Programme Area: Carbon Capture and Storage

Project: Hydrogen Turbines Follow On

Title: Assessment of LMS 100 Heat Management Options and Techno-Economic Parameters of Gas Turbine Power Plants

Abstract:

Development aspects and assessments of Gas Turbines (fired by methane and/or H_2) 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 CO_2, 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 CO_2 is in precombustion technologies, so use of H_2 in GT is also included.

ASSESSMENT OF LMS100 HEAT MANAGEMENT OPTIONS AND TECHNO-ECONOMIC PARAMETERS FOR GAS TURBINE POWER PLANTS

Report for

Energy Technologies Institute

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SUMMARY

LMS100 gas turbine

The steam cycle of an LMS100 gas turbine could provide more than sufficient steam for an amine CO_2 capture unit. The energy efficiency penalty for CO_2 capture and compression at an LMS100 combined cycle plant would be about 9 percentage points, the same as at a combined cycle plant based on Frame 9F gas turbines.

Simple cycle LMS100 plants are aimed at low-intermediate load power generation. An option for capturing CO_2 at such plants would be to install a simple HRSG that generates only low pressure steam for a CO_2 capture unit. Whether or not capturing CO_2 at such plants would be a feasible option would depend on the dynamic performance capabilities of the capture and compression units and the economics of CO_2 capture.

There is limited scope to use heat from the LMS100's compressor intercooler for a steam cycle or to generate low pressure steam for an amine CO_2 capture unit, due to the low temperature.

Recuperation is unlikely to be an attractive modification to the LMS100 turbine, due to the low temperature difference between the turbine and compressor exit temperatures.

Hydrogen and co-fired gas turbines

A wide range of gas turbines are reported to be suitable for high-H_2 fuel gas, with addition of diluents (nitrogen or steam) for control of emissions. Up to 95% H_2 gas has been used in commercial turbines.

The effects of using hydrogen and a diluent on the power output and efficiency of gas turbines depend on how the turbine is designed and operated and the system boundary, i.e. whether compression of nitrogen diluent is included. If nitrogen compression is outside of the system boundary, use of H_2 can result in a significant increase in the thermal efficiency of a combined cycle plant but if it is within the system boundary there would typically be a small decrease.

Published costs of gas turbines and combined cycle plants differ significantly between different sites and studies. Costs in UK£ are particularly uncertain due to variations in currency exchange rates. Typical costs of combined cycle and simple cycle plants based on large H or F class frame gas turbines appear to be around £500/kW and £330/kW respectively. Costs of plants based on smaller frame gas turbines or aero-derivatives appear to be about 50% greater, although this cost difference may be lower if plants based on large numbers of these smaller gas turbines were built.

1. INTRODUCTION

The Energy Technologies Institute (ETI) is focused on accelerating the deployment of affordable, secure low-carbon energy systems for 2020 to 2050. One of the key technologies is carbon capture and storage (CCS). CCS could be particularly suitable for decarbonisation of intermediate load power generation, to complement other lower carbon generation technologies. There is currently a large requirement for intermediate load generation in the UK due to the variability of power demand and this requirement is expected to increase in future as greater amounts of variable renewable generation are used. Technologies which may be well suited in this regard are post combustion capture and production, storage and use of hydrogen in gas turbine power plants. The ETI has requested supporting information on aspects of both of these technology options.

2. OUTLINE ASSESSMENT OF GE LMS100 HEAT MANAGEMENT OPTIONS

This task provides a "sense check" of the following three potential heat management options based around the GE LMS100 natural gas fired turbine with post combustion capture of CO_2.

- Use of heat from the turbine exhaust for a combined cycle and CCS unit.
- Use of heat from the compressor intercooler for a combined cycle and CCS unit.
- Use of some of the heat from the turbine exhaust to heat the compressor exit air by means of a recuperator. Note this would require modification to the existing LMS100.

Multiple turbines with different uses of the heat are considered. Direct use of heat from the turbine exhaust gas for amine reboiling is also considered.

This report is an initial assessment derived from published information. If the ETI wishes to further pursue any of the options they should obtain definitive information from the turbine vendor.

2.1 Description of GE LMS100 gas turbine

The GE LMS100 gas turbine is designed with an emphasis on flexibility and high efficiency particularly in simple cycle operation and as such it is particularly well suited to intermediate and peak load power generation. Diagrams of the LMS100 are shown in Figures 1 and 2.

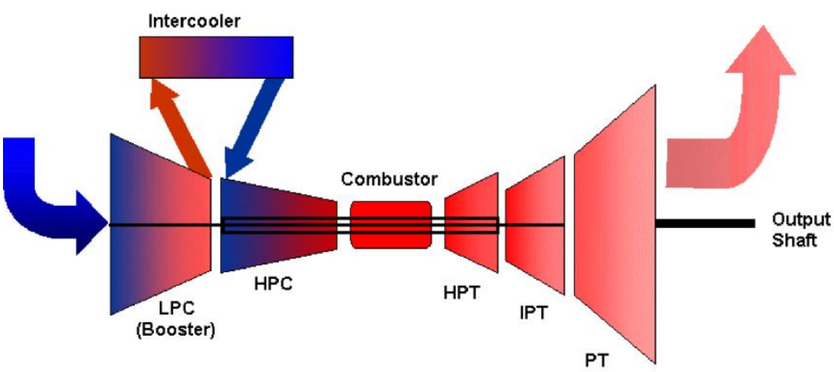

Figure 1 GE LMS100 (GE Energy, 2004)
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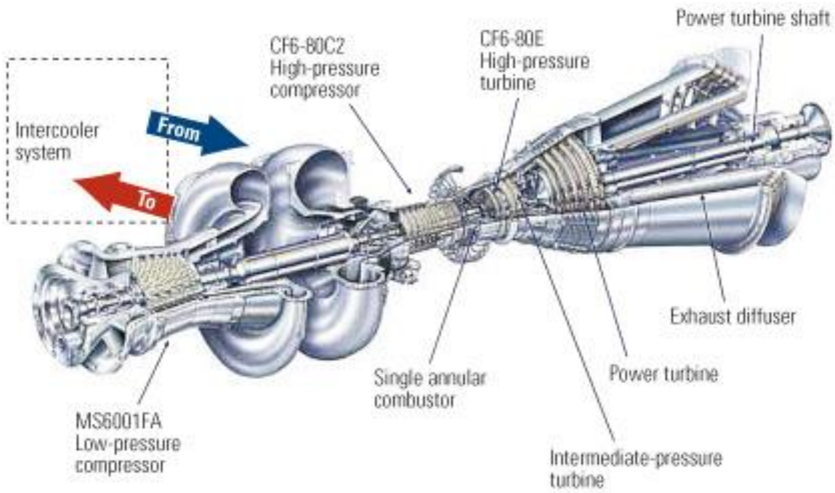

Figure 2 Cut-away diagram of the LMS100 (Peltier, 2007)
Source: General Electric

The LMS100 includes a low pressure compressor (LPC) based on the first 6 stages of the compressor of GE's 6FA frame gas turbine. Air from the LPC is cooled in an intercooler, which is a shell and tube heat exchanger. The cooled air is then fed to the high pressure core which is based on components of GE's CF6-80 aero engine. The core includes a 14 stage high pressure compressor, combustor and high pressure and intermediate pressure turbines. The high pressure turbine is connected by a shaft to the high pressure compressor and the intermediate pressure turbine is connected by a separate shaft to the low pressure compressor. Exhaust gas from the intermediate pressure turbine passes to a low pressure power turbine, designed specifically for the LMS100, which is connected to a generator. In the LMS100PA water is injected into the combustor in order to achieve 25ppm NOx. The more recently introduced LMS 100PB uses dry low emission combustion.

The LMS100 has a gross power output of 115 MW and an efficiency of 43.4% (LHV basis) when operated as a simple cycle plant. In a combined cycle the net power output is 135 MW and the efficiency is 51.4% (GE Power, 2016a). The turbine exhaust temperature is 422C, which is relatively low compared to modern large frame gas turbines, which typically have exhaust temperatures greater than 600C. As a consequence of the relatively low turbine exhaust temperature and heat availability the LMS100 uses a simpler, lower pressure steam cycle than current large frame gas turbines. The steam cycle is a 2-pressure non-reheat steam cycle with a high pressure steam turbine throttle pressure and temperature of 62.1 bar, 402C (GE Power, 2016a).

2.2 Use of heat from the turbine exhaust for a steam cycle and CCS unit

2.2.1 Modelling of LMS100 heat utilisation

An Excel-based simplified model of an LMS100 steam cycle has been developed to provide an initial estimate of how the steam required for post combustion capture could be provided, and the resulting power output and thermal efficiency of a combined cycle plant.

It is assumed that in the plant with 90% CO_2 capture, 4.5bar$_a$ steam is used to provide 3MJ of heat per kg of CO_2 captured, as in Foster Wheeler's baseline study for the ETI (ETI, 2010). The quantity of CO_2 in the turbine exhaust gas is calculated from the gas turbine fuel feed rate, which in turn is calculated from GE's quoted power output and efficiency (GE Power, 2016a). For simplicity the fuel is assumed to be pure methane. The auxiliary power consumptions for CO_2 capture and compression are estimated by scaling data from the baseline study carried out by Foster Wheeler for the ETI (ETI, 2010).

2.2.2 Steam cycle description and performance

In plants with and without CO_2 capture, high pressure (HP) and low pressure (LP) steam are generated in the HRSG. In the plant without capture all of this steam is fed through the steam turbine. In the plant with capture all of the HP steam is also fed to the steam turbine but the LP steam from the HRSG is fed to the solvent reboiler in the post combustion capture unit. The quantity of LP steam generated in the HRSG is insufficient for the reboiler so some of the steam flow through the turbine has to be extracted as LP steam to fully satisfy the needs of the reboiler.

The HRSG includes the following main tube banks:

- HP superheater
- HP evaporator
- HP/HT economiser
- LP evaporator
- LP and HP/LT economisers

The steam system has three pinch points:

1. At the high temperature end of the HRSG, where there is a 20C difference between the turbine exhaust gas temperature and the superheated steam temperature. Heat above this pinch point can be used for the HP superheater and evaporator.
2. At the gas exit from the high pressure evaporator, where there is assumed to be a 10C ΔT between the gas and steam side. Heat above this pinch point but below the first pinch point is used for the HP/HT economiser and the LP evaporator.
3. At the gas exit from the low pressure evaporator, where there is also assumed to be a 10C ΔT. Heat below this pinch point can be used for the LP and HP/LT economisers.

The amount of heat that is potentially available below the third pinch point is greater than that which is needed for the LP and HP/LT economisers. The HRSG gas exit temperature is therefore determined by the amount of heat that is needed for these economisers, rather than any other limits. This has implications for the potential to utilise low temperature heat from the compressor intercooler, which is discussed in section 2.3.

There is a greater surplus of heat for low temperature feedwater heating in the plant with CO_2 capture, because less of the steam flows to the steam turbine condenser and also the condensate from the CO_2 capture reboiler is returned at a higher temperature, close to its saturation temperature.

The estimated CO_2 and steam flowrates of plants with and without CO_2 capture are summarised in Table 1 and the power outputs and efficiencies are summarised in Table 2.

Table 1 Summary of LMS100 CO_2 and steam flows

		Without capture	With capture
CO_2 emitted	kg/s	14.6	1.5
CO_2 captured	kg/s	0	13.1
HP steam generation	kg/s	17.1	17.1
LP steam generation	kg/s	12.3	12.3
LP steam required for CO_2 capture	kg/s	0	18.6
LP steam from turbine extraction	kg/s	0	6.3

Table 2 Summary of LMS100 power outputs and efficiencies

		Without capture	With capture
Fuel input	MW (LHV)	265.0	265.0
Gas turbine power	MW	115.0	115.0
Steam turbine power	MW	21.9	7.7
Ancillary power: CO_2 capture	MW	0	4.9
CO_2 compression	MW	0	4.3
Power plant and misc.	MW	1.9	1.9
Net power output	MW	135.0	111.6
Net efficiency	% (LHV)	51.0	42.1

The estimated efficiency penalty for CO_2 capture is 8.9 percentage points, which is the same as for plants based on large frame gas turbines according to ETI's baseline study (ETI, 2010).

2.2.3 Multiple turbines with different uses of heat

The ETI has requested that multiple turbines with different uses of the heat should be considered. LMS100 turbines normally appear to be configured with a dedicated HRSG for each turbine, which is the configuration used normally in large gas turbine combined cycle plants. Multiple gas turbines could exhaust into a single HRSG. Whether this would be worthwhile would depend on various factors, particularly the trade-off between economies of scale in the HRSG versus the extra costs of longer gas turbine exhaust gas ducting and dampers to enable gas turbines to be isolated. Plants can be configured with multiple gas turbines and HRSGs feeding into a single steam turbine. GE provided data for LMS100 plants consisting of 1 gas turbine plus 1 steam turbine, and 2 gas turbines plus 1 common steam turbine. The latter has a 0.2 percentage point higher efficiency (GE Power, 2016a).

The addition of post combustion capture of CO_2 offers the possibility of more different configurations. For example in a power plant with multiple gas turbines, some HRSGs could produce steam only for use in a steam turbine (i.e. a conventional combined cycle plant) while other HRSGs could produce only steam for the capture units of more than one gas turbine. These latter HRSGs could generate high pressure steam which would be let down in a back pressure steam turbine to the lower pressure required for CO_2 capture. The amount of low pressure steam that could be generated from one gas turbine is estimated to be 27% less than the amount that would be needed to capture CO_2 from two gas turbines. The deficit of steam could be overcome if necessary by supplementary firing of the HRSG, but this would reduce the overall thermal efficiency. In conclusion, the configuration described above would not provide any efficiency benefits. It may limit the operational flexibility of the power plant because it would no longer be possible to operate some of the gas turbines independently while still capturing CO_2 from them. This alternative configuration therefore does not appear to be attractive.

2.2.4 Simple cycle with CO_2 capture

An option that may be of interest for relatively low capacity factor plants with CO_2 capture would be an LMS100 simple cycle in which the exhaust gas from the gas turbine is fed to a simple HRSG that would generate only sufficient low pressure for post combustion capture. The flue gas leaving this HRSG could be quench cooled prior to feeding to the post combustion capture unit. A simple low pressure, low temperature HRSG may impose fewer constraints on start-up, ramping etc than a conventional high pressure HRSG and steam turbine. However, whether this would be significant would depend on whether the main operating constraint in a plant with CO_2 capture would be the capture and compression units rather than an HRSG and steam cycle. Detailed assessment would be needed to investigate this. The capital cost of a simple low pressure HRSG should be significantly lower than that of a conventional high pressure steam cycle HRSG because of the larger gas/steam temperature differences and the low steam pressure. It would not be necessary to install HRSGs to provide steam for CO_2 capture at all of the gas turbines in a multi-unit power plant. For example, it is estimated that one LMS100 gas turbine would be able to generate about 88% of the steam required to capture 90% of the CO_2 from two LMS100 gas turbines. The efficiency of an "LMS100 simple cycle with capture" would be about 39%.

2.2.5 Direct heating of amine

It has been suggested that some of the amine for CO_2 capture could be regenerated by heating it directly in the HRSG instead of using low pressure steam (Botero, 2009). This reduces the size and number of external reboilers for the amine stripper, thereby reducing some practical constraints on the plant design. Replacing two-stage heat exchange (turbine exhaust gas to steam to amine) with a single stage (turbine exhaust gas to amine) reduces the overall temperature difference for heat transfer, which makes it possible to utilise more of the heat in the turbine exhaust gas. A further advantage is reported to be a lower capital cost (Botero, 2009).

The amine reboiler tubes would be positioned in the HRSG where heat could be transferred efficiently to the amine at about 120C, in the case of MEA. Thermal degradation of amine can be a serious issue in CO_2 capture units, so temperatures of no more than 130C in the hottest spots close to the tube wall must be guaranteed. This is reported to correspond to a temperature of around 250C in the bulk of the gas in contact with the amine reboiler tubes (Botero, 2009). Nevertheless, there may be a higher risk of hot spots and degradation when amine is reboiled using direct heat exchange from flue gas rather than the conventional technique of condensation of low pressure steam, which has a constant temperature that can be closely controlled.

In the plant evaluated by Botero, which is based on a 400MW GE 9FB frame gas turbine, 80% of the amine regeneration heat is provided directly in the HRSG with the remainder provided by steam extracted from the steam turbine. Similarly in an LMS100 plant some of the reboiler heat would have to be supplied by steam extraction. This may be advantageous as it may help to improve the control of the heat flow to the amine stripper, for example at different gas turbine loads.

It is estimated that using direct amine reboiling in an LMS100 plant would increase the overall thermal efficiency by <0.5 percentage points. The main advantage may be a reduction in capital cost. The main concerns would be the possibility of greater thermal degradation of amine. Control of the system at off-design and part load conditions would need to be investigated.

2.3 Use of heat from the compressor intercooler

The heat duty and temperature of the intercooler in the LMS100 depend on ambient conditions. The LMS100 intercooler temperatures and heat duty at standard ISO conditions (15C, sea level) are not known to be available in the public domain but some information at “hot and humid” conditions is available in a presentation that describes the use of heat from an LMS100 for seawater desalination (Mehmetli, 2014). In this presentation the intercooler air inlet conditions are shown to be 3.9bar and 200C and the exit conditions are 3.8bar and 35C. In this case the intercooler duty is $34MW_{th}$, which is around 15% of the total fuel input to the turbine. At ISO conditions (i.e. lower ambient temperature) the intercooler inlet temperature would be expected to be slightly lower due to a lower compressor air inlet temperature but the lower temperature should increase the mass flow which will tend to increase the intercooler heat duty.

The amount of heat transferred in the intercooler can also be estimated from an overall energy balance on the turbine. GE’s LMS100 fact sheet (GE Power, 2016a) specifies the power output, thermal efficiency and exhaust gas heat content. After taking account of mechanical, generator and miscellaneous energy losses, which are expected to be relatively small, the intercooler heat can be estimated by difference to be around 38MW at ISO conditions, which corresponds to an intercooler inlet air temperature of the order of 200C. The exhaust pressure of an air compressor with ISO inlet conditions and a 3.9 bar exit pressure is estimated to be about 180C, which tends to broadly confirm the earlier temperature estimate.

There are various possibilities for use of heat from the intercooler. It has been proposed that if an LMS100 was installed at a coal fired power plant the heat could be used to preheat cold low temperature boiler feedwater from the steam turbine condenser (Modern Power Systems, 2004). Much of the heat from the intercooler could be utilised in this way. Some of the intercooler heat could also potentially be utilised for district heating or desalination.

Another option considered here is to generate low pressure steam for use either in a steam cycle or in the reboiler of a CO_2 capture unit. The saturation temperature of $4.5bar_a$ steam, as required for the reboiler of the CO_2 capture unit in Foster Wheeler’s baseline study for ETI (ETI, 2010) is 148C. Allowing a 10C ΔT for the intercooler, only energy available from cooling gas above 158C would be suitable to generate steam for CO_2 capture. If the intercooler inlet gas temperature is 180C for example, approximately 5MW of heat could be utilised to generate low pressure steam, which would improve the overall thermal efficiency by around 0.5 percentage points.

As discussed earlier, in an LMS100 plant there is surplus heat available in the HRSG below the low pressure steam evaporation pinch point. Most of the heat available from the intercooler is below this pinch point and therefore cannot be utilised in the steam cycle or for amine reboiling, because there is surplus heat available from the HRSG. It is expected that the intercooler gas inlet temperature will be lower when the turbine is operating at part load, which would be an added complication for using the intercooler to generate steam for CO_2 capture.

2.4 Recuperated gas turbine

In a recuperated 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 3.

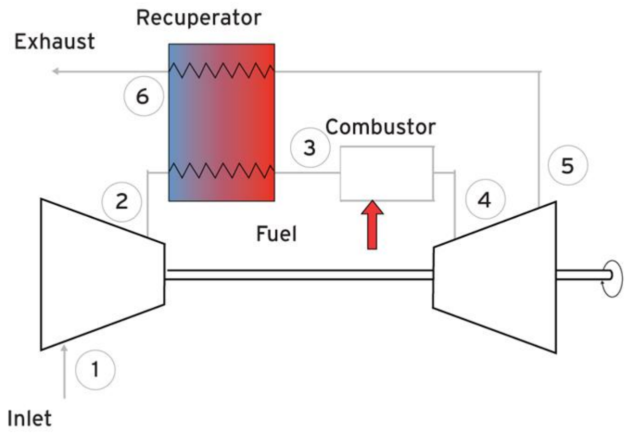

Figure 3 Recuperated gas turbine
Source: Ricardo

The heat transferred to the compressed air reduces the need for fuel in the combustor, thereby increasing the thermal efficiency of a simple cycle. In a combined cycle plant, recuperation reduces the amount and temperature of heat available for steam generation, so the benefits of recuperation are lower. The amount of heat that can be transferred in a recuperator depends on the temperature difference between the turbine and compressor exit gases. Recuperated gas turbines usually have a relatively low pressure ratio, which results in a low compressor exit temperature and high turbine exit temperature, which enables a larger amount of high to be recovered from the turbine exhaust gas. When higher pressures ratios are used in recuperated gas turbines, compressor inter-cooling is normally used, which decreases the compressor exit temperature and hence increases the amount of heat that can be transferred in the recuperator.

The only large (>10MWe) 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 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 pressure ratio of 16.2, which is lower than that of comparable non-recuperated turbines developed by Rolls Royce (the pressure ratios of the industrial RB211 and Trent are around 21 and 39 respectively) (GTW, 2016). In contrast, the GE LMS100 gas turbine has a pressure ratio of 42.5, which is the highest of any industrial gas turbine (GTW, 2016).

The turbine exhaust temperature of the 50 Hz version of the LMS100 is 422C (GE Power, 2016a). The compressor discharge temperature is estimated to be of the order of 380C. The amount of heat that could be transferred in a recuperator would therefore be small, after taking account of the necessary ΔT in a heat exchanger. To give an indication of the hypothetical benefits of recuperation, if the turbine exhaust gas could be cooled by 20C in a recuperator, it is estimated that the thermal efficiency of a simple cycle LMS100 would be increased by 0.8 percentage points, assuming there were no pressure drops through the recuperator and ducting. This is unlikely to be sufficient to justify the substantial modifications to the turbine.

3. TECHNO-ECONOMIC PARAMETERS FOR GAS TURBINES FIRING H_2 AND CH_4

3.1 ETI's requirements for hydrogen and co-fired gas turbines

The ETI is interested in scenarios in which gas turbine power plants will burn hydrogen or combinations of hydrogen and natural gas. The hydrogen will be stored in salt caverns to enable the hydrogen production plants to be operated continuously while the power plants operate intermittently, to satisfy the needs of the electricity grid. The gas turbines will need to be able to operate flexibly and start up and shut down rapidly. The ETI is interested in power plants with an overall power output of 600-1000MW, based on single or multiple gas turbines.

The ETI has employed Baringa to predict the loading scenarios of the overall fleet of gas turbines in 2030. The total number of hot, warm and cold starts per year for existing and new CCGTs in three scenarios are shown in Table 3.

Table 3 Starts per unit for existing and new CCGTs.

	Total number of starts per unit for existing and new CCGTs					
	Hot		Warm		Cold	
	Existing	New	Existing	New	Existing	New
Scenario 1	23	68	45	85	25	14
Scenario 2	30	136	58	88	26	7
Scenario 3	15	59	36	82	24	20

New CCGTs have higher efficiencies and hence lower marginal operating costs, which means they tend to be operated in preference to older turbines, resulting in higher annual capacity factors. The new turbines bear the brunt of the variability in power demand and they have to make larger numbers of hot and warm starts than older turbines, for example to cope with the lower electricity demands overnight and at weekends. It can be seen that the older existing turbines are predicted to have proportionately more cold starts, because they are forced to shut down for long periods of time more often than the new turbines.

3.2 Types of gas turbines

3.2.1 Natural gas fired turbines

A database and summary of current gas turbines was produced for the ETI in August 2014 (Davison, 2014). Supplier are continuously improving their gas turbines to maintain competitiveness and even since the time of the earlier report further improvements have been made to the efficiencies of some gas turbines and combined cycle plants.

The large frame gas turbines from the three largest manufacturers (GE, Siemens and Mitsubishi) tend to have similar sizes and efficiencies. The different manufacturers introduce new models at different times so there is a "leapfrogging" amongst them for greatest efficiency and size. The largest and most efficient 50Hz turbines currently appear to be GE's 9HA.01 and 9HA.02, and Mitsubishi Hitachi Power Systems' M701JAC, which have combined cycle efficiencies in excess of 63%. Siemens' SGT-8000H is reported to have an efficiency of 61%. The other large turbine manufacturer is Ansaldo which recently announced its GT36 reheat gas turbine. This is reported to have a combined cycle efficiency of 61.5% and like its predecessor the GT26 it has particularly good flexibility and turn-down capabilities.

The main manufacturers also produce medium and small frame gas turbines which are aimed mainly at industrial and CHP plants. These are either older, less efficient large turbines or more modern turbines specifically designed to address the smaller size market but which nevertheless tend to have lower inlet temperatures and efficiencies than the latest large power generation turbines.

At smaller sizes (<70MW) are aero-derivative gas turbines, which are used mainly for low load simple-cycle power generation or for industrial plants.

A unique turbine is the GE LMS100, described in section 2, which is a combination of aero-derivative and frame gas turbine technology. This has a simple cycle power output of 115MW and a high efficiency of over 43%.

Turbine efficiencies are expected to continue to increase in future due to the availability of more advanced materials and improved cooling techniques, which will enable turbine inlet temperatures to be increased. Other improvements such as improved aerodynamic design may also contribute to higher efficiencies. Quantifying future improvements, particularly over the long term, is subject to uncertainty but 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).

3.2.2 Experience of using hydrogen in current commercial gas turbines

The capabilities of existing commercial gas turbines to use hydrogen was summarised in a recent report prepared for the ETI (Davison, 2016). GE in particular has reported on the capabilities of its turbines to use fuel gases containing hydrogen (GE Power, 2016b). Particularly notable are:

- GE6B.03 has operated with up to 95% hydrogen
- LM2500+ aero-derivative has operated with up to 65% hydrogen
- LM6000 aero-derivative has operated with up to 33% hydrogen
- 9F, 9HA and LMS100, amongst others, are suitable for H_2 blends and "high H_2" fuel gas. F class turbines have operated with hydrogen contents up to 45% by volume for more than 15 years (GE Energy, 2011)

3.2.3 Turbine selection

Some of GE's gas turbines have been selected to represent different types of turbines that could be used in ETI's techno-economic assessments. This should not be considered to be a particular endorsement of this turbine vendor, the selection has been influenced by the availability of published data on performance, costs and ability to use hydrogen (all of the turbines are reported to be suitable for "high H_2" fuel gas). The selected turbines are as follows:

- The 6B is an example of a relatively old technology mid-sized turbine. It was selected because there is experience of operating it on 95% H_2 fuel gas.
- The LMS100 is targeted at the low/intermediate load market that is of particular interest to the ETI and it has a high simple cycle efficiency.
- The 9F was selected because it has been a market-leading large gas turbine since the 1990s and performance simulations for H_2 fired turbines are available. Data is provided for the widely used .05 variant although there is now a larger and slightly more efficient .06 variant.
- The 9HA.01 is an example of the latest generation of large high efficiency gas turbines.

3.3 Techno-economic data for natural gas fired turbines

3.3.1 Performance data

Power outputs and efficiencies of selected natural gas fired gas turbine power plants are given in Table 4. The data are for ISO conditions (15C, sea level, 60% humidity), which is the basis normally used by gas turbine vendors. Average UK temperatures are somewhat colder than ISO conditions, particularly during winter when demand is at its highest, and this can increase the power output and efficiency. Turbines suffer degradation during operation, particularly due to blade fouling, some of which can be regained by washing and some of which cannot. The efficiencies quoted in this report do not take account of degradation.

The combined cycle plant data in Table 4 are for plants consisting of one gas turbine and one steam turbine. Plants with two gas turbines and one steam turbine have efficiencies 0.1-0.2 percentage points higher (GTW, 2016). To achieve the ETI's target output of 600-1000MW, combined cycle plants would require one 9HA gas turbine, two 9Fs, 5-7 LMS100s or 9-14 6Bs. For simple cycles, 2 9HA, 2-3 9Fs, 6-8 LMS100s or 14-22 6Bs would be required.

Table 4 also shows start-up times and minimum loads. Table 3 shows that most of the start-ups of new gas turbines are expected to be from a "hot" or "warm" state, and there will be few "cold" starts. Table 4 shows hot start-up times, which are the only data that are usually publically quoted by turbine vendors. In some cases turbine vendors quote "peaking start-up" times as well as normal hot start-up times and these are shown in brackets in Table 4. "Peaking" start-up is faster but it incurs additional maintenance costs. All of the plants appear to be able to comply with the start-up times shown in a load cycle diagram provided by the ETI. The 6B combined cycle would appear to not be able to achieve the minimum load for ETI's Scenario 2 but it would be possible to achieve the minimum load in a multi-turbine plant by shutting down some of the turbines.

Table 4 Performance data for selected gas turbines

	Simple cycle				Combined cycle			
	Power MW	Efficiency %	Start-up minutes	Min. load %	Power MW	Efficiency %	Start-up minutes	Min. load %
6B.03	44	33.5	12 (10)	50	67	51.5	30	57
LMS100	115	43.4	8	15	135	51.4	30	13
9F.05	314	38.2	23 (12)	35	493	60.7	30	46
9HA.01	446	43.1	23	30	659	63.4	<30	38

3.3.2 Capital cost reference data

Costs of natural gas fired combined cycle and simple cycle plants obtained from various references are shown in Tables 5 and 6 and an overview of average costs based on these reference data are given in Table 7.

The combined cycle plants shown in Table 5 consist of either one, two or three gas turbines and a single steam turbine. The Leigh Fisher costs are from a report produced in 2016 for the UK DECC (Leigh Fisher, 2016). The cost data are based on PEACE software from Thermoflow (the suppliers of GTPRO), adjusted upwards to reflect the contractor's experience of real plants. The Parsons Brinckerhoff data are from a study on gas fired power plants with CO_2 capture carried out for IEAGHG (IEAGHG, 2012). The EIA data are an assessment of average costs for plants in the USA published by the US Energy Information Administration. The Gas Turbine World costs are from the 2014-15 GTW Handbook (GTW, 2015). The Brattle costs are for plants at 5 sites within the USA (Brattle Group, 2014) and the costs in the tables are the average reported costs. The difference between the highest and lowest cost sites was about 20%. The EEE costs are a compilation of costs from several US sources (Energy and Environmental Economics, 2014). The Analysis Group/Lummus costs are the average of costs for six sites in a study undertaken for the New York Independent Supply System Operator (Analysis Group, 2016). Capital costs per kW in this study varied substantially between different sites, by around 25% for simple cycle plants and as much as 70% for combined cycle plants. Most of the costs in Tables 5 and 6 are EPC costs, excluding interest during construction, owner's costs, interconnections etc. Data shown in the tables in italics are on a different basis but they have been included to provide further information on the relative costs of different turbines. The EEE and Lummus costs are believed to be total plant costs including the additional costs mentioned above, and as a consequence they are higher. The Gas Turbine World simple cycle plant costs are only for the major equipment, and hence they are lower (note the GTW combined cycle costs are EPC costs).

Most of the equipment in gas turbine power plants originates outside of the UK. The source cost data for gas turbines are usually quoted in US$. Translation of US$ costs into UK£ is uncertain, especially in view of the recent large variations in currency exchange rates, particularly the strength of the US$. Costs also depend on market factors (supply/demand etc.) and local site conditions which mean that costs can vary substantially even within a country, as mentioned above. For the purposes of Tables 5 and 6 any costs in US$ have been converted to UK£ using an exchange rate of 1.55 $/£, which was typical of the exchange rate around the dates of the references.

Table 5 Combined cycle plant cost reference data

Source	Gas turbine	Number of gas turbines	Power output MW	Capital cost £/kW	Fixed OPEX £k/MW.y	Variable OPEX £/MWh
Leigh Fisher	9H	2	1200	516	14	1.4
	9F	3	1471	475	13	1.4
Parsons Brinckerhoff	9FB	2	910	559	20	0.7
EIA, 2013	7F	2	620	493	8	2.3
	7H	1	400	550	10	2.1
Gas Turbine World	9HA.01	2	1181	405	-	-
	9HA.01	1	592	428	-	-
	9F.05	2	923	412	-	-
	9F.05	1	460	430	-	-
	6B.03	2	135	607	-	-
	LMS-100PA	1	127	632	-	-
Brattle	7FA	2	585	523	20	1.7
EEE (Total plant costs)	7FA	2	-	*726*	6	-
	7G/H	1	-	*790*	6	-
Lummus (Total plant costs)	SGT6-8000H	1	417	*970*	11	-
	SGT6-5000F5	1	340	*1074*	12	-

Table 6 Simple cycle plant cost reference data

Source	Gas turbine	Number of gas turbines	Power output MW	Capital cost £/kW	Fixed OPEX £k/MW.y	Variable OPEX £/MWh
Leigh Fisher	9H	1	400	330	9	1.0
	9F	2	625	291	8	0.9
	6F	1	80	577	14	1.5
	Aero-derivative	-	97	779	22	2.2
EIA	7E	1	85	523	5	10.0
	7F	1	210	363	5	6.7
Brattle	7FA	2	390	279	12	2.7
GTW (Major equipment costs only)	*9HA.01*	*1*	*397*	*146*	-	-
	9F.05	*1*	*299*	*152*	-	-
	LMS100	*1*	*106*	*230*	-	-
	6F.03	*1*	*80*	*248*	-	-
	6B.03	*1*	*44*	*260*	-	-
EEE (Total plant costs)	E/F Frame	1	85-210	*532*	6	-
	Aero/LMS100	-	50-200	*774*	10	-
Lummus (Total plant costs)	7HA.02	1	333	*599*	6	-
	SGT6-5000F5	1	228	*665*	7	-
	LMS100	2	208	*988*	8	-

3.3.3 Operating cost reference data

Operating costs consist of fixed costs (£/installed MW.y) and variable costs (£/MWh). Costs from various references are given in Tables 5 and 6, although not all of the references provided operating cost data.

Fixed costs include fixed maintenance costs, insurance, staff costs, general and administrative costs and business rates. The Leigh Fisher reference also gives information on UK grid connection and system costs, which amount to about £3k/MW.y in the “medium” case but up to £23k/MW.y in the “high” case. These costs have not been included in the tables.

Variable costs include miscellaneous consumables and the cost of a long term service agreement (LTSA), which is often expressed in £ per fired hour per MW of installed capacity. For Tables 5 and 6 the LTSA costs have been converted into a £/MWh cost assuming the plants operate at full load when they are operating. The classification of maintenance costs as fixed or variable costs is sometimes a matter of judgement, which probably accounts for some of the differences between the different references.

When operating at lower capacity factors, such as shown in the load cycles used by Atkins for ETI, there will be relatively high numbers of start-ups and shut-downs which can result in additional maintenance costs. These costs are sometimes expressed in terms of additional equivalent operating hours (EOH), although this measure is not universally preferred by turbine manufacturers. Leigh Fisher assume 8 EOH for a start. There are also additional fuel costs associated with start-ups and shut-downs, which would depend on the specific fuel price (£/GJ), but these costs are not included in the tables in this report.

3.3.4 Overall cost data for selected plants

The following conclusions can be drawn based on the information in Tables 5 and 6:

- Specific capital and operating costs of H and F class turbine plants appear to be similar, some sources indicating F class turbines are cheaper and others indicating the opposite. Actual costs would depend on market factors.
- Typical capital costs of combined cycle plants based on H or F class turbines are around 500 £/kW and simple cycle plant costs are around 330 £/kW.
- Capital costs of combined cycle plants based on smaller frame gas turbines (e.g. Frame 6) and/or older technologies (B or E class turbines) or aero-derivative turbines (including the LMS100) are about £750/kW, i.e. about 50% higher than those of H or F class combined cycle plants.
- Capital costs of simple cycle plants based on smaller frame gas turbines or aero-derivatives are around 50% higher than simple cycle plants based on H or F class turbines, i.e. around £500/kW.
- Operating costs of smaller frame and aero-derivative turbine plants appear to be higher than those of large frame gas turbine plants.

Based on the data given in the previous sections, power outputs, efficiencies and costs of selected turbines are given in Table 7 (performance data are given in Table 4). The costs are for the typical plant sizes of the reference data in Tables 5 and 6.

Table 7 Cost data for selected gas turbine power plants

	Capital cost £/kW	Fixed operating cost £k/installed MW.y	Variable operating cost £/MWh
Simple cycle			
6B	500	10	1.5
LMS100	500	13	2.2
9F	330	8	1.3
9HA	330	8	1.3
Combined cycle			
6B	750	15	1.7
LMS100	750	18	2.0
9F	500	12	1.6
9HA	500	12	1.6

The simple cycle plants shown in Table 6 mostly have lower power outputs than the combined cycle plants shown in Table 5. There should be some cost savings if multiple simple cycle turbines were installed at a site to give the same power output as larger combined cycle plants, due to economies of scale in balance of plant equipment and site costs. There would also be economies of scale for larger combined cycle plants with multiple gas turbines for each steam turbine. According to the Gas Turbine World Handbook the costs/kW for F-class combined cycle plants based on 2 gas turbines and a single steam turbine are only about 5% lower than for plants based on one gas turbine and one steam turbine. For smaller industrial frame and aero-derivative turbines the reductions in specific costs for 2-gas turbine plants are larger, around 15%. Some further savings would be expected for plants based on more than two gas turbines.

3.4 Gas turbines using hydrogen

3.4.1 Impacts of using hydrogen on power output and efficiency

The impacts of the use of hydrogen on the efficiency and power output of gas turbines and combined cycle plants were discussed in section 5.5 of an earlier report for the ETI (Davison, 2016).

The performance of natural gas and hydrogen-fired CCGTs based on F-class gas turbines has been predicted using a detailed turbine model (Chiesa, 2005; Grazzani, 2014). The main conclusions of the analysis are summarised below:

- If no diluent is used to control emissions, using hydrogen 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.
- If nitrogen or steam are used as a diluent in the combustor, the efficiency of a hydrogen fired combined cycle plant is the same as or lower than that of a natural gas fired plant. 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, taking into account the power required for compression of nitrogen diluent.
- If the compressor air flow and the turbine nozzle area are kept constant, burning hydrogen with an added diluent increases the pressure ratio of the turbine from 17:1 to 18.5:1 in the case of steam and to 19.7:1 in the case of nitrogen. Given the compressor stall margins available on existing machines it may not be possible to achieve this highest pressure ratio without any modification to the machine and probably one or more high pressure compressor stages would need to be added to shift the compressor surge limit upwards (Chiesa, 2005).
- A method of keeping the pressure ratio constant despite the increased mass input from hydrogen fuel and diluents, thereby avoiding the risk of compressor surge, is to reduce the compressor air flow by closing the variable guide vanes of the compressor. Closing compressor guide vanes is a normal procedure for turn-down of a gas turbine. A disadvantage of closing the guide vanes when using hydrogen is there would be less scope for using this technique to efficiently turn down the gas turbine.
- 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 ratio.
- Using hydrogen as the fuel instead of natural gas increases the power output of a gas turbine. 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 should reduce the specific cost (£/kW) it may exceed the mechanical limits of the turbine. If the compressor air flow is reduced by adjusting the variable guide vanes of the compressor, while keeping the pressure ratio constant, the increase in power output is reported to be lower, about 16% (Chiesa, 2005).
- 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.

As mentioned above, the ability of a gas turbine to operate using hydrogen and nitrogen and/or steam diluents depends on the compressor surge limits, which are proprietary information of the turbine manufacturers. Reducing the compressor air flow rate using the variable guide vanes is less likely to result in surge than increasing the pressure ratio (Chiesa, 2005). The performance of a natural gas fired combined cycle plant and a plant fired on H_2 diluted by N_2, with the air flow reduced using variable guide vanes, is summarised in Table 8.

Table 8 Combined cycle performance using natural gas and H_2/N_2 (Chiesa, 2005)

		Natural gas	**H_2/N_2**
Compressor air flow	kg/s	633.8	550.7
Fuel flow	kg/s	15.02	5.52
Nitrogen diluents flow	kg/s	0	79.67
Turbine exhaust temperature	C	585.1	574.2
Gas turbine power	MW	256.8	297.6
Combined cycle power, (including nitrogen compressor)	MW	387.2	380.2
Combined cycle power (excluding nitrogen compressor)	MW	387.2	422.7
Combined cycle efficiency (including nitrogen compressor)	%	57.57	57.46
Combined cycle efficiency (excluding nitrogen compressor)	%	57.57	63.91

The efficiency of the combined cycle plant using H_2 diluted by N_2 is 0.1 percentage point lower than that of the plant fired by natural gas fired, when the nitrogen compressor power consumption is taken into account. Excluding the nitrogen compressor, the efficiency of the H_2/N_2 fired plant is 5.6 percentage points higher than that of the natural gas fired plant.

The performance of combined cycle plants based on two GE 9F gas turbines fired by natural gas or hydrogen diluted by nitrogen has been reported by Parsons Brickerhoff in a report for IEAGHG (IEAGHG, 2012). In the plant fired by hydrogen the compressor air flow was reduced by varying the compressor guide vanes. The performance data modelled using GTPRO is summarised in Table 9. The hydrogen fired plant includes a H_2/N_2 compressor with an inlet pressure of 20bar, i.e. substantially higher than in the study by Chiesa shown in Table 8.

It can be seen that in this case the H_2 fired plant has an efficiency 1.3 percentage points higher than the natural gas fired plant. The increase in gas turbine power output when firing hydrogen is about 8%, i.e. less than the 15% in the study by Chiesa reported in Table 8.

Table 9 GE 9FB combined cycle performance using natural gas and H_2/N_2

		Natural gas	**H_2/N_2**

Gas turbine data			
Compressor air flow	kg/s	656.9	599.7
Fuel and diluent flow	kg/s	16.6	81.6
Turbine exhaust flow	kg/s	673.6	681.3
Turbine exhaust temperature	C	639.5	619.5
Gas turbine power	MW	295.2	320.0
Overall combined cycle plant data			
Fuel feed (LHV)	MW	1546.2	1536.5
Gas turbine power (2 turbines)	MW	590.5	640.1
Steam turbine power	MW	343.5	323.3
Auxiliary power consumption	MW	23.7	38.2
Net power output	MW	910.3	925.1
Combined cycle net efficiency	%	58.9	60.2

3.4.2 Impacts of using hydrogen on turbine costs

The impact of firing hydrogen on plant capital costs is uncertain. Some costs will increase, for example there will be larger and more expensive fuel and diluent pipes and valves, different combustors etc. If nitrogen compression is considered to be within the scope of the power plant, this will also increase the power plant cost. Conversely, the power output from the gas turbine may be higher, which would reduce the cost per kW. The duties of the HRSG, steam turbine and ancillaries will be different but the specific costs are not expected to differ substantially. A study carried out for IEAGHG by Parsons Brinckerhoff gives capital costs for F-class combined cycle plants fired by natural gas and H_2 (with nitrogen dilution) (IEAGHG, 2012). The net power output of the H_2 fired combined cycle plant is 1.6% higher and the cost per net kW is 1% higher than that of the natural gas fired plant.

In the short term, vendors may consider hydrogen fired turbines to be a niche product for which they may charge a higher price, to recover costs of non-standard manufacturing, product development costs etc., but this would not be expected to continue if there becomes a large market for hydrogen fired turbines.

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