation to a more benign regime, to plan future maintenance strategies, etc.

Monitoring and diagnostics plays a vital role in the competitive operation of power plants by improving performance, reliability and availability, by enabling the optimal scheduling of maintenance activities, and by minimizing the risk of costly, unscheduled outages. Traditionally, local-to-plant monitoring was used mainly for performance analysis and remote-from-plant monitoring by technical specialists provided longer-term diagnostic and early warning support. Many systems have been developed and used successfully for monitoring and diagnostics of, for example, gas turbines and electrical generators. There are also a number of boiler component life utilisation software systems, which use on-line monitoring data as inputs for calculations of creep and fatigue damage accumulation, and these are commercially available from a number of plant manufacturers, R&D organisations and service providers.

As an example of an integrated monitoring and diagnostics system, ‘AMODIS’ has been developed by ALSTOM. This consists of a modular system at the power plant that can be configured to meet the plant’s specific requirements. Data stored by *AMODIS* and other relevant data such as fast transient logs, can also be transmitted back to a remote support centre for further analysis on a regular or event basis. All data transmitted to the support centre is stored in a central database, accessible only by authorized employees, where the data is analysed using specialized software tools by power plant component specialists.

Whenever possible, the *AMODIS* system uses existing plant instrumentation and most of the data is acquired from the DCS, although additional data can be acquired from vibration measurement systems, intelligent sensors, additional thermocouples, etc. The system is designed to provide a flexible, integrated approach towards the monitoring and diagnostics of many plant components and processes based on the concept of a platform and attached modules. The platform provides the core functions required by *AMODIS*, including data acquisition, data storage, operator display functions, secure remote access and secure data transfer. The modules provide the essential value-added data processing and health warning support for a particular component or process. Each module uses a combination of theory, statistical techniques and the OEM's knowledge to perform monitoring and diagnostics of particular components or processes.

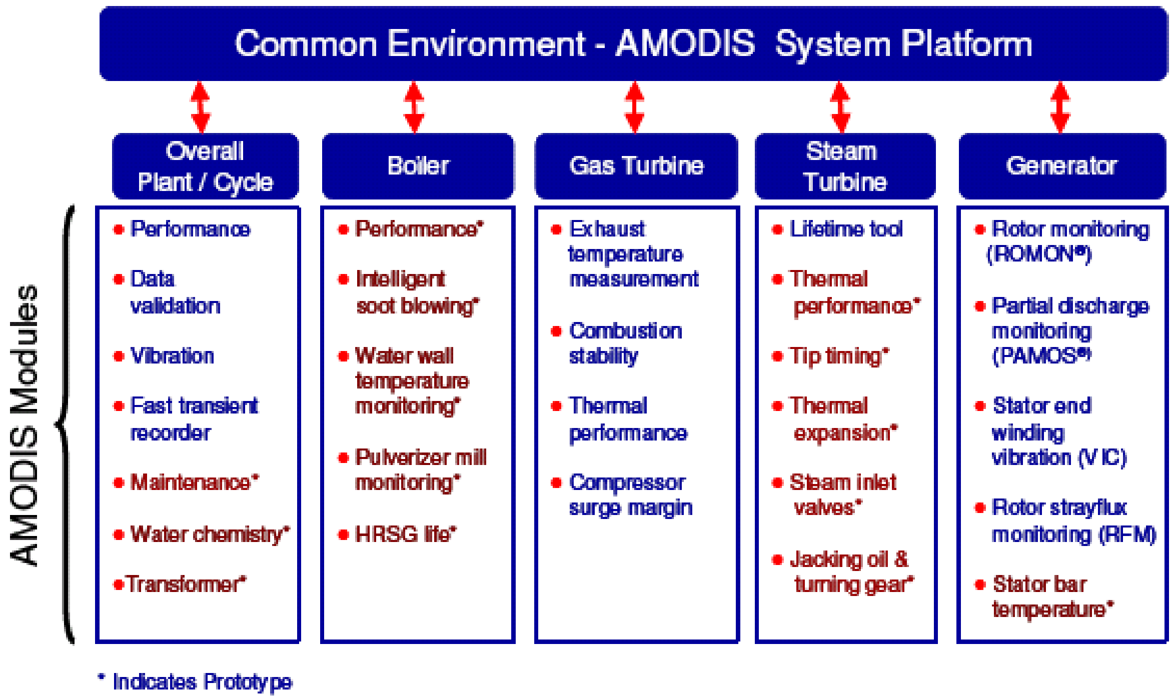


Figure B-2: Available AMODIS modules

For many plant operators, on-line monitoring of their ***boilers*** has been performed in response to recognised problems, rather than installing monitoring tools in advance. For remaining life assessment purposes, existing thermocouple and pressure indications at different stages of the steam path through the boiler may be incorporated into Level 1 assessments, and thus the accumulated creep and fatigue damage can be determined. In some cases, the data can be supplemented by the judicious placement of thermocouples to monitor the thermal response of components to various plant operations. This can be a useful tool in observing the behaviour of components and identifying operating conditions that may otherwise have been assumed to be benign.

Thermocouples can be installed to monitor defective components and determine the root cause of the problems affecting headers, reheater tubing and superheater tubing, evaporator tube banks, etc. For example, a plant operator was experiencing a corrosion-fatigue problem on the evaporators at one of their stations. To determine the root cause, instrumentation in the form of strain gauges, thermocouples and potentiometers was installed at various high stress locations around the evaporators, i.e. at bends, attachments, etc. While this was not a condition monitoring application in terms of fatigue life consumption, it was used as a means of identifying the operational events that caused the boiler evaporator tubes to see the greatest stress. Using the information gained from the strain gauge, thermocouple and potentiometer measurements, the operational procedures were modified to reduce the tube stresses and hence to promote longer component life by slowing down the rate of crack growth. A logger box was installed to collect the data, which was stored as it was recorded, and then downloaded via a modem connection to a PC with compatible software to convert the data into a useful form. Then the data were analysed and interpreted in relation to operational events. Thus, this was not a real-time monitoring application, but it was sufficient for the purpose of the investigation.

A number of *boiler component life monitoring software systems* have been developed by research organisations and by OEMs over the past 15-20 years. However, these systems do not appear to have been widely adopted by the plant operators, and some of the systems available in the past are now no longer available. Nevertheless, a number of new component lifting tools have been developed in the last few years. These systems may prove to be useful tools for monitoring fatigue (or creep-fatigue) damage in critical components, as a significant number of premature failures have occurred as a result of damaging thermal transients in units that are being cycled. One power plant operator that has not used component life monitoring systems in the past has recently informed ETD that they will soon be installing a system during a control and instrumentation upgrade of one of their boilers, in order to be able to run the boiler up in a shorter and more consistent time period without using up excessive amounts of component life.

Monitoring of ***steam turbines*** can be divided into two main areas: condition monitoring and operational monitoring. *Condition monitoring* systems are used to identify any faults or abnormal behaviour, which can then initiate further urgent investigation of the equipment or even result in an equipment shutdown. There are two main techniques available for detecting indications of faults or abnormal behaviour: assessment of vibration/noise, and measurement of temperature. Turbine vibration monitoring systems monitor the vibration characteristics of the turbine during start-up, normal operation, and shutdown. These systems detect abnormal vibration signals, which may indicate problems with the blading, and also provide balancing information. Temperature monitors detect overheating due to abnormal operating conditions.

Operational monitoring systems are used to simply monitor the operation of the turbine and, in addition to detecting abnormal conditions, can be used to optimise operating conditions. The parameters to be monitored for the steam are temperature, pressure, flow rate and steam humidity.

Increasingly, the condition and operational monitoring systems have become integrated into systems based on modern low cost sensors and electronics integrated into the plant management system. OEMs now include an increased range of monitoring equipment and retrofit systems are also available. Typically, a life consumption monitoring tool will be supplied by the OEM. They may also offer systems for monitoring the thermal performance, thermal expansion, steam inlet valves, and the jacking oil and turning gear.

A torsion monitoring system can continuously monitor the turbine-generator shaft for torsional oscillations. If a high-stress condition occurs, the recorder automatically captures the stresses acting on the shaft and transfers the information to a PC-based analysis system. This calculates the torsional mechanical response and performs a shaft stress and fatigue analysis, which includes loss-of-life per event as well as cumulative fatigue life consumption of the rotor for all recorded events. These systems allow operators to assess the impact of an event on the turbine-generator shaft as it occurs, allowing the power plant operator to assess operating practices that may cause damage. They can also be used as an aid in planning maintenance or inspections preventing costly repairs.

Some plant operators have informed ETD that while they have used *Turbine Stress Evaluators* (TSEs) for many years, they do not use them for analysis of stress cycles and evaluation of the current condition of the turbine. Instead, the TSEs are used as an operational aid to try and ensure that excessive stresses are not exceeded during run-ups. Their experience has been that OEM delivered TSEs inspired confidence in the operating staff and have been used on an on-going basis for operating the turbines. However, TSEs delivered by third parties, even when using full data supplied by the OEM, have never seemed to inspire the same level of confidence in operations staff. This may be due to the fact that a turbine had been operated successfully for 20 years without a TSE, and then the TSE was not trusted fully after it had been installed. Their experience indicated that best use is made of a TSE when it is supplied by the OEM and is used from the outset.

The steam chest and HP and IP turbine casings are thick-walled components which may well suffer from thermal fatigue (or creep-fatigue) cracking at section changes and other sites of stress concentration, such as flanges and grooves. Thermocouples can be installed at the high stress locations in order to monitor temperature gradients and ramp rates for use in fatigue (or creep-fatigue life assessment).

B.3 Component Life Monitoring Software Systems

In a typical software system for component life monitoring, temperature and pressure readings from sensors are input into algorithms for calculation of the creep and/or fatigue damage fractions for critical geometrical features. For creep damage, the consumed life fraction is calculated based on the stress from pressure sensors and the temperature from thermocouples, and knowledge of the creep rupture strength. Computation of the fatigue damage at critical locations is based on the stress/time/temperature history, which has to be divided into cycles at different amplitude bands, and the S-N fatigue endurance curves. The system can issue warnings when the life consumption rate is high, and issue alerts for inspection at preset life fractions, and identify operating practices that cause high fatigue life consumption.

The problem with fatigue monitoring systems is calculation of the stress/time history on-line, and there are three techniques for dealing with this:

- Based on a library of standard start-up and shut down cycles: Events are analysed off-line, and the fatigue damage is calculated. Then the system is set up to recognise events and add appropriate damage fractions. The system will not respond to any variations in the start-up and shut down cycles. This approach has limited use for on-line plant optimisation.
- Based on the rate of temperature increase: Using codes (e.g. TRD 301), thermal stress is calculated from the gradient of the temperature versus time curve. Approximate solutions are obtained, based on flat plate geometry. The results are conservative, but not very accurate for complex geometries such as nozzles.
- Based on thermal gradient measurement in the component wall: Temperature can be measured at the component surface and by a thermocouple inserted in a hole drilled in the header, but this is not popular with the plant operators. The best solution requires a FEM model and reverse Fast Fourier Transform technique to estimate the thermal gradient from simple external thermocouple measurements.

Software systems are available for monitoring creep-fatigue interaction in critical components. The creep and fatigue damage fractions are displayed on two separate axes (as per ASME Code Case N47). Lower bound material properties are used. Variation in properties and local effects at weldments cannot be taken into account. The systems can give a warning of when to employ other techniques, e.g. replication. Continuous data monitoring shows the effects of temperature excursions, and hot and cold starts. The output can be used to optimise operating practices (e.g. introduce softer starts to minimize damage) or to estimate the effect of rapid start-up procedures on lifetime.

A number of in-house and semi-commercial systems have been developed. Some of these were developed by research organisations for commercial purposes, some by manufacturers for use on their clients' new own plant, while others were developed by plant operators for use on their own plant. Some of these systems are quite simple and work on an inverse-design life rule, while others are more sophisticated in terms of modelling and take into account various creep, fatigue, crack initiation and growth algorithms. All of these operate on the principle of monitoring temperatures and stresses (or pressures) in various parts of critical components and calculate operating/ remaining life. Any temperature excursions due to cycling are thus taken into account in predicting the safe operating period. Some of these systems have been integrated with the SCADA systems now used for plant operation control and referred to elsewhere in this report.

According to an ETD survey, none of these systems so far have become uniquely popular and successful in commercial terms. The reasons for this could be as follows:

- Manufacturers have tended to build low cost simple systems for use on their own plant. The intention was not active commercial sale of the system. Although these systems are economical they are based on material average life (not necessarily cast specific) and the predictions therefore have large margins of uncertainty. However, increasingly sophisticated systems have become available.

- Research and technology organisations have built costly but comprehensive systems which incorporate creep fatigue algorithms, crack growth methodologies etc. However, because of the complexity of software development for commercial use and associated hardware compatibility very often these systems have tended be problematic. Both the high cost of software development (over US$1.5 million reported in one case) and the post-installation technical support combined with reluctance of utilities to invest large sums of money during these days of competition have discouraged the developers from further development of these systems.

B.4 Vibration Monitoring Systems

Computerised vibration monitoring systems, such as those available from *Prosig*, *Beran*, *Bently-Nevada*, etc, are very valuable in ensuring the long-term integrity of turbo-alternators and major plant auxiliaries such as feed pumps. They are also vital to diagnosing problems with the operation of the plant, be it returning from a major overhaul where there may be alignment / balance issues or when a sudden unexplained change occurs after a period of trouble-free operation.

It is important to remember that a vibration signal from an item of rotating plant has both magnitude and phase. The latter is the relationship between the rise and fall of the vibration with the shaft position as the shaft rotates. No item will be perfectly balanced nor will every shaft be perfectly straight. The vibration signal may also include components other than the fundamental usually twice or three times rotational frequency or, occasionally, sub-rotational frequency. The presence of these components and the phase information is utilised in computerised vibration monitoring systems.

If there is a change in the vibration signature of a turbine-alternator, then it might not be immediately apparent from measuring (and displaying) the overall vibration level. An example of this is the loss of part of a turbine blade or a piece of shrouding from a blade tip. Providing the loss is relatively small (i.e. not a whole blade - when the vibration will probably be so large that it could be heard standing next to the turbine), then the change in vibration may add to or subtract from the existing natural out of balance vibration. This may result in only a small change in overall vibration level, which may be far from obvious – particularly if it is a decrease. A computerised monitoring system will determine the vector change (magnitude and phase) in vibration and so will always be able to identify a change whether it increases, decreases or does not change the overall vibration level. This ability to identify relatively small changes makes it much easier to accurately diagnose the cause of even relatively small changes in vibration. Clearly, it is possible to establish significant information about the condition of the machine by breaking the overall vibration signal down into its component parts, and by comparing the vibration components with other process parameters.

Whilst these systems come with excellent software, which allows staff with relatively little knowledge of plant dynamics to assess plant conditions, it is easy to network the systems to allow experts off-site to access the data and provide guidance as to the extent of the problem.

Application of computerised vibration monitoring systems

The use of on-line computerised vibration monitoring systems is generally restricted to the more critical parts of a generating unit. It is essential to monitor the turbine and alternator. The steam feed pumps and to a lesser extent electric feed pumps are also very good candidates for monitoring. It is harder to justify on-line monitoring of the lower speed and simpler items such as fans and cooling water pumps.

Items such as turbine vibration measurement will be included at a basic level as standard on a power station build. Sophisticated vibration monitoring software systems may not be included. Systems from the likes of *Beran*, *Prosig*, *Bently-Nevada*, etc, may cost about $30,000-50,000 for a 4 turbine centralised unit. This assumes that the dynamic signals are available in the control room, which could be unlikely - in which case, the options are either to cable all the dynamic signals back to the control room or, preferably, to mount remote racks next to the turbine where the dynamic signals are available, and use communications links back to the control room. The costs involved depend on distances and cabling options.

Computerised vibration monitoring systems are quite user-friendly, but they do assume some degree of computer knowledge and understanding of vibration terms - harmonics, sub-synchronous, etc, to make sense of them. To set the alarms correctly will need someone who understands vibration. Clearly, it is good practice to archive data to guard against computer crashes and to store data over longer periods than the on-line disks can manage. The software usually has facilities to do this in a way that allows archived disks to be mounted, so that the older data can be read and compared with the current data. This is as an alternative to having to restore all the data from a back-up DVD, for example, onto the system hard drive.

Companies such as *Beran* and *Prosig* will offer annual contracts where they check the system and provide telephone support. They are really looking after the system they have supplied - not the transducers feeding it, and not the turbine on the end of it. The transducers are clearly station responsibility and the turbine vibration data needs to be interpreted by someone with adequate knowledge.

Some plant operators have informed ETD that they have never installed vibration monitoring software systems on their plant for a number of reasons, including:

- Analysis of raw vibration data needs experienced/trained practitioners
- Analysis packages were put on the market before they were sufficiently developed
- OEMs provide analysis services but they are relatively expensive and it is difficult to quantify the payback

This reflects an old problem in that unless a package has been properly developed with visible payback, then utilities will not invest, and without the investment further development is limited. However, now that the technology has become cheaper, and analysis and modelling techniques have improved, there should be more widespread use of condition monitoring systems. Providing references that demonstrate a solid return on the investment is key to getting acceptance by the decision makers and power stations.

Other major plant items

Often it is not practical to fit on-line vibration and temperature monitoring to all the major items of plant. The really important ones such as the main turbine-generator and the feed pumps are almost always covered by on-line vibration monitoring as well as recording of bearing temperatures and lubricating oil conditions. It is clearly very expensive to on-line monitor all the other major items of a power station. There are many of them (fans, mills, pumps, etc) and some can be very remote from the control room making retrofitting on-line monitoring an expensive option. Some, coal mills, for example can be very difficult to cable. It may be feasible to install some less sophisticated monitoring on the main boiler fans, but it becomes more and more difficult to justify the cost for the smaller items of plant, although failure of these items can cause significant loss of generation.

A very worthwhile alternative is to fix monitoring point studs to the plant items and then for someone to go round periodically with a hand-held instrument to record the vibration levels and collect other data, which are then input into a computer database. Companies such as Entek (part of Rockwell Automation) provide both portable data collectors and software. The type of instrument used nowadays is processor based and prompts the operator to follow a specific route around the plant.

At an item of plant, the operator will be prompted to utilise the vibration probe to take a measurement, and to take additional readings such as temperature or pressure at the same time. Back in the office, the data can then be loaded onto a computer and analysed by the software, which will produce a report highlighting any vibrations outside acceptable limits or significant changes from the historical data. There are many contracting companies who will provide a service of setting up agreed routes, carrying out the periodic monitoring, and reporting on the results if the company does not want to use its own staff for the work. Whilst not strictly on-line monitoring, this is a very useful extension where the full cost of on-line monitoring cannot be justified.

If an operator without the data collector thinks a plant item is running “rough”, then the data collector may be used to take some spot readings, which can also be loaded into the database to determine if the plant condition has really deteriorated and by how much since the last planned reading. This information allows decisions to be made as to when plant items should be taken out of service before a major failure occurs.

Although not as sophisticated as full on-line systems, these data collectors are very capable. They carry out frequency analysis of the vibration data and if the database is populated with bearing details, such as the number of rollers in a roller bearing, then this can allow identification of a damaged roller, for example, before it fails. This will allow repairs to be planned rather than suffering a breakdown.

B.5 Equipment Modifications and Improvements for Rankine Plants

When a former base load unit is required to be operated in a two-shift mode, there are a number of relatively simple and low cost actions which can be taken to improve the existing plant performance. There are also a number of higher technology modifications which can be considered to assist in two-shift and low load operation. These modifications will have significant impact on the way in which a unit is operated and it is therefore essential that these modifications should only be implemented following a detailed review of the system requirements, design implications and interaction with the rest of the plant. *The circumstances affecting their usefulness and commercial benefit will vary between locations and must therefore be reviewed on a station-by-station basis. Not all modifications will be of benefit to all stations.*

Increased Drainage Capacity

Progressive warming through of a boiler, steam legs and turbine in a well-controlled manner during two-shift operation is achieved using drains to promote a flow. Many boiler and steam pipework systems were designed for the removal of condensate and draining down. As such, many drains do not have adequate capacity to accommodate the necessary volumes to achieve a progressive drain warm through.

Upgrading the drains system to increase the capacity and their operability is a relatively simple and low cost modification that can be carried out to greatly improve the two-shift capability of a unit.

In doing this, one needs to consider the size of drain connections and drain pots fitted to steam legs, the integrity of the valves, the sizing of the drain lines and the capacity and performance of drains vessels. Consideration should be given to the recovery of clean drains water for re-use. Due account will also need to be taken of the materials used to accommodate the temperature of operation. Pressure reduction and vent silencing may also be considerations, especially where noise may have a local impact.

__Improved Thermal Insulation__

General experience of two-shift cycling suggests that the thermal effects are less troublesome when the unit is maintained at high temperature during the daily shut down period. Most operators find that 'warm' starts present the most arduous conditions for matching steam to metal temperatures.

High temperature can be achieved by ensuring high integrity of the thermal insulation. Furthermore good thermal insulation can assist in start up sequences and reduce start up times.

Poor insulation can have unforeseen effects. For example, in one instance, defective boiler cladding and insulation on the rear enclosure of a boiler, where a flow of saturated steam became exposed to cold air ingress, resulted in the condensation of the steam, during start up. The condensate restricted the flow of steam in the exposed tubes, which then led to overheating of the top of the tubes. In short, tubes failed by steam starvation/overheating at a site remote from the actual source of the problem.

Target areas for improved insulation include headers within top and bottom dead spaces, steam pipework and boiler stop valves, turbine control valves and turbine casings.

Heat may also be lost due to passing gas or air dampers during the shut down. Attention to maintenance / adjustment of these can help to retain heat during the shut down period.

It is wise to ensure that the seal at the bottom of the boiler near to the ash hopper, in the case of a coal-fired boiler, is in good condition as this is an ideal location to let in cold air at the bottom of the furnace and for hot air to escape from higher up in the furnace.

Drain and vent valves play an important role in retaining heat during a shut down. If these pass then it is a waste of water / steam and energy whether a unit is on or off load. When a boiler is off load, however, passing valves can result in the boiler pressure falling more than normal. This not only requires more energy to get the boiler back up to temperature and pressure but the different starting point means that a slightly different procedure may be required, probably taking more time, to achieve the steam to set conditions without over-stressing the boiler components.

__Improved Oil Burner Reliability, Stability and Turndown__

Perhaps the most important requirement to achieve two-shift capability is light up burner reliability and operability. This is especially the case for coal-fired units where smaller gas/oil burners are used for coal burner ignition and flame stabilisation. Some stations may wish to use the burners to carry a significant load to provide additional plant flexibility.

Older coal-fired stations may be equipped with systems which are less reliable than modern burners and their associated control systems. Lack of use may be a contributing factor in their poor reliability. Base load stations will only use oil burners infrequently at light up. As such, the burners tend to be inherently less reliable and are not maintained to the levels required for regular use on a two-shifting station.

Boiler Off-load and Economiser Recirculation

When a unit is taken off load, the lower parts of the boiler will cool more quickly that the upper parts. This is due in part to the tendency of heat to rise, the ingress of tramp air at the furnace hopper, the lower heat inertia of furnace wall tubes and feeding of cold feed water to maintain boiler levels. Furnace tubes will cool more quickly than the heavier walled downcomers and distribution pipework, resulting in thermal stresses. Temperature differentials between adjacent tubes may develop which also induce significant thermal loadings on the boiler wall structure and header stubs. This will result in tube failures, header stub cracking and boiler structural attachment failures.

On natural circulation boilers local temperature differences may also induce a flow reversal (down flow) in sections of the furnace wall tubes. Subsequently this may give difficulties in re-establishing stable flow and circulation when the boiler is re-ignited. It is less of a problem on forced circulation systems.

Such difficulties, both with the development of excessive temperature gradients and flow reversal, can be reduced by through off load recirculation of boiler water. Such a system comprises a pump which takes water from the downcomers and feeds it into the bottom water walls. Forced circulation boilers could use the existing boiler circulating pumps, but may benefit from the installation of a smaller pump specifically for off load duty. Sizing of the circulating pump and distribution feed points need careful consideration The system needs to be large enough to promote a flow throughout the boiler and must not introduce any dead zones where circulation is not present. More than one pump may be necessary as shown in Figure B-3.

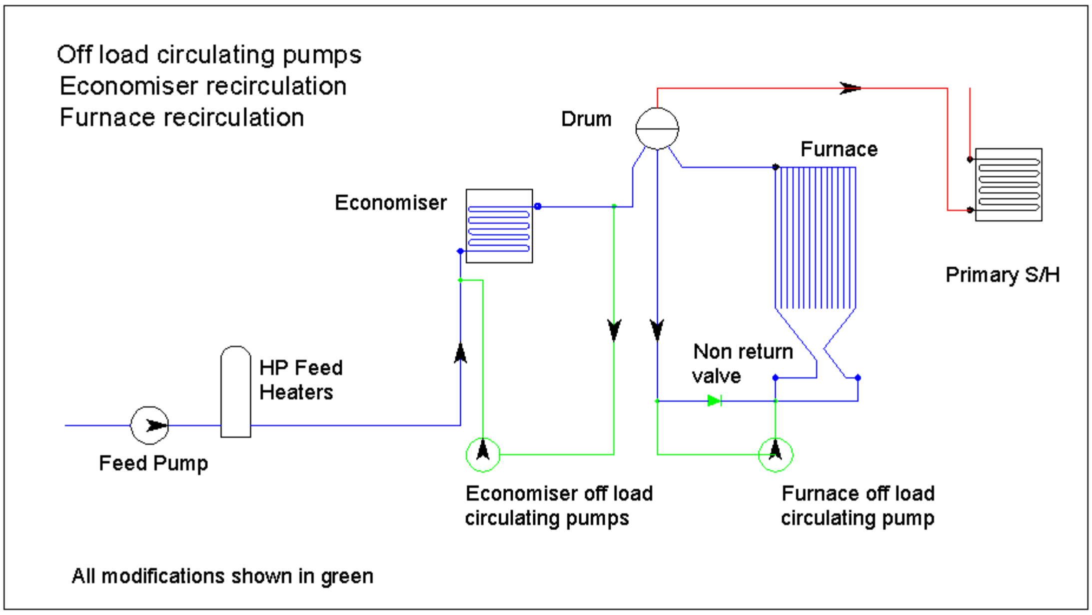


Fig. B-3: Off-load recirculation pump system

Boiler Hot Filling

Boiler hot filling systems allow hot water to be transferred from an operating boiler to one that is off load. During the shut-down period, prior to a hot-start, the boiler can be topped up without the need to use a feed pump. Filling with hot water from another unit will reduce the thermal shock to economiser inlet headers and reduce the tendency to ligament cracking in this vulnerable area.

In most cases boilers transfer water from the economiser inlet of the operating boiler to the economiser inlet of the off-load boiler. However, other configurations are possible; the transfer of water from the economiser inlet of one boiler to the economiser outlet of the other, transfer between economiser outlets or between economiser outlet and inlet.

Inter-stage Drains

A typical feature of rapid boiler start up are the delays in establishing steam flow through the superheater to 'boil out' any condensate in the superheater platens, thereby avoiding any risk of quenching of the outlet headers. These delays obviously conflict with the need to establish a flow elsewhere in the boiler to avoid local overheating. Hence a priority is the installation of inter-stage drains, prior to platen sections. This is a relatively low cost installation which can facilitate rapid boiler firing in two-shift situations.

Tube Attachments

Thermal cycling of a boiler gives rise to mechanical cycling of attachments between pressure parts and the boiler framing. High rates of attachment failures are generally the most significant factor in unit reliability when moving to a two-shifting regime. It is therefore worthy of consideration to change to better designed versions that are more able to accommodate mechanical fatigue. Figure B-4 shows an improved 'cranked' attachment bracket.

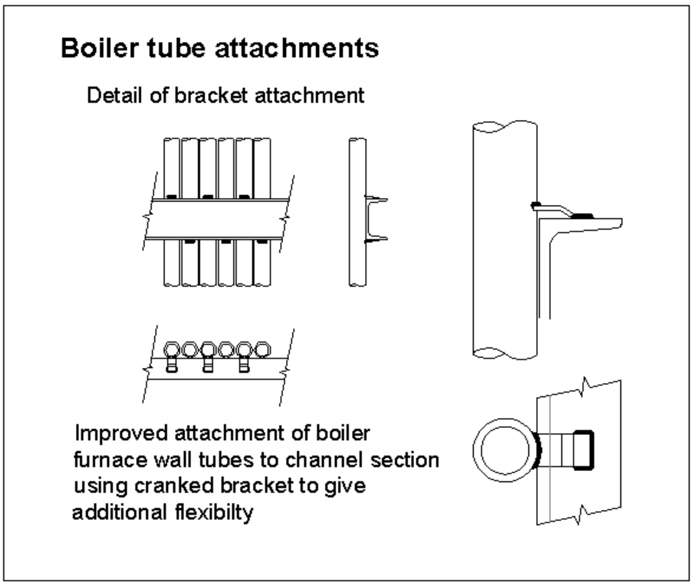


Fig. B-4: Improved attachment bracket

HP Turbine Bypass

One of the fundamental issues in start-up, of typical base load units, is that until such time as steam is admitted to the HP turbine, there is no flow through the reheater. It is therefore necessary to restrict the firing so as not to cause overheating of the reheater elements. It may be possible to increase the firing rates, and thereby reduce the start up time, by the installation of an HP turbine bypass system to permit a flow of steam through the cold reheat steam legs and into the reheater. This will also have the added advantage of warming the reheat steam legs.

The system comprises a by-pass line taken from a tee prior to the HP turbine throttle valve, and then linking back into the system after the HP turbine exhaust. It includes a pressure reducing station and attemperator spray. Sizing of the flow is only required to provide an adequate warming flow, but some by-pass systems are sized to accommodate a much greater flow for load shedding purposes. The greater capacity system may not be suitable for the lower flow situation found in two-shift operation. Consideration also needs to be given, however, to the formation of condensate within the bypass, when the line is shut. In short inter stage drains will probably be necessary, and the design must be such to minimise thermal shock.

Some locations have extended the bypass concept to include an IP turbine by-pass system, but this is not necessary for two-shift operation. This system may also prove advantageous where variable pressure operation is employed.

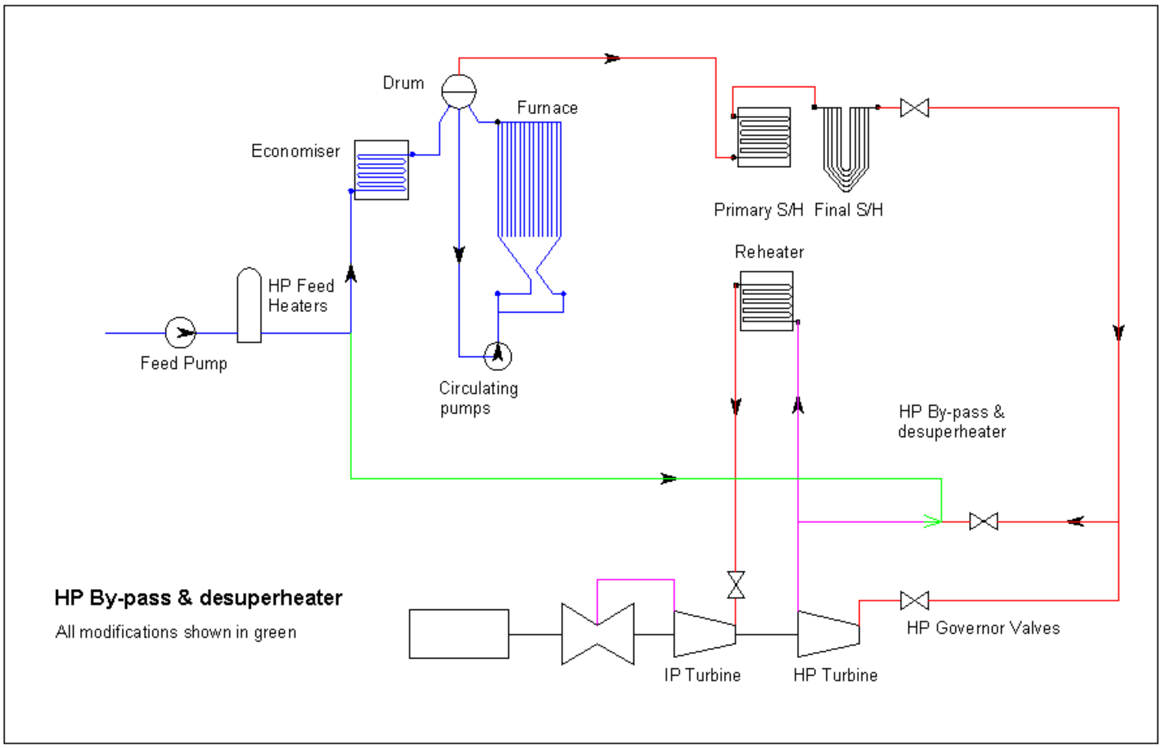


Fig. B-5: Turbine bypass system

Condenser Air Extraction and Vacuum Raising

Most modern power plant now uses rotary vacuum pumps to raise and sustain vacuum. In the event that the existing installation is unable to raise vacuum within the required timescales, it will be necessary to install additional or improved capacity.

Auxiliary Steam Supplies

A constraint on raising vacuum may be poor gland sealing. Most systems rely on steam sealing of turbine seals, taken from the main steam legs. This is therefore only available after the steam legs have been charged and may thus cause delay in the start-up sequence.

An auxiliary steam supply can be used to provide 'off-load' steam for gland sealing. This is probably more beneficial on cold and warm starts, but may be advantageous during two-shift starts at some locations.

B.6 Impact of Two-Shifting in Rankine Plants

One item which is often forgotten is the effect on the staff of moving to a different operating regime. During the early life of the station they move from the excitement and pressure of getting the station up and running to the relatively relaxed regime of continuous operation. Many of the shift operating staff can go for a long time without experiencing the start-up of a unit. This depends somewhat on the reliability of the plant and the periodic overhaul regime but it does often occur. In these circumstances, the prospect of starting up a unit can be frightening for some operators.

In the CEGB in the 1970s although some of the 500 MWe coal-fired units had some significant issues, several of the units managed runs of over 200 days without coming off load apart from the mandatory physical over-speed test which was required every six months. All this required, however, was to come to zero load, open the main circuit breaker, carry out the physical over-speed of the unit, re-synchronise the unit and load it back up.

The maintenance or engineering staff can also get into the base load operational mindset. The reliable operation of certain items of plant – boiler drains valves or start-up oil burners on a coal-fired station, for example, become less of a priority. Whilst the plant is running it is also difficult or impossible to work on some of these items. Sometimes the concept of cyclic operation can fill everyone on the station with horror. In order to overcome these problems there are a number of useful techniques which can be employed. Some of the possibilities are explored in the following paragraphs.

Forming a committee made up of management, operational end maintenance / engineering staff can be very beneficial. Often staff do not fully understand the necessities for the plant to change its operating regime. A committee with this make up provides the ideal vehicle for management to explain and discuss the issues with a small group who can then relay the message to their working groups.

The acceptance of the need to change by the whole station is really beneficial to obtaining a team effort to make the changes required and getting everyone pulling in the same direction.

A committee allows all the interested parties to bring their thoughts and suggestions to the table. Some staff may have already seen this transition to flexible operation before at other plants before and may have a lot to contribute. The committee can be proactive in considering information available and in seeking information from other sources to help the process along.

The station design documents, manufacturers' information and the results of any start-up trials carried out either when the station was commissioned (it is always a good idea to require the manufacturer to demonstrate the capability of the plant to start-up under a number of different conditions – hot, warm and cold – during the commissioning period) or as any subsequent opportunities have arisen or have been planned.

The experience of similar plants is very valuable in trying to avoid the same mistakes or in identifying essential modifications required to the plant to enable reliable start-ups whilst minimising plant damage. The similar plants may be within the same company where access to staff and data should be relatively simple or perhaps through the manufacturer but in different companies.

There are a number of examples of work within companies both on a national and international basis. National Power (one of the privatised companies formed out of the CEGB in the UK) had a cross-station two-shifting committee in the 1990s. This committee had a representative from each station. Some were operational staff and others were from engineering. Although the plant designs were often quite different the committee compared the performances of the stations in a number of respects such as:

1. Minimum shut down time.
2. Minimum on time.
3. Loading up rates.
4. Modification required to plant.
5. Plant damage seen as a result of flexible operation.
6. Additional instrumentation fitted to aid flexible operation.

The main success of this committee was to challenge the thinking. If a station had always carried out a start up in a certain way was it the best way. Looking at how others worked was most helpful. Sometimes it would merely confirm that the station was right. Other techniques simply could not be applied to their station.

Into the late 1990s and even slightly later TXU Europe (a subsidiary of Texas Utilities – TXU) took flexible operations to an international level when working with PVO in Finland. Staff in the UK held exchange visits with staff from Finland to demonstrate what progress had been made in the UK with flexible operation both from the operational and engineering point of view.

This work not only covered the technical aspect of the plant operation but also staffing levels. During the transition to very flexible operation in the 1990s the UK stations had also undergone significant restructuring driven largely by privatisation where staff numbers were actually reduced during this period and not increased as some may have expected.

The Finns found it most useful to discuss and observe the operations in the UK. Seeing what had been done and what could be done enabled them to return home with sound knowledge and to talk with authority and concrete examples to their own staff as to what was possible.

Again it was critical that the visits involved not only managers but also operations and maintenance staff. This gave the message much more credibility when it was taken back to Finland.

B.7 Staff Training Requirements for Rankine Steam Plants

Under traditional base load operation of high merit plant, plant personnel often gained comparatively little experience of unit start-up and shut-down. Typically these plants might see fewer than ten starts per year. Hence, some personnel might see fewer than two starts per year. His or her role was more that of a monitor, making fine adjustments and dealing with occasional emergencies. The time factors associated with infrequent start-up enabled the operator to approach the task over a relatively lengthy period of several hours.

Under two-shifting conditions, the plant operator is not only required to carry out start-up and shut-down regularly, but to do it quickly and efficiently. The requirements to be able to understand and operate the plant under the highly dynamic conditions of two-shifting places a high burden of responsibility on the plant operator. The scope to cause damage to the plant is greatly increased.

The operating staff also needs to be aware not only of the functional requirements of two-shifting, but also the commercial aspects of plant running costs and efficiency and the long-term effects of operation on the life expectancy of the plant.

Whilst the capabilities of the operator can be alleviated by the adoption of automation, and improved data display, there is still a need for a higher level of knowledge and understanding than that necessary under base load operation. It implies that there will be little or no scope for reduction in staffing in the first years of two-shifting.

It is vital that the senior operating staff be made aware of the issues in terms of “unseen” damage to superheaters and turbine etc., which tends to result from two-shifting. A two-way dialogue should be implemented to give the operating staff confidence in the new way of doing things. Once two-shift operation has started station management should monitor the behaviour of individual shifts (i.e. groups of operators) on the plant so as to identify best practice in terms of both start-up and shut-down times and the likely damage to the plant.

Nevertheless the operating staff will be aware of some of the issues that occur during normal start-ups and the station management needs to tap into this fund of information. It may be that some of the procedures which are used, for example, to control the temperature into the condensers during start-up, may increase the damage to components. Conversely the operating staff may be able to suggest simple changes to equipment, such as provision of additional drains or instrumentation, which could prove helpful.