Programme Area: Carbon Capture and Storage

Project: Hydrogen Turbines Follow On

Title: Review of Gas Turbines and their Ability to use Hydrogen-Containing Fuel Gas

Abstract:

Development aspects and assessments of Gas Turbines (fired by methane and/or H2) are provided, including contemporary OCGT GTs from GE. Additionally, potentially synergistic capture technologies are described.

Context:

Carbon capture is from GTs is relatively expensive, mostly because the lean burn technologies produce a flue gas very dilute in CO2, needing a voracious solvent and very large equipment. This package is a collection of background papers for exploring ways in which the capture technology could be assisted by GT choice , or configuration. One way of concentrating the CO2 is in precombustion technologies, so use of H2 in GT is also included.

A REVIEW OF GAS TURBINES AND THEIR ABILITY TO USE HYDROGEN-CONTAINING FUEL GAS

Report for

Energy Technologies Institute

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EXECUTIVE SUMMARY

This report reviews the performance and costs of existing commercial gas turbines and the capability of gas turbines to operate using hydrogen-containing fuel gases. The report was commissioned by ETI to provide background information for a project they will undertake on salt caverns for storage of hydrogen for use in gas turbine power plants.

The efficiencies of gas turbine have increased in recent years to as high as 42% for simple cycle gas turbines and 62% for combined cycle plants. Further increases are expected in future, for example a major turbine manufacturer has suggested combined cycle efficiencies will increase toward 65% by the early 2020s.

Along with the efficiency increases, the power outputs of the largest gas turbines have increased substantially and the largest single unit combined cycle plant that has been operated has a power output of over 600MW and a model with an output of over 780MW is being offered.

The efficiency of gas turbines decreases at part load and the rate of decrease is greater at lower loads. Typical efficiencies at 50% load are around 80% of the full load value for simple cycle gas turbines and 90% for combined cycle plants.

Gas turbine manufacturers are responding to the need for greater operating flexibility and substantial improvements in start-up times have been achieved. Hot start up times of combined cycle plants based on large frame gas turbines have approximately halved since the early plants, to typically around 30-70 minutes. Cold start times to full output are typically around three times longer than hot start times. Aero-derivative turbines have shorter start times than heavy frame gas turbines, typically around 10 minutes for simple cycle plants. Ramp rates are highly turbine specific. Aero-derivative turbines tend to have high ramp rates.

The minimum continuous operating load of gas turbines is usually constrained by increasing environmental emissions, especially of CO which increases rapidly at low loads. Minimum loads are typically around 30-50% for simple cycle frame gas turbines and 40-60% for combined cycle plants but the minimum loads vary between different turbines. Some turbines have lower minimum loads, the lowest being 10% for simple cycle and 15% for a combined cycle plant.

Most modern gas turbines use dry low-NO_x combustors to limit emissions. Water injection is an alternative that is used particularly in some aero-derivative turbines but it reduces the efficiency of simple cycle gas turbines by around 0.5-2 percentage points. Steam injection is another alternative, which has the advantage of boosting the power output. It typically increases the efficiency of a simple cycle plant but reduces the efficiency of a combined cycle.

The efficiencies of simple cycle gas turbines can be increased by recuperation, i.e. using heat from the turbine exhaust to heat the compressed air. This is used in one small commercial gas turbine and one medium sized marine gas turbine but it is not common except in micro-turbines.

Capital costs of gas turbine power plants are highly site specific. Costs per kW decrease substantially at larger sizes but costs almost level off at sizes greater than about 200MW for simple cycle plants and 400MW for combined cycle plants.

Reciprocating gas engines have higher efficiencies than simple cycle gas turbines, typically 48-50% for engines in the 2.5-20MW range but power plants consisting of multiple gas engines have higher capital costs than plants based on smaller numbers of larger gas turbines, by around 10-15%.

A wide range of gas turbines are reported by manufacturers to be suitable for fuel gases that contain hydrogen. There is significant experience of using gases that contain mixtures of mainly hydrogen, methane and other hydrocarbon gases, especially refinery off-gases and coke oven gas, which typically contains 50-60%vol H_2. Gases with H_2 concentrations of up to 95% are reported to be used. There is also experience of using syngas from gasification which typically contains 25-50%vol H_2 but the other main constituent is CO, which has substantially different properties to methane.

Use of fuel gas containing H_2 presents some significant technical challenges for gas turbines but also some potential benefits.

The biggest technical challenge is reported to be the high flame speed of H_2, which can result in flashback, although it reduces the risk of blowout. The properties of hydrogen-methane mixtures in gas turbines combustors vary non-linearly with concentration. It is reported that only when hydrogen becomes the main constituent is there a large variation in the laminar flame speed.

Lean premix dry low-NO_x combustors, used in most modern gas turbines, are more prone to combustion instabilities than diffusion combustors. Indications are that H_2 can have a positive or negative impact on instabilities, depending on factors such as combustor geometry and design.

The flammability limit is wider for hydrogen than for methane. Addition of modest amounts of hydrogen to methane may enable dry low-NO_x combustors to operate at leaner conditions, i.e. at lower flame temperatures, without extinguishing the flame, which should reduce NO_x production.

If high purity hydrogen was used, the absence of carbon-containing compounds would mean that there would be no emission of CO, which is a significant constraint on gas turbine combustor design and operation, particularly at low loads.

Gases with up to 30% H_2 can be used in dry low-NOx combustors in some commercial turbines and some tests with higher percentages of H_2 have been carried out successfully. However, most commercial gas turbines that use hydrogen-containing gases employ diffusion combustors.

The stoichiometric flame temperature of H_2 is about 150K higher than that of methane. As the production of NO_x in a diffusion flame increases strongly with increasing temperature, this results in higher NO_x emissions unless a diluent (nitrogen, steam or water) is used to reduce the temperature.

Use of steam or nitrogen requires some changes to the operation or design of a gas turbine, to accommodate the increased mass flow through the expansion turbine without causing problems in the compressor and elsewhere in the turbine.

A hydrogen-fired combined cycle plant with dry low-NO_x combustors would have an efficiency about 0.7 percentage points higher than that of a natural gas fired plant based on the same type of turbine.

Hydrogen-fired combined cycle plants using nitrogen or steam to control NO_x emissions would have efficiencies around 0-0.4 and 1.0-1.3 percentage points respectively lower than a natural gas fired combined cycle plant, depending on how the turbine is designed and operated. In a simple cycle plant, the use of steam would result in a significantly higher efficiency but the need for a heat recovery boiler may reduce the operating flexibility.

1. INTRODUCTION

The Energy Technologies Institute (ETI) is focused on accelerating the deployment of affordable, secure low-carbon energy systems for 2020 to 2050. The ETI has identified the potential of using salt caverns to store hydrogen for use as a fuel for power generation during times of peak electricity demand. A high level study has shown that the use of salt caverns would reduce the investment in clean power station capacity and increase the average efficiency of a responsive power system in the UK (ETI, 2015).

Following on from this study, the ETI has recently issued a request for proposals for a more detailed project. Within that project the ETI wishes to identify existing salt caverns in three UK regions that can be utilised in a transition mode from methane to full hydrogen operation. The end goal is to understand the capabilities and costs to create and operate these stores on methane/hydrogen mixtures up to pure hydrogen. The costings developed will include the creation and all installation/plant items required to operate this energy store excluding the hydrogen production plants. This will support a larger piece of work ETI intends undertaking to bring this whole system together as a cost efficient design solution for operation in the UK electricity generation system.

In order to provide background information for the project that it is about to start, the ETI has commissioned this review and database of existing gas turbines and the use of hydrogen-rich gases in gas turbines.

This report consists of the flowing sections:

- An overview of gas turbines which describes the different types of gas turbine that are commercially available, how the performance of turbines has improved over the years and some alternative types of turbine that are used in relatively small numbers, are under development or have been proposed. It also describes start-up and part load operation in general and emission control technologies.
- A description of the gas turbine database and a discussion of the information contained within it. The database includes a comprehensive summary of modern gas turbines from major manufacturers with power outputs greater than $5MW_e$ and combined cycle plants with power outputs greater than $30MW_e$. For each turbine and combined cycle plant the data base includes the manufacturer, model and type of gas turbine and its power output, efficiency, mass flow, pressure ratio and exhaust temperature. Where available, start times, minimum loads and ramp rates are also included. Part load efficiency data are also provided for selected turbines and combined cycle plants.
- A summary of the capital and operating costs of gas turbines and combined cycle power plants.
- A discussion of the impacts of using hydrogen on various aspects of the design and operation of gas turbines. The capability of specific commercial gas turbines to use fuel gases containing hydrogen is also presented.
- A brief overview of reciprocating engines that are suitable for grid-based gas fired power generation, as an alternative to gas turbines.

2. OVERVIEW OF GAS TURBINES

2.1 Brayton cycle

Gas turbines employ the Brayton cycle which is shown in an idealised form in Figure 1. Gas is compressed at constant entropy (1-2 in the figure), it is heated at constant pressure (2-3), expanded at constant entropy (3-4) and cooled at constant pressure (4-5). This is an example of a "closed cycle" in which the same material flows around the cycle and heat is added and withdrawn through heat exchangers. The commercially dominant gas turbines are however "open cycle" in which the fluid that is compressed is air from the atmosphere, the heating is carried out by combustion of fuel gas in the compressed air and the expanded gas is exhausted to the atmosphere (either directly or after passing through a heat recovery steam generator) rather than being cooled and reused within the cycle. In "semi closed" cycles, which are discussed in section 2.4.6, some of the expanded gas is cooled and recycled to the compressor and some is withdrawn from the cycle.

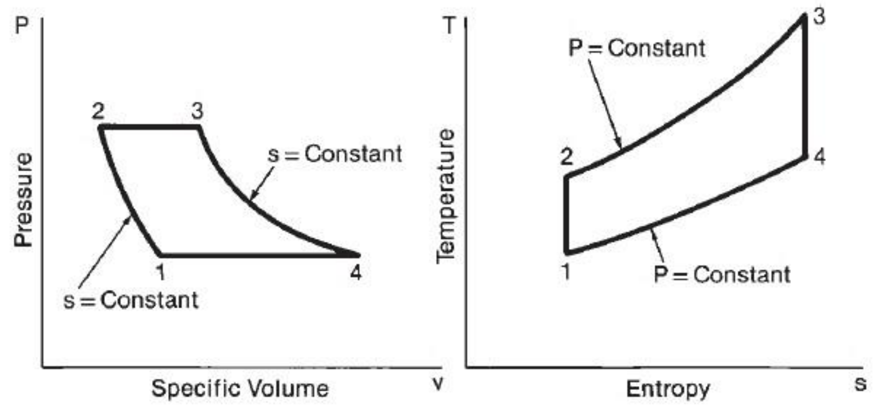


Figure 1 Idealised Brayton cycle

In the idealised Figure 1, compression and expansion are at constant entropy. In practice the entropy increases in both the compression and expansion due to inefficiencies. A further inefficiency arises because some of the compressed air has to be used to cool the hot turbine components, including the high temperature turbine blades, rather than being heated in the combustor.

Gas turbines can be employed as a "simple cycle" consisting of just a Brayton cycle or they can be combined with a steam Rankine cycle to create a "combined cycle". The exhaust gas of modern gas turbines is typically at a temperature in the range of 400-650C. In a combined cycle energy is recovered from this gas in a heat recovery steam generator (HRSG) which generates steam which is expanded in a steam turbine to generate additional power. In order to maximise the power output and thermal efficiency, steam is usually generated at either two or three different pressures. Older and smaller gas turbines tend to have steam systems in which steam is generated at two different pressures. Larger, more modern gas turbines tend to have three pressure steam systems. Three pressure steam systems normally include reheat, in which the high pressure superheated steam is partly expanded in a high pressure steam turbine, returned to the HRSG where it is reheated, and then it is sent back to complete its expansion in the medium and low pressure section of the steam turbine.

The temperature and quantity of steam produced can be increased by using in-duct firing of the HRSG, in which some additional fuel is combusted. This can be a useful feature for combined heat and power plants where there is a need to vary the amount of steam generation but it usually reduces the overall efficiency of a power-only plant. It can also be a useful technique for generating peak power.

Each gas turbine is normally connected to its own HRSG but the steam generated in more than one HRSG can be combined and fed to a common steam turbine. This usually results in a marginal increase in the overall efficiency of a combined cycle plant, as shown in the database.

2.2 Types of commercial gas turbine

Power generation gas turbines are often classified as “heavy frame” or “aero-derivative”. Frame gas turbines are designed specifically for land-based power generation or mechanical drives. They are usually built in a similar way to large steam turbines and the casing is split horizontally, which enables the turbine to be opened up for on-site maintenance. Aero-derivative gas turbines are derived from aircraft jet engines and hence they employ lighter weight construction. They consist of a core compressor/combustor/turbine section of an aero engine combined with further stages of low pressure expansion turbine for power generation. Aero-derivative gas turbines are designed for variable operation, which is an essential requirement for aero engines and they are designed for quick replacement of the entire engine when significant maintenance is required.

The pressure ratios of most frame gas turbines are in the range of around 12-24. Aero-derivative power generation turbines typically have relatively higher pressure ratios, up to 42. The high pressure ratios of aero-derivative turbines makes them particularly well suited to simple cycle power plants because the optimum pressure ratio to achieve high thermal efficiency is greater for simple cycles than combined cycles. As a consequence of the higher pressure ratios, the turbine exhaust temperatures of aero-derivative turbines are relatively low, typically around 450C, compared to around 630C for the latest large frame turbines, The temperature of steam that can be generated is therefore lower and the overall efficiencies of combined cycles based on aero-derivative gas turbines are usually lower than those of large frame gas turbines.

Gas turbines can be classified according how many separate shafts they have. The largest frame gas turbine are all single shaft machines, i.e. the compressor, expansion turbine and electrical generator are all connected to one shaft which rotates at a speed which depends on the frequency of the electricity system. For those countries or regions which have a 50Hz electricity system, which includes the UK, the rotational speed is 3000rpm and for 60Hz systems the speed is 3600rpm. Different models of gas turbines are manufactured to operate at these two speeds, although in many cases they are aerodynamically scaled versions of the same basic design. 50Hz turbines have larger power outputs, typically by a factor of around 1.4-1.5. Most aero-derivative gas turbines and some smaller frame turbines are two shaft machines, consisting of a compressor-turbine core which usually operates at a higher rotational speed, unconstrained by the frequency of the electricity system, and a power turbine and generator on a separate shaft which operates at a different speed. This means that the same core can be used for 50 and 60Hz turbines. Having two shafts can be beneficial for operation and efficiency at part load, because the speed of the core compressor can be reduced, which enables it to operate more efficiently at part load.

Some small and medium sized turbines use a gearbox, which enables the power turbine to operate at an optimum rotational speed unconstrained by the speed required by the generator but these benefits have to be offset against the extra cost, mechanical losses and maintenance requirements of the gearbox.

2.3 Technical developments in commercial gas turbines

Basic thermodynamics dictates that increasing the top temperature of a power generation cycle results in a higher efficiency. Much of the development work on gas turbines has therefore focussed on increasing the turbine inlet temperature. Simply increasing the inlet temperature of a gas turbine however does not necessarily increase the efficiency because the amount of gas needed to cool the hot components can because excessive and offset the benefits of the higher inlet temperature. Various techniques are therefore being used to enable turbine inlet temperatures to be increased while avoiding the need for excessive turbine cooling gas requirements:

- Metals capable of operating at higher temperatures
- Thermal barrier coatings
- More efficient turbine cooling techniques
- Ceramic based components, such as ceramic matrix composites (CMCs)

The ways in which these improvements have and are continuing to enable increases in turbine inlet temperatures are illustrated in Figure 2 (University of Virginia, 2016).

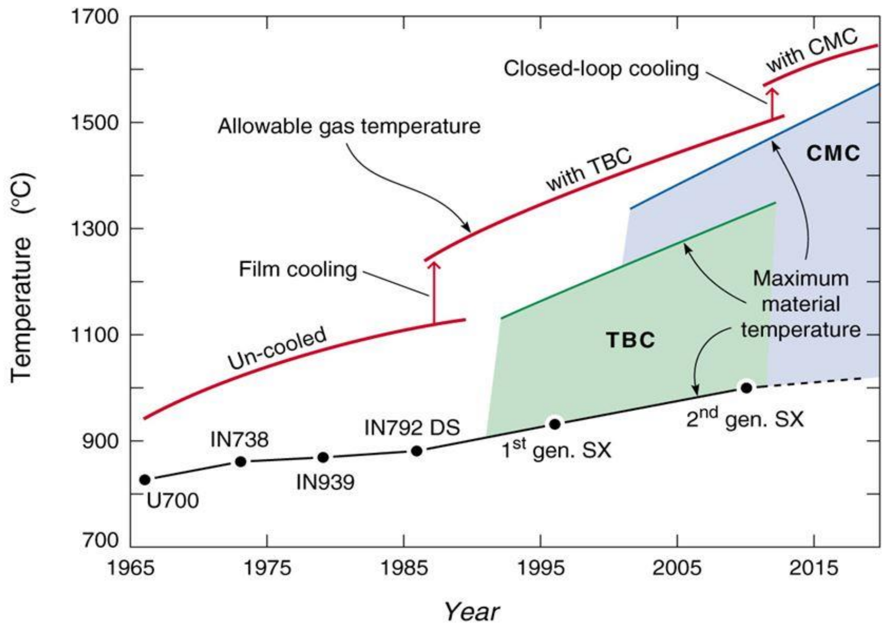


Figure 2 Evolution of gas turbine materials and turbine inlet temperatures
Source: Wadley Research Group, University of Virginia

Another significant contribution to higher efficiencies of gas turbines is improvements to the aerodynamic design of compressor and turbine blades. The availability of increasingly detailed and sophisticated aerodynamic modelling has been a major contributing factor to these improvements.

As an illustration of how the efficiencies of turbines have increased, Figure 3 shows information from one of the major manufacturers, Mitsubishi Hitachi Power Systems (Ai, 2015). The M701D turbine with an inlet temperature of 1100C, which was introduced in 1984, had a combined cycle efficiency of less than 50%. Their latest J class turbines with an inlet temperature of 1600C have a combined cycle efficiency of over 61%. Broadly similar improvements have been achieved by other vendors. Further improvements should enable combined cycles to reach higher efficiencies in future, for example GE's president and CEO of gas power systems has suggested the efficiency could increase towards 65% by the early 2020s (Larson, 2016).

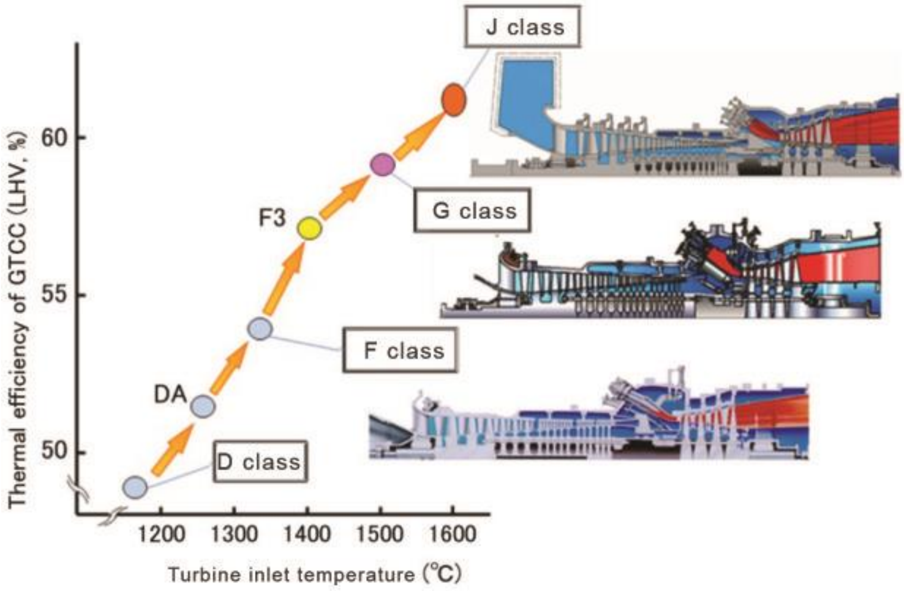


Figure 3 Increases in gas turbine inlet temperatures and efficiencies
Source: Mitsubishi Hitachi Power Systems

The optimum pressure ratio of a gas turbine increases as the turbine inlet temperature increases. For this reason the pressure ratios of commercial gas turbines have also increased, for example from around 13 in a typical E class frame gas turbine to around 23 in a typical H class turbine.

The power outputs of the largest frame gas turbines have also increased substantially over the years, partly due to greater mass flow and partly because of the higher specific power (MW/kg/s of mass flow) that results from higher inlet temperatures and efficiencies. For example, the largest current gas turbine, the GE9HA.02, has a mass flow that is 2.35 times greater than that of GE's 9E gas turbine from the early 1990s and the specific power is 1.65 times greater, resulting in a power output that is almost four times greater.

When gas turbines started to be used for large scale power generation they were focussed mainly on base load power generation in combined cycle plants, so efficiency was the most important criterion. More recently gas turbine combined cycle plants have increasingly been called upon to operate flexibly and at lower annual load factors, in order to meet the variability in consumer power demand. In addition, variable renewable power generation technologies such as wind and solar power are being introduced on a large scale in many countries, which is resulting in the need for gas turbine power plants to operate with even greater flexibility, with more frequent start-ups and shut-down, faster ramping and the ability to operate at low loads with high efficiencies and low emissions. The impact of the need for greater flexibility is illustrated by the large high efficiency gas turbines developed by GE and Mitsubishi Hitachi Power Systems. GE's first H class gas turbine, which entered service in 2002 at a plant at Baglan Bay in Wales, employed closed circuit steam cooling to maximise efficiency. Steam from the exhaust of the high pressure steam turbine is used to cool hot components in the gas turbine and the heated steam is returned to the intermediate pressure steam turbine. In contrast, GE's latest 9HA turbine, the first commercial example of which started up in 2016 in France, uses air cooling of the turbine. The lower degree of integration between the gas turbine and steam system increases operating flexibility. Similarly, Mitsubishi Hitachi Power Systems' 701J gas turbine employs steam cooling but an air cooled variant, the 701JAC, was introduced later, particularly to meet the requirement of regions where there is a need high flexibility.

2.4 Alternative gas turbine features and novel cycles

2.4.1 Reheat combustion

Almost all commercial gas turbine designs employ a single stage of combustion. This corresponds to the Brayton cycle shown in Figure 1. The only exception is the Ansaldo GT26 (and its 60Hz equivalent, the GT24) which is a reheat gas turbine with two sequential stages of combustion separated by a high pressure turbine expansion stage. This is analogous to the reheat steam turbines that are used in most large modern coal fired power plants to maximise efficiency. For the same turbine inlet temperature and component efficiencies a reheat gas turbine should have a higher efficiency than a single combustor turbine. The optimum pressure ratio of a reheat gas turbine is substantially higher than that of a single combustor turbine. The pressure ratio of the GT26 is 35, which is around twice that of single combustor turbines with comparable inlet temperatures. The benefits of reheat need to be balanced against the extra complexity, although reheat may have some flexibility advantages, as discussed in section 3.

2.4.2 Compressor air cooling

A large fraction (typically 50-70%) of the power generated by the expansion stage of a gas turbine is consumed by the compressor. The power consumption of the compressor is roughly proportional to the volume of air passing through it, which in turn is proportional to its absolute temperature. A way to reduce the power consumption of the compressor is by cooling the air, either at the inlet to the compressor or part way through it (inter-cooling). Air cooling also has the advantage of reducing the compressor exit temperature, which may reduce the need for more expensive materials of construction. It also increases the mass flow rate of the compressor, which increases the power output of the overall gas turbine. A downside of the lower compressor exit temperature is that the fuel consumption of the combustor increases, so there is a trade off between reduced air compressor power consumption and increased fuel consumption. Another downside of compressor air cooling is increased complexity and, in some cases, the need for a cooling system.

Figure 4 shows an example of how the power output, exhaust flow rate and heat rate (inverse of efficiency) of a large F class frame turbine (the Mitsubishi Hitachi Power Systems 701F series) varies according to the compressor inlet temperature. Reducing the air inlet temperature results in substantial increases in flow rate and power output and a small reduction in heat rate (i.e. an increase in the thermal efficiency). The sensitivity to inlet temperature is different for each model of gas turbine. Manufacturers' information on the sensitivity of gas turbine performance to air temperature and pressure is available in the public domain for many commercial gas turbines. Reducing the compressor inlet air temperature tends to be more advantageous for aero-derivative turbines than for frame turbines, because they typically have higher pressure ratios (GTW, 2010).

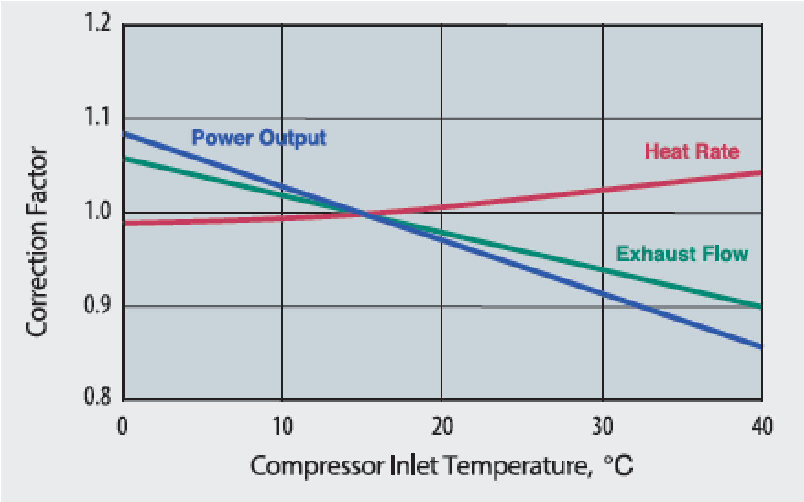


Figure 4 Sensitivity of gas turbine performance to compressor inlet temperature
Source: MHPS, 2014©

The temperature of gas turbine compressor air can be reduced by chilling or water injection.

Inlet air chilling

The compressor inlet air can be cooled by passing it through a heat exchanger where heat is transferred to a chilled heat transfer fluid. The heat transfer fluid can be cooled using an electrically driven mechanical chiller or an absorption chiller that makes use of hot water or steam. An advantage of this approach is that it is not limited by the humidity of the air, the air can be cooled below its wet bulb temperature and there is no requirement for clean injection water.

Water injection

An alternative to inlet air chilling is to inject water into the compressor inlet air, to reduce the air temperature by evaporation. Water can be added to the inlet air by use of wetted media, by fogging or by wet compression. The first of these techniques involves passing the air across a wetted honeycomb-type medium from which water is evaporated, thereby cooling the air. Fogging consists of spraying very fine droplets of water into the inlet air stream. The droplets evaporate to cool the air in a similar way to the wetted media system. Fogging can be controlled to produce various sizes of droplets depending on the ambient temperature and humidity. For wet compression, more finely atomised water is sprayed into the compressor inlet air. The amount of water that is injected is typically three to four or more times the amount of water that is evaporated in the inlet cooling techniques described above (GTW, 2010). The excess water fog is carried forward into the compressor where is evaporates and provides cooling of the air as it passes through the compressor. Fogging and wet compression require the use of high purity water to reduce the risk of formation of deposits.

Water injection has the greatest impact in hot dry climates, although the places that have such climates are often places where water availability is limited. In hot countries where the use of air conditioning is widespread, the peak power demand tends to coincide with the times when ambient temperatures are at their highest. The ability to avoid a derating of the power output of a gas turbine at such times makes water injection and saturation of the compressor air inlet particularly advantageous. Water injection and saturation of the compressor inlet air is less relevant in the UK, where the peak power demand is in winter, when ambient air temperatures are relatively low and humidity levels are usually relatively high. Inlet spray cooling is relatively ineffective in such condition. This is illustrated by Figure 5, which shows that inlet spray inter-cooling reduces the heat rate and increases the power output of the Siemens Trent 60 gas turbine at high temperatures but there is no effect below about 7C.

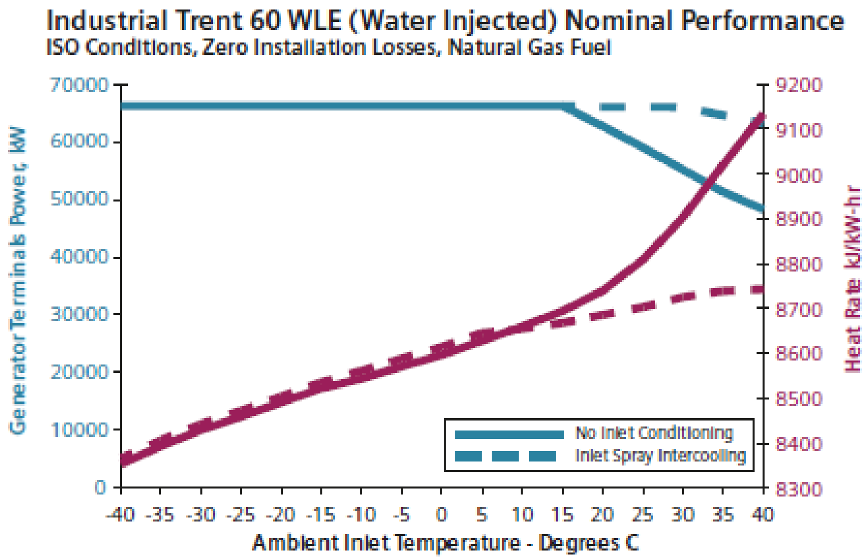


Figure 5 Impact of inlet spray inter-cooling on power output and heat rate
Source: Siemens, 2014©

Compressor inter-cooling

The temperature of air increases as it passes through a gas turbine compressor. A way of reducing the compressor power consumption is by inter-cooling the air part way through the compressor, which is a technique that is commonly used in industrial gas compression. Inter-cooling is however rarely used in current power generation gas turbines. A prominent exception is GE's LMS100 aero-derivative turbine. Inter-cooling is most advantageous in turbines that have high pressure ratios, and the LMS 100 has a pressure ratio of 42.5, the highest of any of the turbines in the database.

2.4.3 Combustor steam injection

Water or steam can be injected into a gas turbine combustor to reduce NOx emissions, which is discussed in section 2.5, and to increase mass flow rate and power output. Steam injection normally increases the efficiency of a simple cycle but does not necessarily increase the efficiency of a combined cycle because it may be more efficient to expand the steam in a steam turbine. A disadvantage, as with compressor water injection, is that the water or steam that is injected is lost to the atmosphere as water vapour in the turbine exhaust gas unless a flue gas cooler and water recovery system is installed. The quantity of water that is lost may be lower than the quantity that is lost in a wet cooling tower of a combined cycle plant but the injected water needs to be high purity and the cost of water treating is an additional burden.

Steam injection is particularly well suited to smaller gas turbines. For some small turbines the cost and complexity of a combined cycle cannot be justified. In such cases a relatively simple gas turbine exhaust steam generator can be used to produce modest pressure steam that can be injected into the turbine combustor to provide additional mass flow through the expansion turbine and hence higher power output. Steam injected gas turbines are reported to have a smaller footprint, shorter construction time, lower capital cost and better operating flexibility than an equivalent combined cycle plant (MHPS, 2016b). The capability to inject steam into the gas turbine can be useful in combined heat and power (CHP) plants, to help balance varying demands for steam and power.

It is reported that most gas turbines can accommodate a steam flow equal to 5% of the compressor air with some turbines being able to accommodate greater amounts, and that 5% steam injection will increase the power output by about 17.5% (OSTI, 2012). The quantity of steam that can be injected into a gas turbine is limited by the compressor surge margin, which is different for different gas turbines.

The Siemens 501KH5 is a well established gas turbine designed for steam injection. The power output is 6.5MW, which is 65% greater than the non-steam injected equivalent and the net efficiency is 41.9% compared to 30.6% for the non-steam injected equivalent.

Another example of a steam injected gas turbine is Mitsubishi Hitachi Power Systems' Smart Advanced Humid Air Turbine (AHAT), shown in Figure 6.

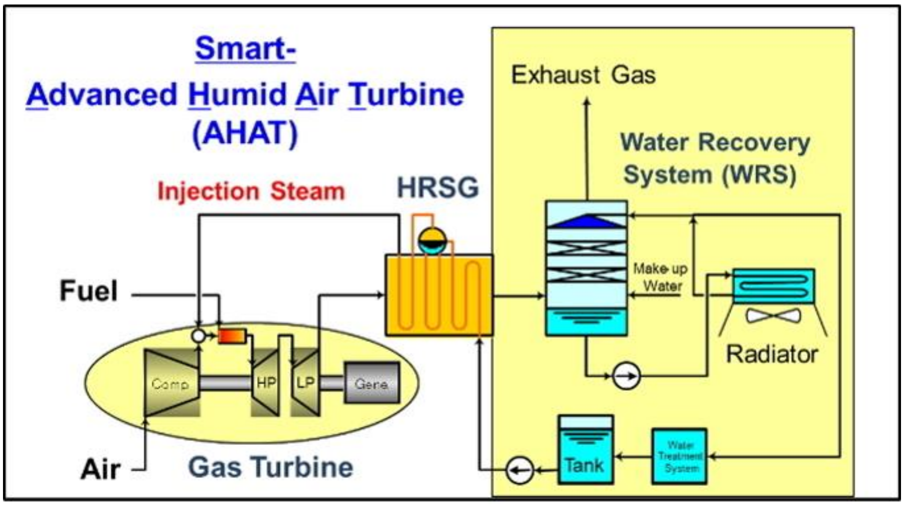


Figure 6 Smart Advanced Humid Air Turbine
Source: Mitsubishi Hitachi Power Systems, 2016©

MHPS' Smart AHAT is based around their H-50 frame gas turbine. The H-50 is specifically designed to accommodate a large amount of injection steam. Its rated power output is 57MW without steam injection and this increases to 70MW with steam injection, i.e. a 23% increase. The efficiency with steam injection is 45%, which is significantly higher than the 37.8% efficiency of the non-steam injected H-50 and which is also higher than any comparable simple cycle turbine (MHPS, 2016b).

A disadvantage of steam injected gas turbines, in common with compressor water injection, is that the injected steam is lost to the atmosphere in the gas turbine exhaust gas. A large amount of demineralised make-up water is needed to compensate for this loss. MHPS's AHAT includes a water recovery system in which the turbine exhaust gas is contacted with sprays of recirculating cooled water which condenses most of the steam from the turbine exhaust gas. The recirculating water from the heat recovery system is then cooled in an air cooler or alternatively a wet cooling tower, although using a wet cooling tower would result in water loss to the atmosphere, negating some of the benefits of the water recovery system. The amount of water lost in the cooled turbine exhaust gas depends on ambient conditions. In some conditions the plant can become a net producer of water, i.e. some of the water produced by combustion as well as all of the injected steam is recovered.

Water can be injected into the combustor instead of steam but this is rarely practiced (except for NO_x control) because the additional fuel that is needed to provide the heat to evaporate the water results in a reduction in overall energy efficiency of the gas turbine.

2.4.4 Recuperative gas turbines

In a recuperative gas turbine, also known as a regenerative gas turbine, the turbine exhaust gas is passed through a heat exchanger where heat is transferred to the high pressure air from the compressor, as shown in Figure 7. ETI has expressed a particular interest in recuperated gas turbines, so they are described at greater length in this report.

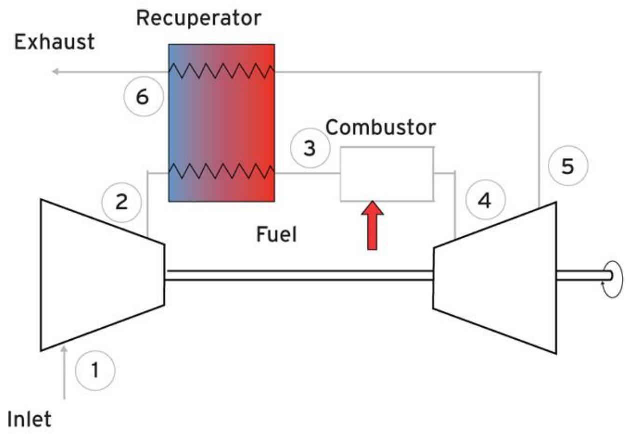


Figure 7 Recuperated gas turbine
Source: Ricardo

The heat transferred to the compressed air reduces the need for fuel in the combustor, thereby increasing the thermal efficiency of a simple cycle. Recuperation also reduces the amount and temperature of heat available for a combined cycle, so the benefits of recuperation in a combined cycle are lower. The amount of heat that can be transferred in the recuperator depends on the temperature difference between the turbine and compressor exit gases. Some recuperated gas turbines have a low pressure ratio, which results in a relatively low compressor exit temperature and high heat recovery from the turbine exhaust gas. When higher pressures ratios are used, compressor inter-cooling is normally employed, which decreases the compressor exit temperature and hence increases the amount of heat that can be transferred in the recuperator. Recuperation is not widely used in large gas turbines but it is conventional in micro-turbines and it enables such machines to achieve high efficiencies despite their small size, for example 33% efficiency in a 200kW turbine (Capstone, 2016). In micro turbines the recuperator is typically an integral part of the machine, as shown in Figure 8.

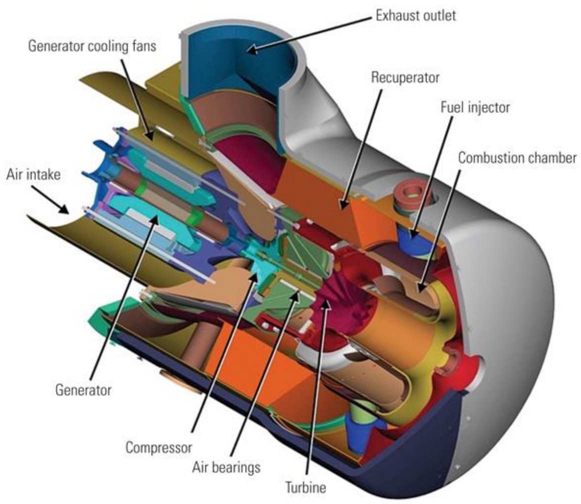


Figure 8 Micro turbine with integral recuperator
Source: Powermag (Gillette, 2010)

The only frame type gas turbine with recuperation that is currently on the market is the Solar Mercury 50, which has a power output of 4.6MW. This is just below the 5MW lower threshold set by the ETI for this study but because of ETI's interest in recuperated gas turbine it has been included in the database. This turbine has an efficiency of 38.5 which is 4-7 percentage points higher than Solar's other <20MW non-recuperated turbines. The optimum pressure ratio of recuperated gas turbines is lower than that of non-recuperated turbines. The pressure ratio of the Mercury 50 is 10 while the pressure ratios of other 5-15 MW turbines in the database are 12-18. The need to accommodate a recuperator means that the orientation of the compressor and turbine in the Mercury 50 is different to that of a conventional gas turbine, as shown in Figure 9.

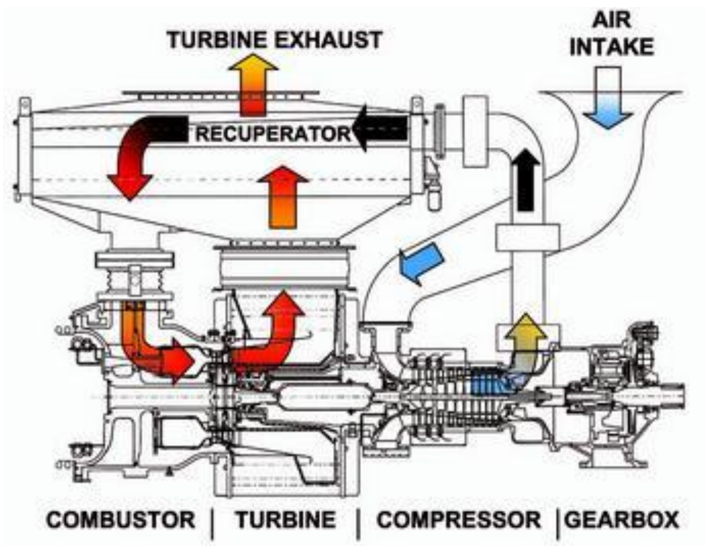


Figure 9 Solar Mercury 50 recuperated gas turbine
Source: Modern Power Systems, 2004

The largest recuperated gas turbine that has been developed in recent years is the Rolls-Royce WR21, which is an intercooled-recuperated turbine built around an RB211/Trent core. The WR21's plate-fin recuperator and the intercooler are made by Northrop Grumman. The WR21 was introduced in 1997 but it is only being used as a military marine engine, in the UK's Type 45 destroyers. The WR21 has a shaft power output of around 25MW, a pressure ratio of 16.2, an efficiency of 42% and an exhaust temperature of 355C (GTW, 2016). The recuperator, along with variable area turbine nozzles, enables the turbine inlet temperature to be maintained at part load. The turbine has significantly better part load efficiency than comparable non-recuperated engines, which is important for warships that operate most of their time at much less than maximum speed. The efficiency at 50% load is essentially the same as at full load and even at 20% load the efficiency is still around 80% of the full load efficiency (English, 2000).

2.4.5 Recuperative humid air turbine cycles

Humid Air Turbine (HAT) cycles that include a recuperator, a compressor intercooler and a saturator to add water vapour to the compressor discharge air have been proposed. An example of such a cycle is shown in Figure 10. The inlet gas to the combustor contains approximately 20% moisture. Use of a saturator is thermodynamically more efficient than generating pure steam in a steam generator and then adding the steam to the air, because the steam evaporates not at its pure component vapour pressure but instead at its partial pressure in the system. This makes it possible to utilise lower temperature heat for evaporation.

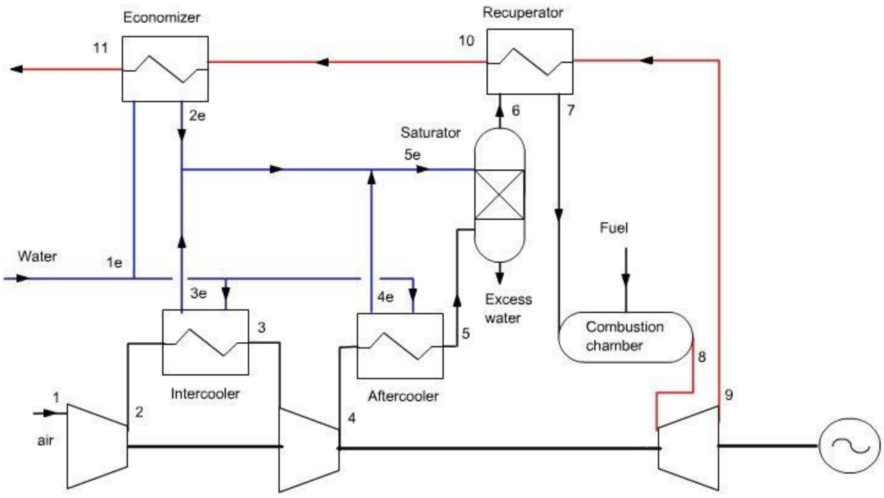


Figure 10 An example of a Humid Air Turbine (HAT) cycle
Source: ParisTech, 2016

Recuperated HAT cycles such as this have better part load performance than combined cycles. It is reported that the heat rate remains essentially constant down to 60% load and increases by only 35% at 20% load whereas in a combined cycle it increases by as much as 70% (Rao, 1991).

Such advanced HAT cycles were studied extensively in the 1990s, particularly in the context of coal gasification power plants (EPRI, 1993). An integrated gasification HAT cycle plant using a turbine derived from the Pratt and Whitney FT-4000 aero-derivative gas turbine was shown to have a heat rate comparable to that of an IGCC plant but with a lower capital cost. A natural gas fired plant had up to a 5% lower heat rate than a combined cycle plant but a higher capital cost, resulting in only a slight reduction in the cost of electricity. Although some components of existing turbines, could be used in advanced HAT cycles, expensive development programmes would have been needed to commercialise large scale turbines and it appears that the advantages of HAT cycles were considered at the time to be insufficient to warrant the expense and commercial risk.

Hitachi has worked on development of a recuperated cycle with a saturator but no turbine inter-cooling as part of the Japanese government's Cool Earth Innovative Energy Technology programme. A 40 MW demonstration plant commenced operation in 2013, with the aim of developing a 100-200MW commercial plant. A plant based on a 100MW gas turbine was reported to have better operating flexibility than a comparable combined cycle plant, with hot and cold start times of 30 and 60 minutes compared to 60 and 180 minutes for a combined cycle, a ramp rate of 8.3-10%/minute (5% for a combined cycle) and a minimum load of 25% (50% for a combined cycle). The reduction in efficiency at part load was also less than for a combined cycle (Gotoh, 2011).

2.4.6 Other novel gas turbines

There is currently significant interest in semi-closed oxy-combustion turbine cycles, in which CO_2 or a combination of H_2O and CO_2 are used as the working fluid and combustion takes place using purified O_2. The main advantage of these cycles is that when burning a carbon-containing fuel they produce an output gas with a high CO_2 concentration that is suitable for underground storage. The ability of oxy-combustion cycles to produce a high concentration CO_2 stream is not relevant to ETI's current interest in hydrogen fired gas turbines.

A review and techno-economic assessment of oxy-combustion turbine cycles was published recently (IEAGHG, 2014). The highest efficiency and lowest cost oxy-combustion turbine cycle was a high pressure recuperated cycle proposed by NET Power that makes use of recycled CO_2 as the working fluid. This cycle (which inherently captures CO_2) is claimed by its developer to have an efficiency comparable to conventional F-class gas turbines without CO_2 capture. It would appear that this cycle could in principle use high-purity hydrogen fuel. The only substance withdrawn from the cycle would then be water and all of the CO_2 would be recycled (a small top-up of CO_2 would be needed to offset fugitive leaks). It should be noted that the hydrogen fuel would need to have very low levels of inert impurities to avoid then building up in the recycle loop. The main relevance of the NET Power cycle however would be as a natural gas-fuelled potential commercial competitor to hydrogen fired conventional gas turbines.

2.5 Gas turbine emission control techniques

CO_2 is a significant emission from gas turbines because of its impact as a greenhouse gas. The CO_2 emission depends on the thermal efficiency and the carbon content of the fuel that is used. The most significant other emissions produced by gas turbines are NO_x, CO and volatile organic compounds (VOC). Sulphur oxides can be a concern when using liquid fuels or sulphur-containing gases and particulate matter can be a marginally significant emission for gas turbines using liquid fuels.

Three techniques are used to limit emissions of NO_x from gas turbines:

- Pre-mixed dry low NO_x combustion
- Dilution in the combustor, mostly by steam, water or nitrogen
- Reduction of pollutants in the turbine exhaust gas, especially by selective catalytic reduction (SCR).

2.5.1 Dry low NO_x combustion

NO_x is produced mainly from atmospheric N_2 and O_2 by the Zeldovich thermal mechanism. Higher temperatures result in substantially greater production NO_x. CO is produced by incomplete combustion.

Turbine inlet temperatures have increased in order to increase thermal efficiency but this tends to increase NO_x production, while at the same time NO_x emission limits have reduced. To meet this challenge, gas turbine combustors have evolved considerably over the years, in particular lean pre-mix (dry low-NO_x) combustors have been developed. Early large gas turbines mostly used large silo combustors, firstly with single diffusion burners and then with multiple dry low-NO_x burners. Later gas turbines use more compact multiple annular and can-annular combustors.

The principle of current dry low-NO_x combustors is to generate a well mixed lean fuel-air mixture prior to entering the combustor. Having a lean mixture results in a low flame temperature, which lowers the rate of NO_x production. A low combustor residence time is also needed to minimise NO production. Gas turbine NO emissions are much lower than the equilibrium value, which for a typical F class gas turbine is about 820ppmv at 15% O_2 (Lieuwen, 2013).

The lean mixture in a gas turbine combustor is close to the lean extinction limit so the fuel-air ratio has to be kept within a narrow band. Another reason why this is necessary is that the lower combustion temperature tends to lead to less complete combustion, resulting in production of CO and unburned hydrocarbons. In contrast to NO_x, CO emissions are above the equilibrium level (e.g. 2ppmv for a typical F class gas turbine), so the need to limit both NO_x and CO leads to conflicting design considerations.

A limitation of lean pre-mix burners is the lean flame stability limit, i.e. the amount of excess air which is permitted for stable combustion. This limit is typically exceeded during start-up and low load operation. Dry low-NO_x combustors in gas turbines typically include a pilot diffusion burner, which is used for start-up and low load operation. Diffusion burners are very stable but they result in high emissions. As the load is increased, premix fuel is introduced spreading the fuel into all of the air and the pilot burner is turned off. The maximum degree of turndown of a gas turbine is usually dictated by increasing emissions of CO. Because of the need for greater operating flexibility and low load operation, gas turbine manufacturers devote considerable effort to development of dry low NO_x combustors that can continue to operate at low load factors.

2.5.2 Dilution in the combustor

The flame temperature, and hence NO_x emissions, can be reduced by injecting a diluent, either steam, water of nitrogen, into the combustor. This is a commonly used technique in turbines firing gases containing hydrogen and it is discussed later in the section on hydrogen fired turbines.

Injecting water into the combustor instead of using dry low-NO_x combustion normally reduces the thermal efficiency of a gas turbine. This can be seen by comparing the information in the database for Siemens Trent gas turbines. The wet low emission variants have efficiencies 1.0-1.8 percentage points lower than the dry low emission (DLE) variants. Similarly water injection variants of GE's aero-derivative LM2500 and LM6000 turbines have efficiencies 1.4-1.9 percentage points lower than DLE variants and for the LMS100 the difference is 0.6 percentage points.

Steam injection was described earlier in the report, in the context of increasing turbine mass flow. Steam injection increases the efficiency of a simple cycle but the efficiency is generally lower than if the steam had been used in a combined cycle.

Steam and/or nitrogen injection into medium and large gas turbines for NO_x reduction is widely used in IGCC plants. The fuel gas in the existing IGCC plants, which do not include CCS, contains a substantial amount of CO. CO has an even higher stoichiometric flame temperature than H_2, so the need for steam or nitrogen addition to the combustor is at least as great. IGCC plants have operated with about 50% nitrogen dilution of the fuel gas or 35% steam dilution, which has enabled NO_x emissions to be reduced to acceptable levels. For example, NO_x emissions at a coal-fired IGCC plant at Buggenum were 6-30 ppm at full load and about 4-20ppm at 40% load (Huth, 1998).

2.5.3 Removal of pollutants from turbine exhaust gas

NO_x

The main technique used to remove NO_x from turbine exhaust gas is Selective Catalytic Reduction (SCR). In SCR, ammonia is injected into the turbine exhaust gas and it reacts with NO in the presence of a fixed bed of catalyst to produce N_2 and H_2O. The most common catalysts are vanadium or titanium based, on a ceramic support. SCR can reduce NO_x in gas turbine exhaust gas by 80-90%, depending on the degree to which the chemical conditions in the exhaust gas are uniform. When used in series with water/steam injection or dry low-NO_x combustion, low single digit NO_x emissions (1.5-5 ppm) can be achieved (USEPA, 2015).

It would not be realistic to expect that SCR could be used to reduce NO_x emissions from the very high levels that would be produced by diffusion combustors without diluent addition, due to high costs of reagent and catalyst, but it could be used to enable hydrogen fired turbines to meet the increasingly stringent NO_x emission regulations which may be difficult and expensive to achieve by combustor diluent addition alone.

The operating temperature of SCR systems depends on the type of catalyst and the flue gas composition. The operating temperature range has traditionally been around 200-425C. The exhaust temperature of modern frame gas turbines is usually above this temperature range, as can be seen from the turbines database, but this is not a problem for combined cycle plants because the SCR unit can be contained within the HRSG at an appropriate temperature. The difference between turbine exhaust and SCR temperatures is more of a concern for simple cycle plants. Cooling systems (air or water) can be used to reduce the gas temperature but there are practical limitations on how much cooling can be applied and the possibility of failure of the cooling system resulting in irreparable damage to the catalyst needs to be considered, as well as the additional cost and complexity. Aero-derivative gas turbines tend to have lower exhaust temperatures than frame gas turbines and some of them are within the range for SCR operating temperatures, which makes it easier to apply SCR to aero-derivative turbines in simple cycle power plants.

"Hot" SCR catalysts, typically zeolite based, have more recently become available which makes SCR a more feasible option for simple cycle gas turbines, especially frame-type machines. For example, a catalyst from BASF is able to operate at up to 580C (BASF, 2007), although this is still lower than the exhaust temperature of some frame turbines, so some cooling would still be needed. Hot SCR catalysts are however reported to be more expensive, less efficient and less durable than lower temperature catalysts (Chupka, 2013).

Although SCR reduces NO_x emissions it results in some emission of unreacted ammonia, which is referred to as "ammonia slip". This is due to the non-uniform distribution of the reacting gases, both the NO_x in the turbine exhaust gas and the injected ammonia. Typical values of ammonia slip are about 5ppm (Lieuwen, 2013). Ammonia emissions lead to increased quantities of fine particulates through reactions in the atmosphere. Another concern regarding SCR is the need for on-site storage and handling of ammonia, which is a hazardous chemical. Aqueous ammonia or urea can be used to reduce hazards.

SCR catalysts have a finite lifetime and have to be replaced when no longer effective and/or ammonia slip reaches unacceptable levels. Catalysts can contain heavy metals such as vanadium and/or titanium, which results in potential health and environmental concerns related to disposal of spent catalyst. Vanadium pentoxide is classed as an extremely hazardous material (Scorr, 1999).

SCR is best suited to base load operation because turbine exhaust temperatures become lower at low loads, as described earlier, and the SCR reactions are sensitive to temperature. Close matching of the ammonia injection rate and turbine exhaust flow rate are needed to avoid lower NO_x abatement or higher ammonia slippage rates.

CO oxidation

Oxidation catalysts promote the reaction of O_2 that is present in turbine exhaust gas with CO and hydrocarbons to produce CO_2 and water. No reactants need to be added. CO oxidation catalysts are usually made of platinum, palladium or rhodium. Emissions of CO are reduced by approximately 90%. The positioning of CO and SCR catalysts in an HRSG depends on the particular catalysts and their optimum operating temperature. The classical positioning of CO catalyst is upstream of the SCR, where the high temperature maximises catalyst activity and minimises the quantity of catalyst. Palladium-based catalysts also oxidise ammonia into molecular nitrogen and may be fitted after SCR catalyst to remove ammonia-slip (Jakobsson). The concentration of CO in gas turbine exhaust gas varies strongly with load and the highest concentrations usually occur at low loads. The percentage conversion of CO is almost independent of the concentration of CO (Jakobsson).

Catalytic absorption

An alternative process for reduction of NO_x and CO emissions is the $SCONO_x$™ process which can reduce NO_x emissions to less than 2.5ppm and almost completely remove CO. In this process CO and NO are catalytically oxidised to CO_2 and NO_2. The NO_2 is subsequently absorbed on the treated surface of the catalyst, which is coated with potassium carbonate and platinum. The resulting potassium nitrites and nitrates are then reconverted to potassium carbonate through a regeneration process that involves passing a mixture of regeneration gas (H_2 and CO_2) across the surface of the catalyst in the absence of oxygen. The catalyst is divided into sections and a set of dampers is located upstream of each section to achieve the required oxygen free environment. The system operates at a temperature within a range of 150 and 370C.

The $SCONO_x$™ process does not use ammonia reagent, so there is no ammonia slip. The $SCONO_x$™ technology is still in the early stages of market introduction. Although it can achieve very low emission levels, there are issues of concern, including its relatively high capital cost, system complexity and high demand for utilities (steam, natural gas, compressed air and electricity are required), and a gradual rise in NO emissions over time (USEPA, 2015).

2.6 Gas turbine operation

2.6.1 Start-up

The requirement for gas turbine combined cycle plant start-ups has changed considerably in recent years. Combined cycle plants built in the 1990s were mostly designed as base load plants with typically 5 hot starts, 4 warm starts, 3 cold starts and 2 trips per year. In contrast modern plants more usually operate in two-shift mode for around 4,000 hours per year with typically 200 hot starts, 50 cold starts and 4 trips (Parsons Brinckerhoff, 2014).

In a combined cycle plant the ramp rate of the gas turbine is constrained by limitations imposed by equipment in the steam cycle. To protect that equipment, the gas turbine is traditionally ramped to a low load hold point, which lets the rest of the cycle warm up and achieve appropriate steam conditions before it is ramped further. At the hold point the gas turbine produces much higher CO emissions than at base load, which results in low power and high emissions during the hold. Gas turbines with a 3-pressure reheat combined cycle can experience two such holds prior to allowing the steam turbine to go to full load. Newly designed combined cycle plants are designed for faster start up with less need for hold periods. Fast plant operation can be enabled by use of a Benson once-through HRSG , which eliminates the thick walled drum and allows for unrestricted gas turbine ramping. HRSGs with thinner walled drums are an alternative choice, which offers much faster ramp rates than traditional plants but somewhat slower than the Benson design.

Another technique to reduce the start-up time of a combined cycle plant is to reduce the cooling of the HRSG when it is not in operation. This can be achieved by installing a stack damper to minimise cooling by natural convection. Some manufacturers also provide active measures to keep the steam generator warm between hot start-ups, introducing an auxiliary boiler that generates low pressure steam that is used to keep components warm.

The conventional hot start-up schedule for a combined cycle plant and the schedule in modern plants are illustrated qualitatively in Figure 11 (IEAGHG, 2012a). The improved start-up techniques approximately halve the start-up time.

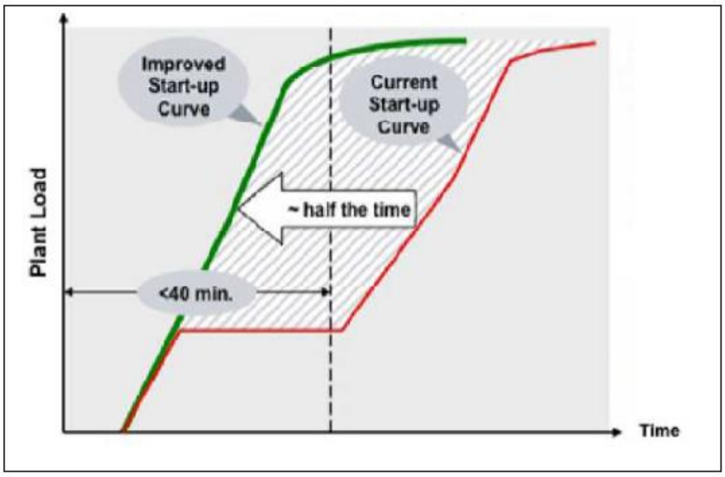


Figure 11 Improvement in start up schedules
Source: IEAGHG

Another technique that can be used is to operate the gas turbine in a combined cycle plant in simple cycle mode. This can be achieved by installing a by-pass stack that can divert the gas turbine exhaust gas directly to the atmosphere, rather than passing it through the HRSG. Downsides of this are the cost, maintenance requirements and leakage through the diverter valve during normal combined cycle operation.

Gas turbine manufacturers often offer the option of normal start-ups and "fast" start-ups. "Fast" start-ups involve more stresses on the turbine which increase maintenance costs. The plant operator can balance the increased maintenance costs against increased revenues depending on the prevailing electricity market prices.

Cold start-up times for combined cycle plants are longer than hot start-up times, typically by a factor of about three for existing plants (IEAGHG, 2012a, Parsons Brinckerhoff, 2014).

Aero-derivative turbines usually operate as simple cycle plants, except when they are part of combined heat and power schemes. Aero-derivative turbines usually have faster start-up times than frame gas turbines because of their lighter weight construction and less need to manage thermal expansion and stresses. Hot and cold start-up time classifications are not needed.

2.6.2 Part load operation

Power plants need to be able to operate at part load in order to match generation and power demand at all times. Gas turbines are turned down using two main techniques; reducing the mass flow and reducing the turbine inlet temperature. In some cases additional peak power can be generated by increasing the mass flow by steam or water injection, as discussed earlier. An increase in the turbine inlet temperature beyond the normal maximum continuous rating may be employed in some cases but this has a severe impact on turbine component lifetimes and maintenance costs.

The first technique that is normally applied to turn down a gas turbine is to reduce the mass flow, which is achieved by closing the compressor inlet guide vanes. Some turbines also employ variable stator vanes in the first few stages of the compressor to improve the ability to reduce the compressor mass flow rate. The inlet flow to the turbine is close to be being choked and as a simplification, $M.\sqrt{T}/P$ is a constant at full and part load, where M is the mass flow, T is the absolute temperature and P is the pressure. As a consequence, when the mass flow is reduced, the pressure at the inlet to the turbine reduces by approximately the same ratio. Because the turbine exhausts at an almost constant pressure, close to the atmospheric pressure, the pressure ratio of the turbine decreases, which further reduces the power output. If the turbine inlet temperature is kept constant, the lower pressure ratio results in an increase in the turbine exhaust temperature. If required, the turbine inlet temperature can be reduced in order to keep the turbine exhaust temperature constant. The impact of reduced mass flow and pressure ratio on the efficiency of a gas turbine is relatively modest. However, the ability to reduce the mass flow into the compressor is limited. Once the mass flow has been reduced by the maximum possible amount, the turbine inlet temperature has to be reduced, which tends to have a greater impact on the gas turbine efficiency. When the gas turbine is in a combined cycle plant, the lower turbine exhaust temperature also means that it is not possible to maintain the superheated steam temperature and the efficiency of the steam cycle also decreases.

The minimum load at which gas turbines can operate is usually dictated by environmental emissions. The critical emission is usually CO which increases greatly at low load due to lower firing temperatures and more incomplete combustion. Operation at low load factors during start-up should be minimised in order to avoid high emissions.

3. GAS TURBINE PERFORMANCE DATABASE

3.1 Description of the database and data sources

A database of information on commercial gas turbines has been created as an Excel file. Printouts of the database are included in the Appendix to this report.

This first sheet of the database includes a list of current modern 50Hz gas turbines with power outputs >5MW from major manufacturers. For each turbine the following information is provided for full load operation:

- Manufacturer
- Gas turbine model
- Type of turbine e.g. heavy frame or aero-derivative, reheat, water/steam injected
- Net power output
- Net efficiency
- Pressure ratio
- Exhaust mass flow
- Exhaust temperature

The second sheet includes a list of current 50Hz gas turbine combined cycle plants with power outputs >30MW from major manufacturers. For each plant the following information is provided for full load operation:

- Manufacturer
- Gas turbine model
- Net power outputs for 1 gas turbine + 1 steam turbine and 2GT+1ST plants
- Net efficiencies for 1GT + 1ST and 2GT + 1ST plants

The following data are also provided for selected simple and combined cycle plants, depending on the data availability:

- Start-up times; hot (normal and “fast”) and cold
- Minimum load (%)
- Ramp rate (MW and % per minute)
- Part load efficiency (50%)

In most cases the data have been obtained from manufacturers’ data sheets, compilations of gas turbine data and conference papers, which are listed in the references (Section 7 of the report). In most cases the information is from 2016 sources. Continuing improvements are being made to models of gas turbines so the performance data often change over time, which accounts for some of the discrepancies which sometimes occur between different sources. The basis and definitions are sometimes not well defined and they differ between sources, which also accounts for discrepancies. The information in this report and database is indicative only and no warranty is given that the information is complete or correct. Information should be obtained from manufacturers for any projects which have commercial implications. It should be noted that it has not been possible to supply data on every criteria for every turbine, due to limitations on the availability of dynamic performance data in the public domain.

3.2 Turbine Manufacturers

The market for gas turbines with power outputs greater than $5MW_e$ (as specified by ETI) is dominated by a small number of manufacturers and this is reflected in the database. The market for large frame power generation gas turbines is currently dominated by four manufacturers: GE, Siemens, Mitsubishi Hitachi Power Systems (MHPS) and Ansaldo. There have been significant company take-overs and transfers of assets in recent years, which have seen the disappearance of other manufacturers of large turbines. In particular, Westinghouse was taken over by Siemens and ABB was taken over by Alstom, whose overall power generation business was in turn taken over by GE. One of Alstom's large turbines then had to be divested to Ansaldo to comply with a regulatory requirement. Mitsubishi Heavy Industries and Hitachi merged their gas turbines businesses into Mitsubishi Hitachi Power Systems.

GE also supplies aero-derivative power generation turbines based on its own aero engines. The other two large aero engine manufacturers, Rolls-Royce and Pratt and Whitney, also used to supply aero-derivative power generation gas turbines but Rolls Royce sold its power turbines business to Siemens and Pratt and Whitney sold its power systems business to Mitsubishi Heavy Industries, who now supply aero-derivative turbines through a subsidiary called PW Power Systems.

In addition to aero-derivative and large ($>50MW_e$) frame gas turbines, GE, Siemens and MHPS also supply smaller turbines in the $5\text{-}50MW_e$ range. These turbines are usually aimed mainly at industrial CHP and mechanical drive applications but they are also sometimes used solely for electricity generation. Other companies also manufacture such turbines, in particular Kawasaki Heavy Industries, the Solar Turbines division of Caterpillar and MAN Diesel & Turbo.

As well as the primary turbine manufacturers there are also companies that supply gas turbine power generation packages based on turbines developed by the major manufacturers described above, including IHI, Bharat Heavy Electrical, Centrax and Dresser Rand (now part of Siemens). These suppliers have not been included in the database because their products are very similar to the products from the main manufacturers. Also not included in the database are some other turbine manufacturers that do not have a significant market presence in Western Europe, including Aviadvigatel, a Russian manufacturer of aero-derivative turbines, and the Iranian company Mapna.

3.3 Definitions of parameters in the database

3.3.1 Power output and efficiency

In line with the normal convention for gas turbines and combined cycle plants, data in the database are based on ISO conditions: 15C ambient temperature, 1.013 bar pressure (sea level) and 60% relative humidity. Also in line with the convention for gas turbines, efficiencies are on a lower heating value (LHV) basis, and the fuel is assumed to be pipeline quality natural gas.

Gas turbine performance varies substantially according to the ambient conditions. Manufacturers often publish graphs showing the variations in power output and efficiency due to differences in ambient pressure (due to elevation) and temperature but this is beyond the scope of this report.

Reducing the pressure and/or increasing the temperature reduces the mass flow into the compressor and hence reduces the throughput and power output of the turbine. An increase in the ambient temperature also increases the temperature throughout the compressor, which increases the compressor power consumption and reduces the thermal efficiency. Increasing the elevation reduces the ambient pressure. A 100m increase in elevation results in a reduction of typically about 1.1% in simple cycle power output but there is no change the efficiency. A 10C increase in temperature reduces power output by about 7% and reduces the efficiency by about 2.5% (not percentage points), but the impacts of ambient conditions are different for different turbines (GTW, 2016).

The power outputs and efficiencies of simple cycle gas turbines can be specified on a gross basis at the generator terminal or on a net basis including losses due to the pressure drop through the inlet air filter, inlet and exhaust ducts, stack and silencer and auxiliary loads. The difference between gross and net generally amounts to about a 1.8% reduction of power output and a 0.6% (not percentage point) reduction in the efficiency (GTW 2016). The efficiencies in the database are on a net basis.

Performance data published by manufacturers are for new turbines, representing data that would be obtained during acceptance tests before extended operation. Gas turbine performance declines over time due to fouling and wear and tear and power output and efficiency can decrease by around 2-3% compared to the new plant rating. Some of this degradation can be recovered by routine maintenance and washing. It is reported that degradation could reach 5% between overhauls and following the overhaul the performance can normally be restored to within 1-1.5% of the “new” rating (GTW 2016).

Combined cycle plant performance can be specified on a gross basis, or a net basis taking account of auxiliary consumptions. Net basis data are provided in the database. There is no industry standard set of parameters for combined cycles and the assumptions that have been used are not always specified by manufacturers, so there are inevitably some inconsistencies. The choice of cooling system (air or water cooling) and its design parameters can have significant effects on the steam turbine performance. The reported combined cycle performance data are for steam turbine condenser pressures of around 0.034-0.051 bar, which are reasonable for water cooled condensers and ISO conditions.

As with simple cycles, there are rules of thumb that can be used to quantify the effects of ambient conditions on combined cycle performance but it should be recognised that the magnitude of the effects differ between different turbines. A 10C increase in ambient temperature reduces the power output by about 4.5% and reduces the efficiency by about 0.9% (not percentage points). A 100m increase in elevation is reported to reduce the power output by about 1.2% and reduce the efficiency by less than 0.1% (GTW, 2016).

Combined cycle plants can consist of one steam turbine combined with one, two or more gas turbines. Power output and efficiency data for plants based on one and two gas turbines are included in the database. The efficiencies of plants with two gas turbines and one steam turbine are slightly higher due to the higher efficiencies of larger steam turbines. Plants based on two gas turbines are often able to operate at lower minimum loads by shutting down on of the gas turbines. It should however be noted that the same effect could be achieved by having two plants each based on one gas turbine.

3.3.2 Minimum load, start-up times and ramp rates

The minimum continuous operating load of gas turbines is normally set by environmental emissions, in particular by emissions of CO which increase substantially at low loads due to incomplete combustion.

Gas turbine start times are normally classified as hot, warm or cold. The definition of hot, warm and cold starts can differ between manufacturers. A hot start is generally defined as after a downtime of around 8 hours or less, e.g. after a night time shutdown. A warm start is after a shutdown of up to around 8-48 hours, e.g. a weekend, and a cold start is after a long term shutdown of greater than about 48-120 hours.

Hot start times are sometimes quoted as “conventional” and “fast” or “peaking”. The peaking rates result in increased maintenance costs but this may be worthwhile in circumstances where there is a strong need for rapid start-up. The start times quoted in the database are assumed to be conventional unless specified otherwise, although in some cases there is ambiguity, so the definition of the start times would need to be ascertained by contacting the manufacturers.

The ramp rates are the maximum average rate at which the plant output can be increased between the minimum load and full load.

3.4 Summary of information in the database

3.4.1 Power output and efficiency at full load

The relationship between power output and efficiency of simple cycle gas turbines (frame and aero-derivative) is shown in Figure 12.

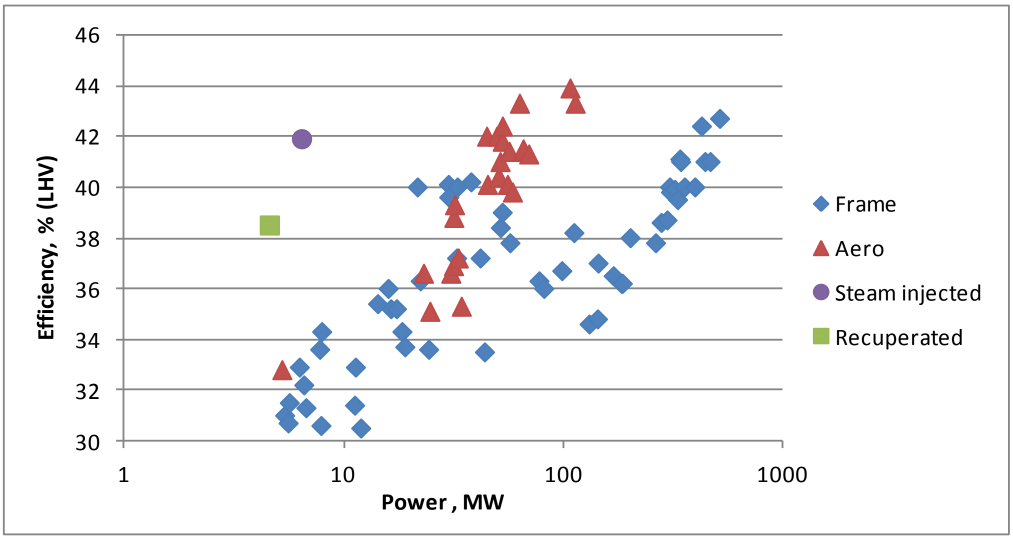


Figure 12 Power output and efficiency of simple cycle gas turbines

It can be seen that there is a trend towards higher efficiency at higher power output. This is partly because larger turbines tend on average to be more recent models with higher inlet temperatures. The higher efficiencies at higher sizes are therefore to some extent due to the age of the design of gas turbine rather than a direct function of size. This is particularly noticeable with turbines in the 150-400MW range. The turbines at the lower end of this range are mostly older E-class turbines, the ones in the middle are F-class turbines and the largest, highest efficiency turbines are H and J-class turbines. The same is also true for aero-derivative turbines, where the power output has increased over the years in line with the increase in the size of the commercial aero engines from which they are derived. However, even for the most modern designs, higher inlet temperatures tend to be applied in first in large size turbines rather than small and medium sized turbines and this contributes to the general trend to higher efficiencies at higher power outputs.

Aero-derivative turbines tend to have slightly higher simple cycle efficiencies than frame turbines of the same size, although this is not the case for smaller aero-derivatives which are based on older aero-engines. The higher efficiencies of aero-derivatives are mainly due to their higher pressure ratios.

The two out-lying data points in Figure 12 are a recuperated gas turbine (Solar Mercury 50) and a turbine with a large amount of steam injection (the Siemens 501-KH5), which have significantly higher efficiencies for their sizes. As discussed in section 2, these features increase the efficiencies of simple cycle gas turbines.

The relationship between power output and efficiencies of combined cycle plants is shown in Figure 13.

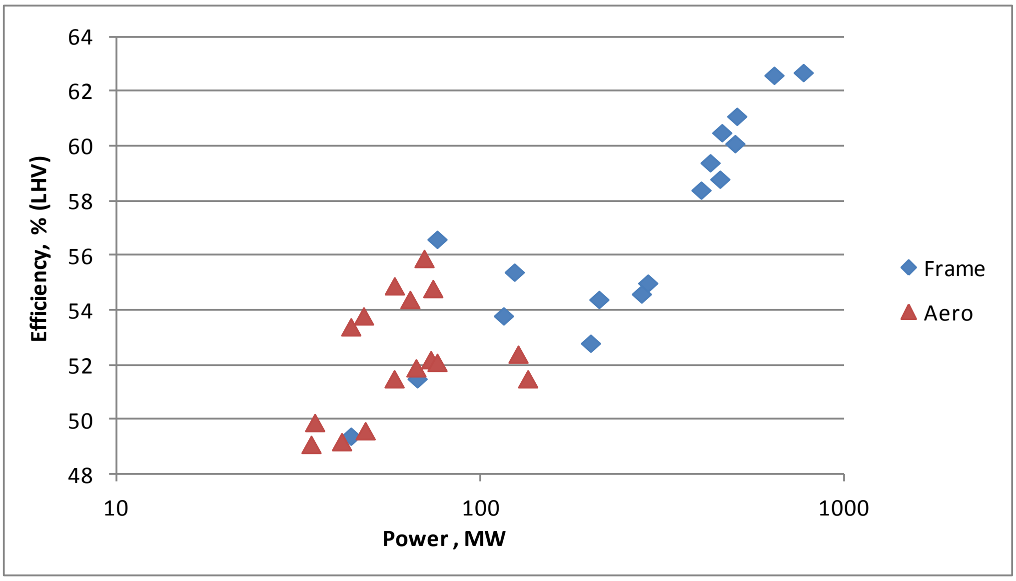


Figure 13 Power output and efficiency of combined cycle plants

The comments made earlier regarding higher efficiencies at higher power outputs also apply to combined cycle plants. The most significant difference compared to Figure 12 is that the efficiencies of aero-derivative combined cycle are on average broadly similar to or lower than those of similar sized plants based on frame gas turbines.

3.4.2 Efficiencies at part load

Indicative efficiencies of selected simple and combined cycle plants at 100% and 50% load are given in Table 1. Manufacturers do not usually publish part load efficiency data. The data in Table 1 are obtained from various published sources, often in graphical form, and it is subject to greater uncertainty than the full load data. In some cases the full load efficiency quoted in the part load data reference is slightly different to that in the database, mostly likely because of the on-going improvements that are being made to models of gas turbines. In such cases the part load efficiency has been scaled pro-rata to the full load efficiency for inclusion in Table 1.

Table 1 Indicative efficiencies of selected gas turbines at part load

	Full load power MW	Efficiency, % LHV basis		Ratio of efficiencies, 50%/100% load
		100% load	50% load	
Simple cycle				
Solar Mars 100	11.3	32.9	22.1	67
Kawasaki L20A	18.5	34.3	27.8	81
GE LMS100PA+	114	43.3	36.2	84
Combined cycle				
Siemens SGT-800	74	55.6	48.3	87
MHPS H100	143	53.8	46.0	86
Siemens SGT5-4000F	445	58.7	53.5	91
GE9F	462	60.5	53.6	89
Ansaldo GT26	502	60.1	55.9	93
Siemens SGT-8000H	600	60.5	55.5	92
MHPS 701J	680	61.7	55.0	89

The efficiencies of combined cycle plants operating at 50% load are around 90% of their full load efficiencies. The Ansaldo GT26 has the smallest decrease in efficiency at 50% load, due to its unique feature of reheat combustion. The efficiency reduction at part load is greater for simple cycle gas turbines, typically around 80% of the full load efficiency at 50% load. The relationship between load and efficiency of combined cycle plants is shown more generally in Figure 14, for two models of gas turbine. It can be seen that there is almost no reduction in efficiency at 90% load but the rate of reduction increases at lower loads.

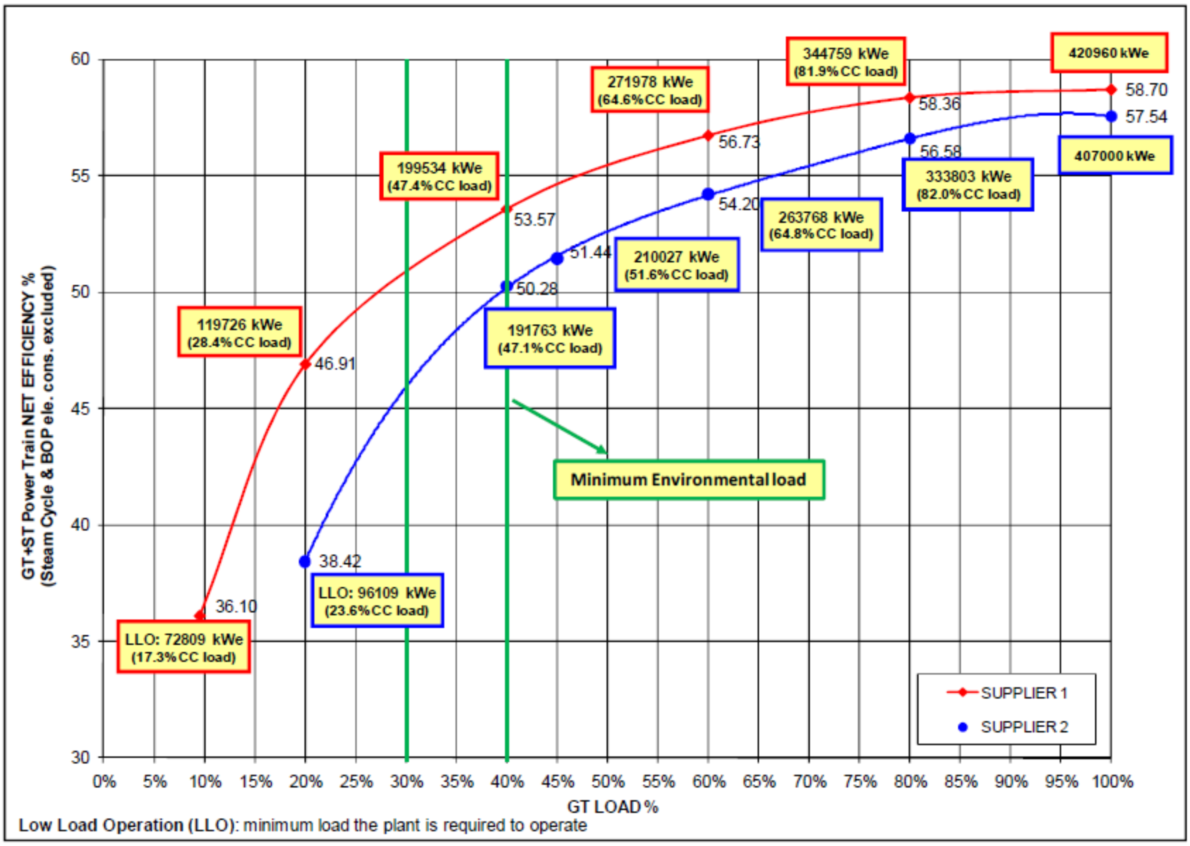


Figure 14 Overall combined cycle efficiency vs. gas turbine load
Source: IEAGHG

3.4.3 Start-up times

For the purposes of the database the start times provided by manufacturers are assumed to be normal hot start times unless stated otherwise. The hot start-up times of simple cycle turbines for which data are available are 12-30 minutes for frame gas turbines and 5-10 minutes for aero-derivatives. The hot start up times of combined cycle plants are around 30-70 minutes.

Although turbine manufacturers often quote hot start times they do not usually quote cold start times. Average cold start times of eight combined cycle plants in the UK based on five types of leading large gas turbine are around 215 minutes, which is 3.5 times their average hot start times (Parsons Brinckerhoff, 2014). Another reference gives typical cold start times of 250 minutes for 1990s base load plants and 180 minutes for recent flexible designs (IEAGHG, 2012a).

3.4.4 Ramp rates

Ramp rates of the simple cycle frame gas turbines in the database for which data are available are in the range of 5-33%/minute of rated output, with most being in the range of 7-15%/min. The rates for combined cycles are 7-45%/min and the typical value is around 10%/min. It is however emphasised that the ramp rates are highly turbine-specific, and manufacturers sometimes quote very different ramp rates for turbines that appear in most respects to be similar.

Aero-derivative turbines have substantially higher ramp rates, in the range of 87-120%/minute in simple cycle and 62-88%/minute in combined cycle. An exception is the GE LMS100, which has an aero-derivative core but which includes some features of heavy duty gas turbines, which has ramp rates of 44-46%/min in simple cycle and 37-39%/min in combined cycle.

3.4.5 Minimum load

The minimum loads of most of the gas turbines are in the range of 30-50% in simple cycle mode, although some are as high as 85%. The minimum loads of combined cycle plants are mostly in the range of 38-60% for frame gas turbines and 19-42% for aero-derivative turbines. An exception is the Ansaldo GT26 which has an exceptionally low minimum load of 10% in simple cycle mode and 15% in combined cycle mode.

4. COSTS

4.1 Capital costs

Capital costs of simple cycle and combined cycle plants are provided in this section of the report and in the database. The main source of data is the Gas Turbine World 2014-15 Handbook (GTW, 2015). The costs in the Gas Turbine World Handbooks are broadly in line with those quoted in the GTPRO turbine modelling software. In the case of some turbines, the power output ratings included in the database are higher than those pertaining at the time of the GTW 2015 reference. In those cases the cost per kW rather than the cost per machine has been assumed to remain constant.

The costs are indicative only and no significance should be attached to differences in costs between different manufacturers. Costs will in practice be determined on a project specific basis, depending on the scope of supply, local geographical factors, transport costs, tariffs, commercial arrangements, etc. The market price of gas turbines varies over time due to variations in supply and demand and other competitive market factors.

The main gas turbine manufacturers are based in the USA, continental Europe and Japan. The costs in this report are quoted in US$ rather than UK£ because most published data are in dollars and at the time of writing currency exchange rates are highly volatile. The main aim of this report is to provide a comparison of different classes of turbines, rather than absolute costs for a specific project at a UK location. It is recommended that the ETI consult an engineering and construction contractor to obtain costs for UK sites.

4.1.1 Simple cycle gas turbines

Prices of simple cycle gas turbines are quoted in this report as equipment prices at the factory gate for the turbine, generator and balance of plant such as air filter, exhaust duct, stack and control system. Total plant costs of simple cycle power plants are higher than the FOB equipment costs. Normal practice for building up the major equipment price into an estimate of the total project price is to apply a factor of 2 to increase the scope from equipment only to complete power island and a further factor of 1.1 to adjust from multi contract to an EPC contracting regime (Parsons Brinckerhoff, 2008).

Costs of simple cycle frame and aero-derivative gas turbines with power outputs greater than $5MW_e$ are shown in Figure 15 on a linear scale of power output and in Figure 16 on a logarithmic scale of power output.

Figure 15 shows that the specific costs ($/kW) decrease substantially at higher power outputs but above about 200MW the specific cost remains more constant. A larger power output would be expected to result in greater economies of scale but within the 200-470MW size range the larger turbines tend to be more modern machines with higher efficiencies. The higher efficiencies are achieved by using more exotic materials and improved cooling techniques which entail greater manufacturing complexity, which tends to offset the economies of scale. Turbine manufacturers also need to recover their substantial costs of developing new turbines though the prices of such turbines. It should also be borne in mind that the price of gas turbines is not necessarily directly related to the cost of development and manufacture. Turbines which have higher efficiencies have lower specific fuel costs, which should enable them to command a higher market price, provided other attributes such as flexibility, reliability and maintenance costs are the same.

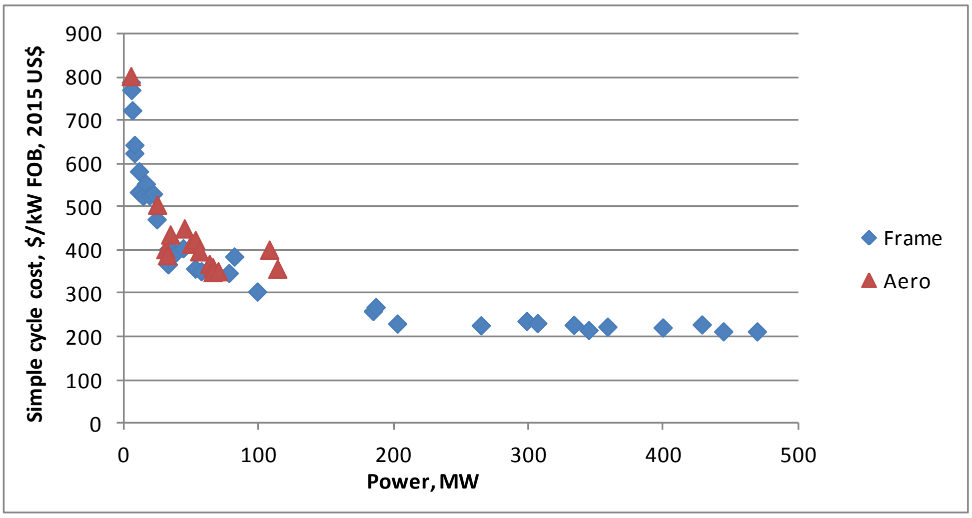


Figure 14 Simple cycle turbine equipment costs, FOB

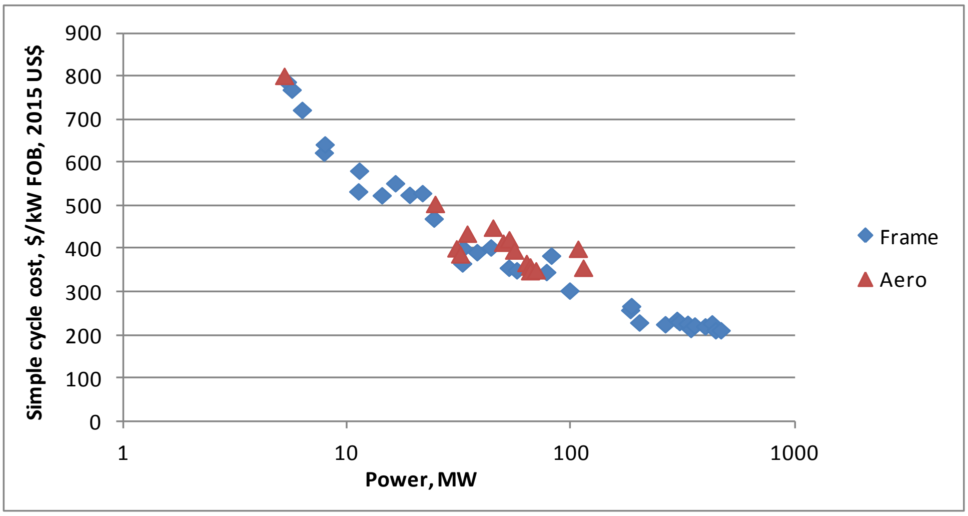


Figure 16 Simple cycle turbine equipment costs, FOB (logarithmic scale)

On a logarithmic basis, Figure 16 shows that the relationship between cost and power output is reasonably linear between about 10 and 470MW. The average cost scale exponent over this range is about 0.75, although as discussed above the specific cost is more constant and the cost exponent will be higher above 200MW.

Figures 15 and 16 distinguish between aero-derivative and non-aero-derivative turbines. Aero-derivative turbines appear on average to have slightly higher capital costs than other turbines with the same power outputs but it is not clear to what extent this difference is significant.

A study of peaking power plants carried out by Lummus Consultants for the New York Independent System Operator provides costs for gas turbine and gas engine plants with total power outputs of around 200MW at six locations within the New York area (Richert, 2016). The plant types are:

- 2 x GE LMS100PA+, aero-derivative gas turbines
- 1 x Siemens SGT6-5000F5, frame gas turbine
- 12 x Wartsila 18V50DF, gas engines

The capital costs of the aero-derivative turbine plants are about 50% higher than those of the frame turbine plants. The cost difference is in part due to the need to provide two aero-derivative turbines to provide the same power output as one frame turbine. The gas engine overall plant costs are 17% higher than the aero-derivative turbine plant costs. Amongst the six plant sites in the New York area, costs varied by as much as 30%, which emphasises the site specific nature of power plant costs.

A study carried out for the Western Electricity Council also provides costs of aero-derivative and frame gas turbines and gas engines from various sources (WECC, 2014). The ratio of costs is similar to in the New York study. The average aero-derivative turbine plant cost is 45% higher than the average frame turbine plant cost and the average gas engine plant cost is 8% higher than the aero-derivative turbine plant cost.

4.1.2 Combined cycle plants

Budget prices for total combined cycle plants including balance of plant and construction are shown in Figures 17 and 18. Costs are given for 1+1 plants, consisting of one gas turbine and HRSG plus one steam turbine, and 2+1 plants consisting of two gas turbines and HRSGs plus one steam turbine.

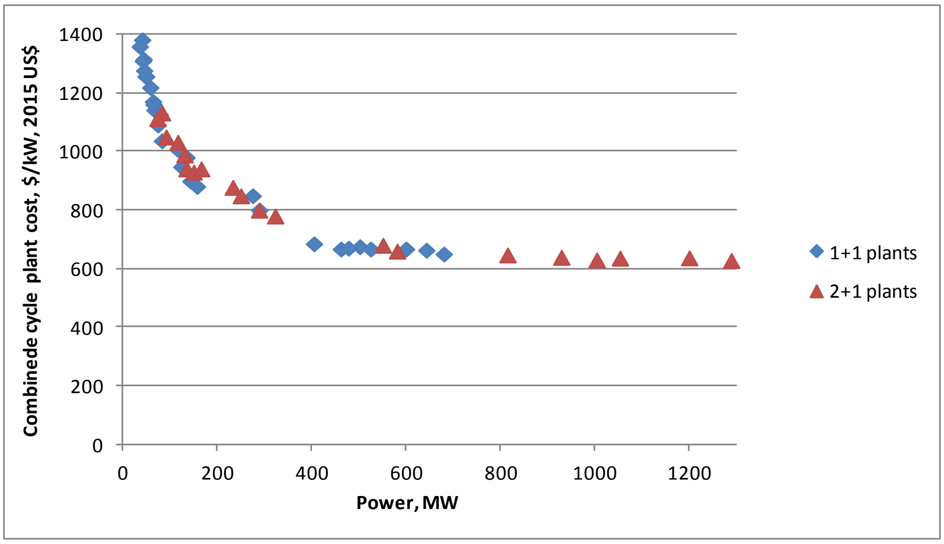


Figure 17 Combined cycle plant costs

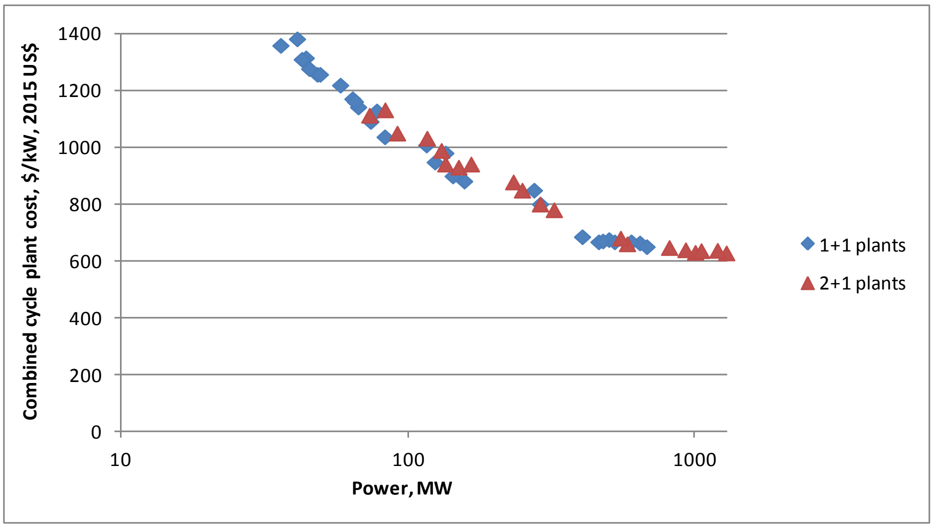


Figure 18 Combined cycle plant costs (logarithmic scale)

In common with simple cycle turbines, the costs per kW of combined cycle plants decrease with increasing size due to economies of scale but the rate of decrease in costs tails off at larger plant sizes. Costs for 1+1 and 2+1 plants of similar capacities do not seem to differ significantly.

The costs in Figures 17 and 18 are total plant "overnight" costs in current money values, excluding escalation and interest during construction. They include equipment supply, plant engineering and construction but exclude transportation, project-specific options, owner's costs, contingencies, commissioning and spare parts. A mechanical draught water cooling system is including but grid connection, fuel gas compression and back-up fuel supply (if required), an HRSG by-pass damper and stack and SCR and CO oxidation are not included. Including SCR and CO oxidation is reported to increase the cost by around 5% and including a by-pass damper and stack increases the cost by around 3% (El Masri, 2013). Use of an air cooling system is reported to increase the cost per kW by 10% because of a lower net power output and a higher capital cost (El Masri, 2013), although the relative costs depend on ambient conditions. Overall, it is reported that total plant costs for combined cycle plants can vary by as much as 30% depending on differences in engineering and design choices (GTW, 2015).

Costs also vary depending on the plant location. The costs presented here are for plants built in the US Gulf Coast region, which is a relatively low-cost region because of its relatively benign ambient conditions, regulatory requirements and a large pool of skilled labour and contractors.

In order to provide a general indication of the relative contributions of different plant areas to the overall cost of a combined cycle plant, Table 2 provides a breakdown of the cost of a plant based on two 60Hz, 210MW F-class gas turbines and one steam turbine (NETL, 2015). The costs are on a June 2011 basis. The Total Plant Cost is the Bare Erected Cost plus engineering and home office fees and contingencies. Start-up and inventory costs, escalation, interest during construction and owner's costs are not included.

Table 2 Breakdown of a combined cycle plant cost

Plant area	Equipment cost	Bare erected cost	Total plant cost	
	M$	M$	M$	$/kW
Fuel and feedwater inputs	28.1	42.6	53.9	86
Gas turbine	104.2	112.4	134.9	214
HRSG, ducting and stack	35.0	44.2	53.3	85
Steam turbine and auxiliaries	52.3	66.6	81.1	129
Cooling water system	5.0	15.9	19.9	32
Accessory electrical plant	21.3	37.8	45.6	72
Instrumentation and control	7.1	13.9	16.9	27
Site preparation and facilities	2.1	9.0	11.8	19
Buildings	0	10.8	13.5	21
Total	255.2	353.2	430.9	685

4.2 Operating costs

Examples of operating costs for base load plants are given in Table 3. Cost breakdowns are available in NETL (NETL, 2015) and IEAGHG (IEAGHG, 2012b) references but are not reported here. The IEAGHG costs are for a plant in Europe. The costs have been translated to US$ using the exchange rate at the time of the study, for comparison with the other studies.

Table 3 Power plant operating costs

	Plant type	Reference	$/year per kW net capacity		
			Fixed cost	Variable cost	Total
Combined cycle	7F, 2+1	NETL, 2015	25	12	37
Combined cycle	9F, 2+1	IEAGHG, 2012b	30	8	38
Aero-derivative simple cycle		WECC, 2014	15		
Frame simple cycle		WECC, 2014	9		
Combined cycle		WECC, 2014	10		
Gas engine		WECC, 2014	18		

Operating costs depend on contractual arrangements, for example sometimes the owners have long term service agreements, particularly for new turbines, and sometimes the maintenance is carried out on a more in-house basis. The main component of the IEAGHG variable cost is the cost of a long term service agreement linked to the number of operating hours. The main variable cost in the NETL study is the cost of water, which is highly site specific.

Operating costs of gas turbine power plants also depend on the number of start-ups and the amount of ramping. When a power plant is turned on and off, the components undergo thermal and pressure stresses, which cause damage which will result in higher plant equivalent forced outage rates (EFOR) and/or higher capital and maintenance costs to replace components at or near the end of their service lives. In addition, the overall plant life may be reduced. How soon these detrimental effects will occur will depend on the specific types and frequency of the cycling.

Median costs due to hot and cold start-ups and a load following excursion are shown in Table 4 (Intertek APTECH, 2012). The costs comprise increased capital and maintenance (C&M) costs and the increase in the equivalent forced outage rate (EFOR) and are in addition to the normal costs of continuous base load operation. Median costs of a cold start are around 1.5-3 times greater than the costs of a hot start. Aero-derivative turbines have lower start-up and cycling costs than large frame gas turbines, which is to be expected as they are based on aero-engines that are specifically designed for flexible operation. The relative costs of cold and hot starts are more similar for aero-derivative than for frame turbines. Table 4 shows median costs but costs vary substantially between different plants, as detailed in the reference. Modern power plants are designed for more flexible operation and it is possible that this may reduce cycling costs.

Table 4 Costs of gas turbine power plant cycling events

	Simple cycle		Combined cycle
	Aero-derivative	**Large frame**	
Hot start			
C&M cost, $/MW capacity	19	32	35
EFOR impact, %	0.0073	0.0020	0.0025
Cold start			
C&M cost, $/MW capacity	32	103	79
EFOR impact, %	0.0088	0.0035	0.0055
Load following			
C&M cost, $/MW capacity at typical ramp rate	0.63	1.59	0.64
Faster ramp rate cost multiplying factor	1-1.2	1.2-4	1.2-4

5. USE OF HYDROGEN-CONTAINING GASES IN GAS TURBINES

5.1 Hydrogen-containing gases used in gas turbines

The main gaseous fuel used in gas turbines is natural gas, which is predominantly CH_4 but various other “opportunity fuels” are also used, such as coke oven gas, refinery off-gas, blast furnace gas and syngas from coal and oil gasification. Typical compositions of these gases are shown in Table 5.

Table 5 Compositions of fuel gases used in gas turbines (Huth, 2012, OGJ, 2008)

	Composition, vol%				
	Natural gas	Coke oven gas	Refinery off-gas	Syngas (undiluted)	Blast furnace gas
H_2	-	50-60	5-35	25-50	2-6
CO	-	4-6	0-1	35-65	20-30
CH_4	80-100	20-30	30-50	0-6	-
C_2H_4	-	-	5-20	-	-
C_2H_6	0-15	1-3	15-25	-	-
C_3H_6	-	-	1-5	-	-
C_3H_8	0-5	0-1	1-5	-	-
C_4H_{10}	0-0.5	0-0.5	0-1	-	-
N_2	0-15	10-12	3-10	1-10	45-60
CO_2	0-6	1-2	0-1	2-20	20-25

Opportunity fuels are often available at lower costs than natural gas, which has provided an incentive for turbine manufacturers to make suitable machines available. The current low prices of natural gas have however resulted in reduced interest in the use of such fuels.

Gasification combined cycle plants with pre-combustion capture of CO_2 were expected to be a significant market for hydrogen-burning gas turbines in future. Such plants produce fuel gas with a hydrogen concentration of around 90% before dilution (assuming 90% CO_2 capture) (NETL, 2015). The level of interest in such plants has however failed to live up to expectations, and the focus of interest in capture of CO_2 has shifted towards post combustion capture rather than pre-combustion capture. This has further reduced the commercial impetus towards development of hydrogen-burning turbines. The money needed to develop and demonstrate hydrogen-burning turbines is substantial, for example Solar Turbines mentions a cost of $50M to develop a fuel delivery package, control and safety systems for a 15MW turbine (Solar Turbines, 2013).

The opportunity fuel that is most directly relevant to the ETI's proposed scheme of using stored hydrogen and methane in gas turbines is coke oven gas, which consists of 50-60% H_2 together with 20-30% CH_4. The composition of refinery off-gas is variable, the H_2 concentration is typically 5-35% but it can be as high as 95%. Syngas from gasification (without CCS) is also relevant even through it contains only small amounts of CH_4. In some respects the large concentration of CO creates additional difficulties but in other respects it reduces problems compared to pure H_2, as discussed later, as discussed later.

5.2 Properties of H_2, CH_4 and CO

Properties of H_2 are shown in Table 6, along with properties of CH_4 and CO, which is a major constituent of some other fuel gases used in gas turbines.

Table 6 Properties of gases (Huth, 2013; Smith, 2009)

	CH_4	H_2	CO
LHV, MJ/Nm^3	35.91	10.78	12.63
LHV, MJ/kg	50.06	119.97	10.10
Density, kg/Nm^3	0.72	0.09	1.25
Stoichiometric combustion temperature, K	2206	2376	2370
Stoichiometric air demand, kg/kg	17.35	34.53	2.49
Laminar flame speed (max), cm/s	54	770	2.7
Laminar flame speed (excess air ratio=2), cm/s	12	43	0.8
Chemical reaction time (min), s	2E-5	2E-7	6E-3
Chemical reaction time (excess air ratio=2), s	4E-4	4E-5	8E-2
Autoignition time at 1000C (min), s	1E-3	2E-5	4E-2
Flammability limits, vol%	5-15	4-75	12.5-74
Wobbe index, MJ/m^3	48.21	40.9	12.85

Properties at 18 bar, fuel-air mixtures calculated with 420C air temperature and 200C fuel temperature

Compared to CH_4, H_2 has a lower density, a higher stoichiometric combustion temperate and flame speed, shorter autoignition and reaction times and a wider flammability limit, especially a higher upper flammability limit. The heating value of H_2 is lower on a volumetric basis but higher on a mass basis.

The Wobbe index is a common indicator of fuel characteristics and inter-changeability in combustion systems and gas turbines. Wobbe index is defined as the heating value divided by the square root of the specific gravity of the fuel (the density of the fuel divided by the density of air). The pressure drop in the fuel system will be the same for different fuels with the same Wobbe index and in general direct substitution is possible and no changes have to be made to the fuel system. Table 6 shows that H_2 and CH_4 have broadly similar Wobbe indices. The lower density of hydrogen is counteracted by its lower volumetric heat of combustion. In contrast, CO has a much lower Wobbe index than CH_4 because although it also has a lower volumetric heat of combustion it has a higher density. Vendors sometimes specify the Wobbe index tolerances of their gas turbines. Some examples are given in Table 7. For comparison, the Wobbe index of H_2 is 18% higher than that of CH_4.

Table 7 Wobbe Index variation of gas turbines (GE, 2016a)

Gas turbine model	Wobbe index variation
GE6B.03	>±30%
GE9E.03/.04	>±30%
GE9F.06	±15%
GE9HA	±15%
GE LM6000PF	±25%
GE LMS100PA+	±20%
GE LMS100PB+	±25%

There are however limitations of the use of Wobbe Index when burning fuels with very different compositions. The Wobbe index does not reflect changes in other fuel properties such as flame speed and combustion chemistry. If more reactive species such as H_2 are present in significant quantities, additional changes to the fuel system may be required.

5.3 Impacts of hydrogen on combustor design, operation and emissions

Important issues that are affected by the use of hydrogen-containing gases in gas turbine combustors include flashback, blowout, autoignition, stoichiometric flame temperature and emissions, flammability limits and combustion instability.

5.3.1 Flashback and blowout

Blowout, also known as static stability is when the flame becomes detached from where it is anchored and is blown out of the combustor. The opposite of blowout is flashback, which is when the turbulent flame speed is greater than the flow velocity and the flame propagates backwards into the premixing section of the combustor. Flashback often occurs in the flow boundary layer, since this is the point of lowest flow velocity. Hydrogen has a high flame speed, which means that flashback is more significant when burning hydrogen and blowout is less significant.

Most issues are related to the turbulent flame speed, which depends on the laminar flame speed, the level of turbulence in the combustor and other factors such as diffusion characteristics of the chemical components. Two different fuels having the same laminar flame speed, turbulence intensity and burner can have appreciably different turbulent flame speeds. In swirling flows flashback can potentially occur by other mechanisms such as vortex breakdown. In this case flashback can occur even if the turbulent flame speed is less than the flow velocity. (Lieuwen, 2008).

The laminar flame speed is defined as the velocity at which unburned gases move through the combustion wave in the direction normal to the wave surface. The laminar flame speed does not vary linearly between the respective pure values of the mixture constituents. The addition of H_2 to CH_4 does not have a substantial impact on the laminar flame speed until H_2 is the dominant constituent of the mixture, as shown in Figure 19 (Lieuwen, 2008).

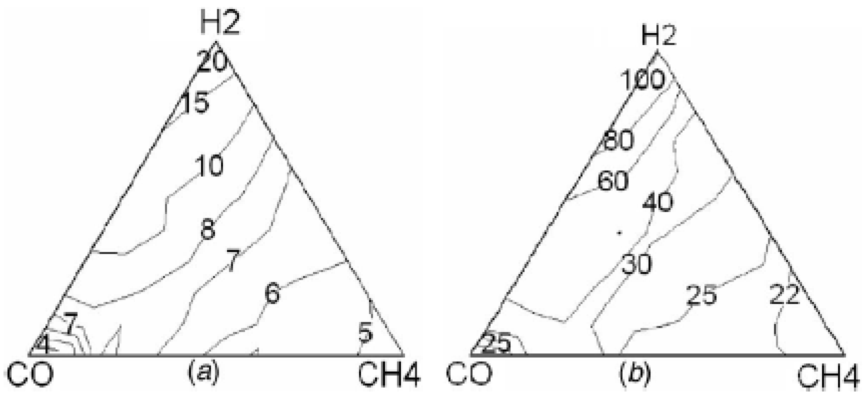


Figure 19 Dependence of laminar flame speed (cm/s) upon composition

1500K (left), 1900K (right) adiabatic flame temperatures at 4.4 atm with 460K reactant temperatures

Source: Lieuwen, 2008

The EU has funded a research project, H2-IGCC, that involves the development of syngas and hydrogen fired gas turbines (H2-IGCC, 2014). The potential for flashback was identified as the largest design challenge when using high hydrogen fuels, which can be overcome to a certain extent by increasing the bulk velocity of the reactants through the burner, but this results in a configuration that is hard to stabilise when operating with natural gas. If the final burner design must be fuel flexible (operation with H_2-rich fuel gas as well as with natural gas) then design conflicts are apparent at this stage.

5.3.2 Autoignition

Autoignition involves spontaneous combustion in the absence of a concentrated source of ignition, for example a spark or flame. It is similar in some respects to flashback in that it results in combustion upstream in the premix section of a combustor but it has different physical origins. The autoignition time delay is the time required for a mixture to spontaneously ignite at specific conditions. It can be seen from Table 6 that the autoignition time of H_2 is much less than that of CH_4. Higher temperatures and pressures tend to shorten the autoignition time and leaner mixtures tend to have a longer delay time (Kurz, 2012). Autoignition in the premix section can damage the combustor and cause high emission levels. In a lean premix combustor the flow velocities have to be high enough to avoid autoignition inside the injector. While the presence of hydrogen greatly reduces the ignition time delay at high temperatures compared to methane, at typical gas turbine combustor inlet temperatures the autoignition time delay should be sufficiently long to preclude autoignition in a well designed premixer (Lieuwen, 2012). However, few data are reported to be available for mixtures and engine specific tests are often necessary to avoid problems (Kurz, 2012).

5.3.3 Flame temperature and emissions

In standard combustion systems the flame temperature is close to the stoichiometric flame temperature (Kurz, 2012). The stoichiometric flame temperature strongly affects the amount of NO_x that is produced, with higher temperature resulting in more NO_x. As shown in Table 6, H_2 has a higher stoichiometric flame temperature than CH_4, so NO_x production is more of an issue. As discussed in section 2.5.2, the stoichiometric flame temperature can be reduced by dilution with steam, water or N_2, thereby reducing NO_x production.

Although the flame temperature of H_2 is higher than that of CH_4 at the same equivalence ratio, the presence of H_2 extends the lean blowout limit. If the reduction in lean blowout limit is significant then the combustor could be operated with a higher equivalence ratio, which could offset the increase in flame temperature and have an overall positive effect on NO_x emissions. However, modifying the fuel and/or flow rates may change the mixing profile and temperature distribution, potentially leading to an in increase in NO_x emissions (Taamallah, 2015). At lean combustion conditions the impact of hydrogen on NO_x formation does not have a clear trend and the NO_x emission is not believed to change significantly as hydrogen is mixed into the natural gas up to 35 % by volume H_2 (Andersson, 2013).

Another issue that may arise if the fuel is a mixture of H_2 and CH_4 rather than pure H_2 is that emissions of CO may be higher because CO oxidation to CO_2 is more limited at leaner conditions and lower temperatures.

5.3.4 Flammability limits

The fuel air ratio in a gas turbine changes at different loads. In order to avoid flameout, i.e. when the flame in the combustor is extinguished, it must be possible to achieve combustion over a range of fuel-air ratios. The ratio of flammability limits indicates whether it will be possible to operate the combustor at the required range of operating points of the turbine. The upper and lower flammability limits are the maximum and minimum percentages of fuel in a fuel-air mixture that can sustain combustion. It can be seen from the data in Table 6 that hydrogen has a much larger ratio of flammability limits (the upper flammability limit divided by the lower). This indicates that flameout should be less of a concern for H_2 fired turbines.

5.3.5 Combustion instability

Combustion instabilities are characterised by large amplitude pressure oscillations that are driven by unsteady heat release. Combustion instabilities can cause increased sound levels and physical damage to the combustor and in extreme cases they can cause the break-off of material that damages downstream components in the turbine. For instabilities to occur the heat and pressure oscillations must be in phase (or more precisely, their phase difference is less than 90 degrees). The degree of instability is affected by the fuel composition. Of particular significance in premixed combustor systems are two mechanisms: fuel/air ratio oscillations and vortex shedding. In the former mechanism, acoustic oscillations in the pre-mixer section cause fluctuations in the fuel and/or air supply rates, thus producing a reactive mixture whose equivalence ratio varies periodically in time. The resulting mixture fluctuations are convected in the flame where it produces heat release oscillations that drive the instability. Vortex shedding is the result of flow separation from the flameholders and rapid expansions as well as vortex breakdown in swirling flows. They are convected by the flow of the flame where they distort the flame front and thereby cause the rate of heat release to oscillate (Lieuwen, 2008).

Lean premixed combustors are more prone to combustion instabilities than earlier types of gas turbine combustor (Taamallah, 2015). Fuel composition can affect combustion instabilities. The results from the literature point towards the conclusion that fuel change is not an additional complication for premixed combustion dynamic stability. Some conflicting results can be noticed, most probably due to the geometry and specific design of the combustors used. This strengthens the idea that addition of H_2 can have a positive or negative impact on combustor dynamic stability (Taamallah, 2015).

5.4 Impacts on overall gas turbine design and operation

5.4.1 Compressor-turbine matching

Use of nitrogen, steam or water as diluents for NO_x control increases the mass flow rate into the expansion turbine. The swallowing capacity of the turbine (M√T/P) is almost constant so the higher mass flow rate results in a higher turbine inlet pressure, and hence a higher compressor outlet pressure. Increasing the compressor outlet pressure moves it closer towards the surge line, beyond which flow instability and catastrophic damage can occur. To avoid the compressor operating too close to or reaching the surge line, various techniques can be used. The extent to which these techniques are needed depends on the surge margin of the compressor, i.e. how far away from the surge line it operates under normal full load conditions. The surge margin is different for different gas turbines. The simplest technique is to reduce the mass flow of air into the compressor by closing the compressor inlet guide vanes. Closing the inlet guide vanes is however a technique that is normally used to turn down a gas turbine and it is also used to accommodate changes in ambient temperature. If the gas turbine has to operate at full load with the inlet guide vanes partially closed, the ability to efficiently turn down the turbine is reduced. The advantage of closing the inlet guide vanes is that it does not require significant change to the turbine design.

An alternative technique that can be used is to add an extra stage or stages to the compressor, which enables the compressor to operate with a higher pressure ratio without moving closer to the surge line but this also involves some re-engineering. An example of this the V94.2K gas turbine developed by Siemens for IGCC and blast furnace gas. This turbine was created by modifying the existing standard V94.2 turbine by removing the first compressor stage and by adding two additional final compressor stages (Smith, 2009). This turbine is used at the ISAB oil residue IGCC plant in Italy.

Another technique that can be used is to increase the height or angle of the turbine blades, which enables the turbine to swallow a greater mass flow without increasing the pressure, but this requires a significant re-engineering of the turbine.

A further technique that is used in some IGCC plants is to extract some air from after the gas turbine compressor, which reduces the mass flow into the turbine to offset the higher mass flow of fuel gas and diluents. The extracted air is fed via a partial pressure let-down turbine to a cryogenic air separation plant, thereby replacing some of the air that would otherwise be provided by the ASU main air compressors. This technique was applied in some commercial IGCC plants, for example at Buggenum, but a high degree of integration between the ASU and gas turbine was found to result in greater operational difficulties and lower operating flexibility. Air extraction is not suitable if the ASU and gasification plant are not integrated and do not necessarily operate at the same times, which would be the case in ETI's schemes with hydrogen storage.

5.4.2 Turbine heat transfer

Use of hydrogen fuel and the addition of diluents change the gas composition and increase the pressure, which affect the turbine heat transfer.

When H_2 is used as fuel instead of CH_4 the quantity of H_2O in the turbine inlet gas increases and the quantity of CO_2 decreases. This however has no significant impact on the heat flux on the outer surface of the turbine blades (Chiesa, 2005). In contrast, steam dilution increases the thermal flux because the heat transfer coefficient of steam is higher than that or air. The higher heat transfer coefficient would result in higher blade metal temperatures which would be unacceptable as it would reduce the lifetime of the turbine. The temperature profile can in principle be restored by increasing the cooling flow but this would require re-engineering of the turbine. The only feasible alternative is to reduce the temperature of the turbine inlet gas, but this reduces the efficiency of the turbine.

The higher mass flow rate due to the addition of diluents can increase the pressure of the turbine inlet gas, as described earlier. A higher pressure affects the turbine heat transfer in three ways. Firstly, the heat transfer coefficients both inside and outside the blades increase but this is not a neutral effect because the increased heat flux reduces the temperature difference between the fluid and the blade, which would result in a higher blade temperature (unless the temperature of the expansion gas is reduced). Secondly, the temperature of the cooling air from the compressor increases, which reduces the effectiveness of cooling. Thirdly, while the geometry of the cooling air circuit remains the same, the mass flow of cooling air increases because the density of the compressed air increases. This increased cooling air flow rate increases the extent of blade cooling which tends to offset the two other effects of the higher pressure. However, overall the turbine inlet temperature needs to be reduced to avoid an increase in the blade metal temperatures.

5.4.3 Turbine materials

The concentration of H_2O in the turbine inlet gas is higher when firing hydrogen, especially when H_2O is used as a diluent in the combustor. The presence of H_2O speeds up the oxidation mechanism of turbine thermal barrier coatings, shortening the life of the coatings. It also appears that the higher thermal gradients through thermal barrier coatings in hydrogen fired turbines accelerate some modes of cracking and degradation of coatings. It may be necessary to decrease the turbine inlet temperature with respect to natural gas firing in order to preserve the turbine lifetime (Grazzani, 2014). However, combusted hydrogen-enriched syngas did not cause significantly more damage than combusted natural gas in demonstrations in either ENEL's Fusina gas turbine or Cranfield University's burner rig cascade trial (H2-IGCC, 2014).

5.4.4 Hazards

Hazards which need to be considered include flammability and detonation limits and auto-ignition temperatures outside as well as inside the turbine and its associated heat exchangers and ducting need to be considered. Gas turbine enclosures are designed to avoid accumulation of leaked gases and gas detectors are installed. The concentration of leaked gas must not exceed the flammability limits. Hydrogen has an especially low density, which can cause problems in an enclosure because it will rise and may accumulate in high dead spots, although outside it tends to aid its dispersion. Hydrogen has a non-luminous flame which makes detection more difficult. Hydrogen has a negative Joule Thompson coefficient at temperatures at or above ambient temperatures, which results in a temperature increase when it expands but this is unlikely to cause ignition on its own because the increase is too low, e.g. 9-18K for expansion from 50MPa at 9C. A stoichiometric mixture of hydrogen and air has a very low minimum ignition energy, which makes it far more sensitive to ignition than other gases or vaporised flammable materials, and the potential for electrostatic ignition is much greater (Gummer, 2008).

5.5 Performance of hydrogen fired gas turbines

In diffusion flame combustors the flame tends to be close to the stoichiometric flame temperature. The temperature needs to be reduced by dilution to reduce NO_x emissions to acceptable levels. In lean premix combustors the flame temperature is limited by the large excess of air and no diluents need to be added, but hydrogen has a high flame speed which requires high air velocities to obtain short mixing times and high turbulence rates. This may result in high pressure drops.

The sensitivity of performance of gas turbine combined cycle plants to stoichiometric flame temperatures and diluents rates in diffusion combustors, and the combustor pressure drop in a premix combustor has been modelled (Grazzani, 2014). The results are shown in Table 8. The features of the gas turbine are representative of a state of the art large 50Hz F-class frame gas turbine specifically designed for hydrogen combustion.

Table 8 Effects of combustor type, combustor pressure drop and diluent addition on the performance and emissions of hydrogen-fired turbines

		Premix combustor		Diffusive flame combustor			
		No diluent		Nitrogen diluent		Steam diluent	
Stoichiometric Flame Temperature	K	2712	2712	2575	2200	2575	2200
Combustor pressure drop	%	3	10	3	3	3	3
Diluent:hydrogen ratio	kg/kg	0	0	3.46	15.93	1.62	7.56
NO_x (15% O_2 basis)	ppmv	-	-	250	19	250	19
Turbine flows and temperatures							
Compressor air inlet flow	kg/s	662	619	640	560	647	591
Hydrogen fuel flow	kg/s	6.0	5.6	6.0	6.1	6.1	6.4
Diluent flow	kg/s	0	0	21	98	10	48
Gas turbine inlet flow	kg/s	539	503	538	536	532	505
Gas turbine outlet temperature	C	575	586	575	574	578	590
Steam turbine inlet flow	kg/s	72	70	72	72	73	78
Energy flows and efficiencies							
Fuel input	MW (LHV)	724	676	726	735	732	764
Gas turbine gross power	MW	289	262	298	330	296	325
Nitrogen compressor power	MW	0	0	10	48	0	0
Steam turbine gross power	MW	138	133	138	137.3	131	104
Combined cycle net power	MW	423	391	421	415.1	424	426
Combined cycle efficiency	% (LHV)	58.47	57.90	58.04	56.49	57.91	55.76
Efficiency vs pre-mix combustor base case	% points	-	-0.57	-0.43	-1.98	-0.56	-2.71

The mass of diluent has a large impact on the quantity of NO_x produced. In order to achieve NO_x of less than 20ppmv (15% O_2 basis) the stoichiometric flame temperature needs to be reduced to about 2200K, which requires a N_2:H_2 ratio of about 16:1 or a steam:H_2 ratio of about 7.5:1. This corresponds to a concentration of about 53%vol N_2 in a N_2/H_2 mixture and 46%vol steam in a steam/H_2 mixture. Using N_2 diluent to achieve a stoichiometric flame temperature of 2200K reduces the efficiency of a hydrogen fired combined cycle plant to 56.5%, i.e. a 2.0 percentage point reduction compared to a plant with no diluent addition. Using steam as the diluent reduces the efficiency to 55.8%, i.e. 2.7 percentage point reduction. The steam for injection into the gas turbine is assumed to be extracted from the steam turbine of the combined cycle plant and nitrogen is assumed to be compressed from atmospheric pressure. The capital cost of a nitrogen compressor on the one hand and the costs of water treatment for steam injection are additional costs that would need to be taken into account in any economic comparison of steam and nitrogen injection. In the type of hydrogen storage schemes proposed by ETI, nitrogen would have to be extracted from underground storage, which may impose additional energy penalties as well as extra capital costs.

Grazzani's analysis also considered the effects of a conservative approach of reducing the nominal blade metal temperature, in case this is necessary to compensate for faster degradation of thermal barrier coatings, as mentioned earlier, and a more uneven distribution of temperature at the entry to the turbine. As an example, reducing the blade metal temperature by 40C reduced the combined cycle plant efficiencies by an average of around 1.3 percentage points.

Grazzani, 2014 only assesses combined cycle plants. The relative efficiencies of nitrogen and steam injection would be different for simple cycle plants. In a simple cycle plant with nitrogen addition the turbine exhaust gas would be exhausted straight to atmosphere but for steam injection the exhaust gas could be passed through a simple heat recovery steam generator to generate steam which would be fed directly to the gas turbine. Assuming the generated steam was at the same temperature as the steam extracted from the steam turbine in Grazzani's analysis, the efficiency of a simple cycle plant with sufficient steam addition to achieve a stoichiometric flame temperate of 2200K (<20ppmv NO_x) would be about 42.5%, which is significantly higher than the 38.3% efficiency of a corresponding plant with nitrogen addition. A downside of using steam addition in a simple cycle plant is that the addition of a steam generator would increase the start-up time, which may be a disadvantage for a peak load plant.

The analysis described above is based on turbines specifically designed for hydrogen rich fuel gas. The market for turbines burning hydrogen rich gas is currently small, so at least in the short term, existing turbines would have to be used as far as possible and the their operating parameters would have to be modified to accommodate the use of hydrogen rich gases. As discussed earlier, hydrogen fired gas turbines with diluent injection would have to operate either by partially closing the compressor guide vanes to reduce mass flow through the compressor, or by keeping the air flow constant and allowing an increase in turbine inlet pressure. Alternatively, the turbine could be re-engineered to increase the first stage turbine nozzle area, to enable the turbine to accept a higher mass flow without requiring a higher pressure. In practice a combination of these techniques could be used. The effects of these different techniques on the performance of a combined cycle plant were assessed based on a set of assumptions corresponding to a Siemens V94.3A large frame F-class gas turbine, which is the predecessor of the current Ansaldo AE94.3A and the Siemens SGT54000F (Chiesa, 2005). The results are shown in Table 9.

Table 9 Effects of fuel type, diluents and turbine design and operation on the performance of gas turbines

Fuel		**Natural gas**	**Hydrogen**								
Turbine design/operating mode		**Standard**	**Variable guide vanes**			**Increased pressure ratio**			**Re-engineered turbine**		
Diluent		**None**	**None**	**Steam**	**Nitrogen**	**None**	**Steam**	**Nitrogen**	**None**	**Steam**	**Nitrogen**
Turbine pressure & temperatures											
Stoichiometric Flame Temperature	K	2545	2745	2300	2300	2746	2300	2300	2745	2300	2300
Pressure ratio		17.0	17.0	17.0	17.0	17.0	18.5	19.7	17.0	17.0	17.0
Turbine inlet temperature	C	1350	1339	1316	1340	1339	1305	1319	1350	1350	1350
Turbine outlet temperature	C	585	575	577	574	574	563	549	584	591	569
Turbine flow rates											
Compressor air inlet flow	kg/s	634	632	584	551	634	634	634	634	634	634
Fuel flow	kg/s	15.0	5.6	5.7	5.5	5.6	6.0	6.1	5.7	6.3	6.3
Diluent flow	kg/s	0	0	38	80	0	42	94	0	43	91
Diluent:hydrogen ratio	kg/s	0	0	6.8	14.4	0	6.8	15.4	0	6.8	14.5
Energy flows and efficiencies											
Gas turbine gross power	MW	257	264	292	298	265	314	341	266	324	343
Nitrogen compressor power	MW	0	0	0	43	0	0	54	0	0	49
Steam turbine gross power	MW	130	126	91	125	126	92	132	130	105	142
Combined cycle gross power	MW	387	390	383	380	391	406	419	396	429	436
Combined cycle gross efficiency	%, LHV	57.57	58.32	56.38	57.46	58.32	56.25	57.15	58.35	56.60	57.57
Efficiency vs natural gas	% point	-	+0.75	-1.19	-0.11	+0.75	-1.32	-0.42	+0.78	-0.97	0.0

The main conclusions of the analysis are summarised below:

- If no diluent is used, using hydrogen as the fuel rather than natural gas increases the efficiency of a combined cycle plant by 0.75 percentage points and increases the efficiency of a simple cycle gas turbine by 1.36 percentage points, due to the different thermodynamic properties of the fluids.
- Using nitrogen or steam as a diluent in the combustor reduces the efficiency of a combined cycle plant. Compared to a natural gas fired combined cycle plant a hydrogen fired plant with steam injection has an efficiency that is about 1-1.3 percentage points lower and a plant with nitrogen injection has an efficiency that is about 0-0.4 percentage points lower.
- If the compressor air flow is kept constant, adding a diluent increases the pressure ratio of the turbine from 17:1 to 18.5:1 in the case of steam and 19.7:1 in the case of nitrogen, in order to accommodate the larger turbine inlet gas flow rate with the same nozzle area. Given the stall margins available on existing machines it is doubtful whether this highest pressure ratio could be achieved without any modification to the machine and probably one or more high pressure compressor stages would need to be added to shift the surge limit upwards (Chiesa, 2005).
- The highest efficiencies are achieved if the turbine is re-engineered to increase the nozzle area of the turbine to enable it to accept a higher mass flow without an increase in pressure.
- Using hydrogen as the fuel results in a lower gas turbine exhaust temperature. This reduces the efficiency and power output of the steam cycle in a combined cycle plant. The temperature reduction is greatest in the case where the compressor air flow is kept constant, because of the additional effect of the higher pressure ratio across the gas turbine.
- Using hydrogen as the fuel increases the gas turbine power output. The increase is greater when diluents are used; it is as much as a third higher when nitrogen is used and the turbine pressure ratio is allowed to increase. Although a higher power output is generally advantageous it may exceed the mechanical limits of the turbine. Also, a larger generator would be needed.

The optimum choice of diluent would depend on many factors, in particular the duty cycle of the gas turbines (e.g. peak load with frequent start-ups or mid-merit with less variable operation), and whether there is a constraint on water consumption. The need to generate steam in a heat recovery steam generator would increase the start-up time and possibly increase emissions during start-up, which would be a disadvantage for peak load plants. In this case the greater flexibility of nitrogen supply would be an advantage.

An alternative to steam injection would be to add water vapour to the hydrogen fuel in a saturator. Heat would be provided to the saturator using a recirculating stream of warm water, which could be heated in the lower temperature region of the HRSG. This would be thermodynamically more efficient than generating steam and adding this to the hydrogen because the evaporation of water would be achieved using lower grade heat. This is the concept of the HAT cycle described in section 2.4.5.

Another alternative would be to use exhaust gas recycle (EGR), in which a fraction of the turbine exhaust gas is used instead of nitrogen or steam to reduce the flame temperature and NO_x emissions. It is reported that diffusion combustors could be used, but at high enough EGR rates (i.e., a working fluid with a very depleted O_2 level) the use of lean premixed burners becomes also feasible (Ditaranto, 2015).

5.6 Manufacturers' experience with using hydrogen-containing fuel gases

Some commercial gas turbines already operate on fuel gases that contain H_2 or they have been developed to do so. There are heavy duty gas turbines equipped with diffusive flame combustors operating with gaseous fuels containing up to 95%vol hydrogen (Cocchi, 2008). Reduction of NO_x emissions down to acceptable levels is generally achieved by means of steam, water or nitrogen injection.

Major turbine manufacturers' experience of using H_2-containing fuel gases is summarised below.

5.6.1 GE

GE has published a list of the capabilities of its existing turbines to use non-standard fuels (GE 2016a). The capabilities to use hydrogen-containing fuel gases are shown in Table 10.

Table 10 GE turbines' capabilities to use H_2-containing fuel gases

	H_2 blends	High H_2	Syngas (O_2 blown)	Coke oven gas	Refinery/ process off-gas
LM2500	*	*	*	*	*
LM6000	*	*		*	*
LMS100	*	*		*	*
6B.03	*	*	*	*	*
9E	*	*	*	*	*
GT13E2	*				*
6F.01	*	*	*		*
6F.03	*		*	*	*
9F	*	*	*		*
9HA	*	*	*		*

GE's 44MW_e 6B.03 E-class frame gas turbine can operate with up to 95% hydrogen and it can operate with dry low-NOx combustion with up to 30% hydrogen (GE, 2016b). The GE hydrogen fleet leader would appear to be a Frame 6B unit at the Daesan petrochemical plant in Korea, which was installed in 1997 and which is reported to be routinely running with hydrogen concentrations between 85% and 97% (MPS, 2008).

There is experience of using Frame 6B, 6F, 9E and 7F turbines in coal and refinery IGCC plants (Jones, 2003). Fuel gas hydrogen concentrations are mostly around 22-45%vol before dilution but are as high as 62%. Steam and/or nitrogen is used to limit NO_x production and these diluents are injected into the combustor, rather than being mixed with the fuel. NO_x emissions of 9-25ppmv are achieved at US IGCC plants. GE's IGCC turbines are reported to be designed with dual fuel capability because of the dangers of starting on fuels containing hydrogen (Jones, 2003). This also gives the flexibility to operate using natural gas if the supply of syngas is restricted. The turbines can operate on 100% syngas or 100% natural gas or, in the "simplified extended turndown system" down to 15% natural gas with 85% syngas or 35% syngas with 65% natural gas.

GE's aero-derivative turbines are also capable of operating using hydrogen-rich fuels. For pre-mixed, dry-low emissions (DLE) combustion systems, the hydrogen content is limited to 5 percent by volume. The limit is due to fast flame speeds from high hydrogen fuels that can result in flashback or primary zone re-ignition. For single annular combustor systems, limits range from 35 percent H_2 by volume for larger turbines (up to 100 MW_e), to about 85 percent by volume for smaller turbines in the 18 MW_e to 30 MW_e power range. GE's LM2500+ and +G4 aero-derivative turbines are capable of operating with coke oven gas. The LM2500 has experience up to 65% H_2 (coke oven gas) in China and the LM6000 has experience up to 33% H_2 (petrochemical plant off-gas) in the USA (diCampli, 2014).

5.6.2 Siemens

Siemens' V94.2, V94.2K and V94.3 gas turbine designs have been supplied to large coal and residual oil IGCC plants. The V94.2 and V94.3 turbines are the precursors to Siemens' current SGT5-2000E and SGT5-4000F turbines and they are also manufactured by Ansaldo. As mentioned earlier, the V94.2K is a modification of the V94.2 designed to reduce the compressor mass flow and accommodate a higher turbine mass flow resulting from the addition of diluents in the combustor.

Siemens' work on hydrogen combustion has mostly been focussed on 60Hz turbines in the USA, as part of the US government funded Advanced Hydrogen Turbine Development Project. Siemens' large 60Hz turbines were originally developed by Westinghouse, which was taken over by Siemens, and they are not simply scaled versions of Siemens' 50Hz turbines. Nevertheless, going forward some common technology can be used in the different frequency machines, so developments for 60Hz turbines still have relevance to the UK market. Siemens 60Hz SGT6-5000F gas turbine is reported to support all levels of carbon capture in IGCC (i.e. high hydrogen fuel gas), meeting all emission and operability issues, with nitrogen addition (Brown, 2007). Siemens reports that their H class turbine will be available for service in IGCC plants by 2020 (Kraftwerkforschung, 2016).

Siemens also has experience with using high-hydrogen fuel gases in its medium sized industrial turbines, which are made in Sweden (Blomstedt, 2015). The same DLE combustor is used as standard in SGT600, 700 and 800 (25-50MW_e) turbines. Hydrogen enriched natural gas was verified during engine operation in 2012. Stable operation could be achieved using hydrogen fractions around 30-40% by volume. Further analysis of these hydrogen tests indicated that minor modifications to the standard burner could improve the hydrogen capability. Changes were implemented and new tests with modified burners were performed during 2014. A criterion for acceptable burner modifications was that natural gas capability should be kept with acceptable emissions. The tests in 2014 confirmed the possibility to run the SGT-700 on high hydrogen fuels, with results indicating 40-50% H_2 is possible at high loads. Based on these tests the accepted level of H_2 in the SGT700 and 800 was increased to 15%vol. At lower loads, higher hydrogen content is possible. At 10 MW load, 100% H_2 was satisfactorily demonstrated, but the NO_x emissions were about 60% higher than the high load emissions.

Siemens' smaller gas turbines, are also reported to have experience of high-H_2 fuel gases at many locations, including at refineries in the UK. The SGT-200 (currently 6.75MW), manufactured in the UK, is reported to have experience of 80-85% H_2 fuel and the SGT500-600 (currently 19-24MW) has experience of 20-90% H_2. As of 2007, Siemens' 7-25MW turbines were reported to have more than 750,000 operating hours experience on syngas and high hydrogen content refinery fuel gas (Wu,2007).

5.6.3 Ansaldo

Ansaldo's GT26 gas turbine, which it inherited from Alstom, is a reheat gas turbine with two different types of combustor; the first stage (EV) combustors, which operate in a similar way to the combustors in non-reheat turbine, and the second stage reheat (SEV) combustors, which are significantly different. The current SEV combustor is designed for natural gas and utilises large scale mixing devices to create a complex mixing pattern, into which fuel is injected though a lance. This design causes the rapid and uniform mixing of the reactants. As the vitiated air is above the auto-ignition temperature, combustion spontaneously occurs after a characteristic ignition delay time depending on the operating conditions and fuel type. The challenge in utilising hydrogen rich fuel is principally associated with its reduced auto-ignition delay time, which can be addressed in one of three approaches:

1. De-rating the engine – allowing the same mixing time by increasing the auto-ignition delay time through altering the characteristics of the vitiated air (i.e. the inlet temperature of the flow to the SEV).
2. Decreasing the reactivity of the fuel – i.e. by dilution with an inert gas.
3. Modifying the hardware – either to reduce the mixer residence time in-line with the reduced auto-ignition delay time or develop a concept which is less influenced by the reactivity of the fuel.

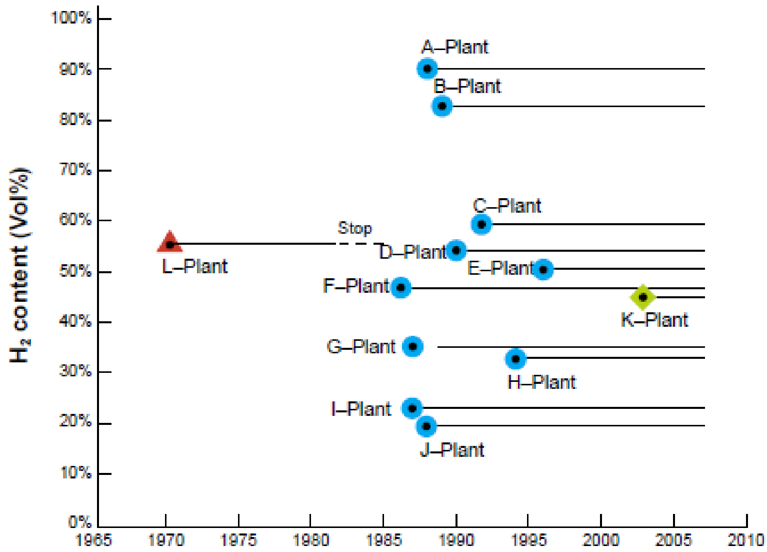
As part of the EU funded DECARBit project SINTEF developed a premixed reheat combustor technology which has demonstrated acceptable NO_x levels with a low dilution fuel consisting of hydrogen and only 30% nitrogen, while maintaining combustor pressure drop at an acceptable level (Erland, 2012).

The GT26 is reported by Alstom to have a capability of up to 10% H_2 with the existing hardware with only minor adaption in the gas supply and control system. More than 10% H_2 is feasible with an advanced SEV burner (Marx, 2013). Different fuels can be used in the two sets of burners.

5.6.4 Mitsubishi Hitachi Power Systems

Mitsubishi's operational experience of gas turbines fired with gases containing 20%vol or more hydrogen is shown in Figure 20 (Garnett, 2014).

In terms of experience of large turbines using high-hydrogen gas, Mitsubishi supplied a M701F gas turbine for an oxygen-blown, residual oil IGCC plant at Negishi, Japan which entered service in 2003. The plant has gross and net power outputs of 431MW and 342MW respectively. The turbine uses diffusion combustors. (Peltier, 2007). Mitsubishi has also installed a M701-DA gas turbine at a 250MW air blown coal fuelled IGCC plant at Nakoso, Japan, which started up in 2007. An M701G turbine was selected for the ZeroGen coal-fuelled, air-blown IGCC plant with CCS that was planned to be built in Australia. That project was however cancelled.



Symbol	Type	NOx reduction method
◆	Syngas	N_2 dilution
▲	COG	–
●	Oil refinery offgas	Steam injection

Total	
Number of units	12
Operating hours	>1300 000 (hrs)

Figure 20 Mitsubishi's operational experience of gas turbines firing hydrogen-rich gases

Source: University of Queensland

5.6.5 Solar Turbines

With support from the US Department of Energy, Solar Turbines has developed a dry low NO_x combustor for its Titan 130 turbine that is suitable for use with coke oven gas with up to 65%vol H_2. The Titan 130 firing natural gas has a power output of 16.5MW. The combustor is able to meet the NOx emission goal of 15ppmv for 50-100% load and showed favourable results on meeting operability and durability goals (USDOE, 2015). The fuel injector is capable of operating on diesel fuel or natural gas or high hydrogen fuels. Engine testing is needed to validate the rig results in future.

6. GAS ENGINES

6.1 Overview of gas engines used for grid-based power generation

Gas engines are spark-ignited lean-burn reciprocating engines, derived from diesel engines. The fuel gas is mixed with air before the inlet valves. During the intake period of engine operation, gas is also fed into a small prechamber, where the gas mixture is rich compared to the gas in the cylinder. At the end of the compression phase the gas-air mixture in the prechamber is ignited by a spark plug. Flames from the prechamber ignite the gas-air mixture in the main cylinder, resulting in rapid combustion. After the working phase the cylinder is emptied of exhaust gas and the cycle starts again.

Gas engines are reported to have some advantages compared to diesel engines, including lower costs (especially in high hours applications), lower emission capabilities, better suitability for variable load applications and no requirement for local fuel storage.

Advantages of gas engines compared to gas turbines include higher thermal efficiency in simple cycle mode, lower costs for small schemes (<10MW_e), better suitability for variable load applications, greater tolerance to high ambient temperatures and high elevations, lower fuel pressure requirements and fast start-up times (Caterpillar, 2014*).* Conversely, turbines have advantages of lower emission capability, less down-time per machine, simple design, compact equipment and better suitability for continuous operation.

6.2 Performance of gas engines

Gas engines are available with power outputs from <1MW up to nearly 20MW. The rotational speed is lower in large engines, for example 500rpm for MAN Diesel and Turbo's 12-19MW engines and 750rpm for its 3-10MW engines.

Performance of a range of gas engines is given in Table 11 (Corin, 2015; Losch, 2014; MAN Diesel & Turbo, 2016; Rolls Royce, 2015; Wärtsilä, 2016a; Wärtsilä, 2016b). Hot start means that the engine is in a pre-heated and pre-lubricated stand-by mode. It can be seen that the stated efficiencies of the engines are all similar, and significantly higher than those of the most efficient simple cycle gas turbine (42.5%).

Table 11 Performance of Gas Engines

Manufacturer	Model	Power (MW)	Efficiency (%)	Start time (minutes)	
				Hot	Cold
Wärtsilä	34SG	4.3-9.7	49	3	10
Wärtsilä	18V50SG	18.3	50	7	12
MAN	20V35/44G	10.4	49	5-short loading 10-normal loading	
Rolls Royce	Bergen B35:40	2.6-9.6	48	8	

The efficiency of a Wärtsilä engine was shown to be 5 percentage points lower at 50% load than at full load (Wärtsilä 2016b). The part load efficiency reduction is broadly similar to that of a typical large modern gas turbine. Gas engines however are often installed in power plants as multiple units and individual engines can be turned off when power demand is lower, resulting in better overall plant part load efficiency than for large gas turbines.

6.3 Costs of gas engines

Costs of gas engines are discussed in Section 4 of this report. Public data suggests that in a large power plant (200MW), gas engines have a slightly higher capital cost than simple cycle aero-derivative gas turbines.

In utility applications, diesel engines are reported to typically operate for 100-500 hours/year to satisfy peak load. The lower operating costs of gas engines allow increased operating times of 100-3000 hours/year (Caterpillar, 2014).

6.4 Use of hydrogen-rich gases in gas engines

Gas engines are operated using a wide range of gaseous fuels, including coal mine gas, landfill gas, sewage gas, flare gas, biogas, steel mill LD converter gas and coke oven gas. However, the only one of these gases that contains substantial quantities of H_2 is coke oven gas, and it is not widely used in engines.

Twelve Jenbacher gas engines have operated on coke oven gas since 1995 at the Profusa coke factory in Bilbao, Spain. Total operating hours are more than 1 million. The engines are specially modified type JGS 316 GS/N.L engines, operating at 1500rpm. They are designed to run with either coke oven gas, natural gas or a mixture with natural gas down to 30%. The engines are relatively small, the total power output of all twelve engines is 5.64MW with 100% coke oven gas and 6.528MW with 60% coke oven gas. The efficiency is 37% in both cases (GE Jenbacher, 2008).

Development work on hydrogen-fuelled internal combustion engines for cars has been undertaken. Although the engines are much smaller than would be needed for utility power generation, it is possible that some of the technology may be relevant larger engines in future. For example, BMW developed a new cylinder head for hydrogen operation based on a production diesel engine. Hydrogen was directly injected into the combustion chamber at pressures up to 300 bar. The engine achieved an efficiency of 42%, comparable to the best automotive turbo diesel engines (BMW, 2009).

7. REFERENCES

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APPENDIX GAS TURBINE DATA

Simple Cycle Gas Turbines														
Manufacturer	**Model**	**GT type**	**Net power**	**Efficiency**	**Pressure ratio**	**Mass flow**	**Exhaust temp**	**Minimum load**	**Start time Peaking**	**Start time Hot**	**Ramp rate**	**Ramp rate**	**50% load efficiency**	**50% load efficiency**	**Comments**
			MW	**%, LHV**		**kg/s**	**C**	**%**	**Minutes**	**Minutes**	**MW/min**	**%/min**	**%**	**load**	
Ansaldo	AE64.3A	Frame	78	36.3	18.3	215	573	45	15	20	7	9			
	AE94.2	Frame	185	36.2	12	555	541	45	15		30	16			
	AE94.2K	Frame	170	36.5	12	540	545								Low LHV fuel
	AE94.3A	Frame	310	39.8	19.5	750	576	43	25		22	7			
	GT26	Reheat	345	41	35	715	616	10		30	33	10			
GE Energy Oil & Gas	NovaLT5-1	Frame	5.6	30.7	14.8	20	574								
	GE10-1	Frame	11.25	31.4	15.5	47	482	50							
	NovaLT16	Frame	16	36	19	54	490								
	PGT20	Frame	17.46	35.2	15.7	63	475								
	PGT25	Frame	22.42	36.3	17.9	69	524								
	PGT25+	Frame	30.23	39.6	21.5	84	500								
	PGT25+G4	Frame	33.06	40	23.2	90	510								
GE Power Aero	LM2500	Aero	24.8	35.1	19	71	525	50		10	30	121			Water injection
	LM2500 DLE	Aero	23.2	36.6	18	68	539								DLE
	LM2500+	Aero	31.8	36.9	23.1	89	490	50		10	30	94			Water injection
	LM2500+ DLE	Aero	31.9	38.8	23.1	87	526								DLE
	LM2500+ G4	Aero	34.5	35.3	24.6	97	519	50		10	30	87			Water injection
	LM2500+ G4DLE	Aero	33.4	37.2	24	93	552								DLE
	LM6000PC	Aero	45.42	40.1	29.7	130	436	25		5	50	110			Water injection
	LM6000PC Sprint	Aero	51.06	40.4	31,5	135	449								Water injection, Spray intercooling
	LM6000PG	Aero	56	40.1	33.5	143	470	25		5	50	89			Water injection
	LM6000PG Sprint	Aero	59	39.8	34	144	480								Water injection, Spray intercooling
	LM6000PF	Aero	45	42	30.1	127	457	50		5	50	111			DLE
	LM6000PF Sprint	Aero	50	42	31.6	133	459								DLE, Spray intercooling
	LM6000PF+	Aero	53	41.8	32.1	135	500	50		5	50	94			DLE
	LM6000PF+ Sprint	Aero	57	41.4	34	143	490								DLE, Spray intercooling
	LMS100PA+	Aero	114	43.3	42.5	231	422	25		10	50	44	36.2	84	Water injection
	LMS100PB+	Aero	108	43.9	42.5	227	421	50		10	50	46			DLE

GE Power Heavy Duty	6B.03	Frame	44	33.5	12.7	145	548	50	10	12	20	45			
	6F.01	Frame	52	38.4	21	126	603	40	10	12	12	23			
	6F.03	Frame	82	36	16.4	213	613	52	-20	29	7	9			
	9E.03	Frame	132	34.6	13.1	419	544	35	10	30	50	38			
	9E.04	Frame	145	37	12.3	415	542	35	10	30	12	8			
	GT13E2	Frame	203	38	18.2	624	501	50	10	15	14	7			
	9F.03	Frame	265	37.8	16.7	665	596	35	20	23	22	8			
	9F.04	Frame	281	38.6	16.9	667	608	35	20	23	23	8			
	9F.05	Frame	299	38.7	18.3	667	642	35	10	23	24	8			
	9F.06	Frame	342	41.1	20	731	618	38	12	23	65	19			
	9HA.01	Frame	429	42.4	22.9	826	633	30	12	23	65	15			
	9HA.02	Frame	519	42.7	23.8	996	636	30	12	23	70	13			
Kawasaki Heavy Industries	M7A-03D	Frame	7.8	33.6	15.6	27	523	70					26.4	79	
	L20A	Frame	18.5	34.3	18.6	60	541	70					27.8	81	
	L30A	Frame	30.1	40.1	24.9	89	470						29.3	73	
MAN Diesel and Turbo	MGT6100	Frame	6.6	32.2	15	26	505	50							
	THM1304-12N	Frame	12	30.5	11	49	515								
Mitsubishi Hitachi Power Systems	H25 (42)	Frame	42.03	37.2	17.5	111	556	50		22	3.5	8			
	H50	Frame	57.45	37.8	19.5	151	564	50		22	4.7	8			
	H100 (100)	Frame	99.05	36.7	18.2	292	534	85		22	8	8			
	H100 (110)	Frame	112.44	38.2	19.3	308	538	70		22	9	8			
	M701DA	Frame	144.09	34.8	14	453	542	75		30	9	6			
	M701F3	Frame						75		30	18				
	M701F4	Frame	324.3	39.9	18	729	592	40		30	22	7			
	M701F5	Frame	359	40	21	730	611	45		30	36	10			
	M701G2	Frame	334	39.5	21	755	587	60		30	22	7			
	M701JAC	Frame	445	>41	23	893	615								
	M701J	Frame	470	41	23	893	638	50		30	58	12			
							-18								
PW Power Systems	FT8 SWIFTPAC	Aero	30.89	36.6	21.3	92	491								WLE
	FT4000 SWIFTPAC	Aero	51.83	41	29.9	150	441			<10					DLE
	FT4000 SWIFTPAC	Aero	70	41.3	36.3	177	425								WLE, wet compression

Siemens Energy	501-KB7S	Aero	5.24	32.8	13.9	21	498								
	501-KH5	Aero-STIG	6.45	41.9	12.5	19	530								Steam injected
	SGT-100	Frame	5.4	31	15.6	21	531								
	SGT-200	Frame	6.75	31.3	12.3	29	466								
	SGT-300	Frame	7.9	30.6	13.7	30	542								
	SGT-400	Frame	14.33	35.4	18.9	44	540								
	SGT-500	Frame	19.06	33.7	13	98	369								
	SGT-600	Frame	24.48	33.6	14	81	543								
	SGT-700	Frame	32.82	37.2	18.7	95	533								
	SGT-750	Frame	38.15	40.2	23.7	114	458								
	SGT-800	Frame	53	39	21.4	137	551	50							
	RB211-GT61 DLE	Aero	32.13	39.3	21.6	94	510								
	Trent 60 DLE	Aero	53.12	42.4	34.5	155	433			10					DLE
	Trent 60 DLE ISI	Aero	63.51	43.3	39.3	177	416								DLE, Inlet spray intercooling
	Trent 60 WLE	Aero	66	41.4	39.3	178	425	30		9					WLE
	Trent 60 WLE ISI	Aero	66	41.5	39.3	180	416								WLE, Inlet spray intercooling
	SGT5-2000E	Frame	187	36.2	12.8	558	536								
	SGT5-4000F	Frame	307	40	18.8	723	579	45							
	SGT5-8000H	Frame	400	40	19.2	869	627		17	30	35	9			
Solar turbines	Mercury 50	Recuperated	4.6	38.5	9.9	18	366								
	Taurus 60	Frame	5.67	31.5	12.4	22	510								
	Taurus 65	Frame	6.3	32.9	15	21	549								
	Taurus 70	Frame	7.96	34.3	17.6	27	507								
	Mars 100	Frame	11.35	32.9	17.7	43	485						22.1	67	
	Titan 130	Frame	16.45	35.2	17.1	50	496								
	Titan 250	Frame	21.74	40	24.1	68	463	40							
Glossary:															
DLE: Dry low NOx emission															
WLE: Wet low emission (water injection)															
ISI: Inlet spray intercooling															
STIG: Steam injected gas turbine															

Combined Cycle Plants

		1GT + 1ST		2GT + 1ST		1GT + 1ST	2GT + 1ST							
Manufacturer	**GT Model**	**Net power**	**Efficiency**	**Net power**	**Efficiency**	**Minimum load**	**Minimum load**	**Start time Hot**	**Start time Cold**	**Ramp rate**	**Ramp rate**	**50% load efficiency**	**50% load efficiency**	**Comments**
		MW	**%, LHV**	**MW**	**%, LHV**	**%**	**%**	**Minutes**	**Minutes**	**MW/min**	**%/min**	**%**	**% full load**	
Ansaldo	AE64.3A	115.8	53.8	233	54	50	50							
	AE94.2	277.5	54.6	561.5	55.2	60	60							
	AE94.2K													Low LHV fuel
	AE94.3A	456.3	58.8	913	58.9	50	50	45		42	9			
	GT26	502	60.1	1004	60.1	15	15		190			55.9	93	Reheat turbine
GE Power Aero	LM2500	34.2	49.1	68.6	49.3	33	17	30		30	88			Water injection
	LM2500 DLE	35	49.9	65.6	52.4									DLE
	LM2500+	41.5	49.2	83.2	49.4	34	17	30		30	72			Water injection
	LM2500+ DLE	44	53.4	88.2	53.6									DLE
	LM2500+ G4	48.2	49.6	96.8	49.7	34	17	30		30	62			Water injection
	LM2500+ G4DLE	47.7	53.8	95.7	54									DLE
	LM6000PC	57.9	51.5	116	51.7	19	19	30		50	86			Water injection
	LM6000PC Sprint	66.5	51.9	133	52									Water injection, Spray intercooling
	LM6000PG	73	52.2	146	52.4	19	19	30		50	68			Water injection
	LM6000PG Sprint	76	52.1	153	52.3									Water injection, Spray intercooling
	LM6000PF	58	54.9	117	55.1	37	19	30		50	86			DLE
	LM6000PF Sprint	64	54.4	128	54.6									DLE, Spray intercooling
	LM6000PF+	70	55.9	140	56.1	37	18	30		50	71			DLE
	LM6000PF+ Sprint	74	54.8	149	54.9									DLE, Spray intercooling
	LMS100PA+	135	51.5	270	51.6	21	21	30		50	37			WLE
	LMS100PB+	127	52.4	256	52.5	42	21	30		50	39			DLE
GE Power Heavy Duty	6B.03	67	51.5	135	51.7	57	29	30		20	30			
	6F.01	76	56.6	154	56.9	53	27	30		12	16			
	6F.03	124	55.4	250	55.9	59	30	45		7	6			
	9E.03	201	52.8	405	53.2	46	22	38		50	25			
	9E.04	212	54.4	428	54.9	46	22	38		12	6			
	GT13E2	289	55	581	55.2	56	56	80	240	14	5			
	9F.03	405	58.4	815	58.7	46	22	30	150	22	5			
	9F.04	429	59.4	861	59.8	45	22	30		22	5			
	9F.05	462	60.5	929	60.8	46	23	30		24	5	53.6	89	
	9F.06	508	61.1	1020	61.4	49	23	30		65	13			
	9HA.01	643	62.6	1289	62.7	38	18	30		65	10			
	9HA.02	774	62.7	1552	62.8	38	18	30		70	9			

Mitsubishi Hitachi Power Systems	H25 (42)	59.1	52.8	119.8	53.6			70						
	H50	83	54.4	166.3	55.2			70						
	H100 (100)	143.2	53.5	288.1	53.8			70						
	H100 (110)	157	54.4	322.8	55.9			70						
	M701DA	212.5	51.4	426.6	51.6			70						
	M701F3							70						
	M701F4	477.9	60	958.8	60.2			60						
	M701F5	525	61	1053.3	61.2			45						
	M701G2	498	59.3	999.4	59.5									
	M701JAC	650	>61											
	M701J	680	61.7						120			55	89	
PW Power Systems	FT8 SWIFTPAC	41	49.1	83.1	49.6									
	FT4000 SWIFTPAC	83.9	50.8	169	51.2									
Siemens Energy	SGT-600	35.9	49.9	73.3	50.9									
	SGT-700	45.2	52.3	91.6	53.1									
	SGT-750	49.3	52.4	99.2	52.7									
	SGT-800	74	55.6	150	56.2		24	30	110	24	32	48.3	87	
	RB211-GT61 DLE	42.6	52.8											
	Trent 60 DLE	65.3	53.6					40						DLE
	Trent 60 DLE ISI	77.7	53.4											DLE, Inlet spray intercooling
	Trent 60 WLE	81.2	51.4											WLE
	Trent 60 WLE ISI	82.9	51.2											WLE, Inlet spray intercooling
	SGT5-2000E	275	53.3	551	53.3									
	SGT5-4000F	445	58.7	890	58.7				220			53.5	91	
	SGT5-8000H	600	>60	1200	>60							55.5	89	
Note: Cold start data for existing plants, new plants may be different.														