ABSTRACT
The enhanced SGT-800 gas turbine was introduced in 2010 at a power rating of 50.5MW with 38.3% electrical efficiency in simple cycle (ISO). Combined-cycle performance of 55.1% is best-in-class. The updated components in the gas turbine are interchangeable from the 47.5MW rating. Performance is mainly enhanced by improved compressor airfoil profile resulting in about 1% higher mass flow and improved aerodynamics, improved turbine stage #1 aerodynamics and cooling air layout and improved turbine stage #3 with increased outlet area (reduced exit losses) and improved aerodynamics. The turbine inlet temperature is not increased compared to the 47.5MW rating; however, vane #1 outlet temperature is increased due to both improved stage #1 cooling design and fewer, thicker vanes, and the combustor-turbine interface is accordingly updated. The annular combustion chamber serial cooling system is improved to reduce pressure drop and optimize cooling layout and the combustor cold parts have been modified to optimize cycle life. Minor burner tuning is used in combination with combustor passive damping which results in low emissions for >50% load and which is insensitive to ambient conditions. The combustion system has shown excellent combustion stability properties, during rapid load changes and large flame temperature range at high loads, which may be used to obtain single-digit Dry Low Emission (DLE) NOx. The combustion system performance has also been proven for a broad range of gas fuel compositions including high amount of hydrogen and DLE liquid operation.

The first SGT-800 with 50.5MW rating was successfully tested during Spring 2010 and the expected performance figures were confirmed. The fleet leader has, up to March 2013, accumulated >17,000 equivalent operation hours (EOH) and a planned hot-section borescope follow-up inspection after 10,000 EOH showed that the core parts were in good condition.
INTRODUCTION
The SGT-800 is the largest industrial gas turbine manufactured by Siemens Industrial Turbomachinery AB and it was launched in 1997 as a 43MW machine named GTX100 [1]. The SGT-800 was shortly after rated to 45MW and in 2007 the engine was enhanced to 47.5MW with 37.7% simple cycle efficiency [2, 3], 53.8% combined cycle operation efficiency and including cogeneration (district heating) the efficiency further increased up to 94% [3]. Siemens has since then continued the stepwise evolutionary development based on experience and proven design solutions in order to always assure high reliability. In 2010, the SGT-800 was upgraded once more with higher power output, higher efficiency and improved combustion performance. The performance resulted in 50.5MW with a 38.3% simple cycle electrical efficiency (ISO) and best in class combined-cycle performance of 55.1%. The design modifications, validation and operation experience associated with the upgrade are described in this paper.

Up to March 2013 about 180 SGT-800 units have been sold. The SGT-800 fleet has accumulated more than 2.2 million Equivalent Operating Hours (EOH) and the fleet leader has passed 90,000 EOH. During 2011 the machine has achieved an excellent average fleet availability and reliability of 96.9% and 99.8%, respectively. Totally about 20 units of the new rating are in operation, under commissioning or in delivery phase. Both the 47.5MW and the 50.5MW ratings are offered to the market.

The SGT-800 is a single shaft engine that consists of inlet housing, 15-stage axial compressor, an annular serial cooled combustor, a 3-stage axial turbine and an outlet diffuser. In order to meet the engine operation requirements the first 3 stages of the compressor are made of variable guide vanes. The combustor is equipped with 30 Dry Low Emissions (DLE) dual-fuel burners with a capability of NOx ≤ 15ppmv (≤ 25-42ppmv on diesel oil) and CO ≤ 5ppmv in the load range of 50-100%. The first two turbine stages are air-cooled and for the stage 1 blades single crystal material is used. In addition to its high efficiency in simple cycle, the machine is especially suitable for combined cycle operation and cogeneration due to its high temperature after the exhaust diffuser.

DESIGN MODIFICATIONS
The up-rate project started with improvement goals in performance and component durability. To keep the high reliability of the hot components, the turbine inlet temperature of the up-rated design was decided to be the same as compared to the 47.5MW version. The turbine inlet pressure and mass flow were slightly increased in order to obtain higher power output and efficiency.

The design changes of the engine are shown in Figure 1, where it can be seen that the major changes were carried out in the turbine section. Table 1 shows how the main flow parameters are changed for the upgraded design as compared to the 47.5MW version, where the reduction in cooling mass flow is the largest factor.
Table 1: 50.5MW versus 47.5MW design.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine inlet total temperature</td>
<td>0%</td>
</tr>
<tr>
<td>Turbine inlet total pressure</td>
<td>4.4%</td>
</tr>
<tr>
<td>Compressor inlet mass flow</td>
<td>1.4%</td>
</tr>
<tr>
<td>Coolant mass flow</td>
<td>-17.2%</td>
</tr>
</tbody>
</table>

Compressor
To meet the design targets of increased pressure ratio and mass flow new enhanced blade profiles for all 15 stages was introduced to improve the compressor aerodynamic performance. By optimizing the front stage geometry using 3D CFD calculations the mass flow increase could be achieved within the existing flow channel. Together with the fact that all new blades are attached to the original blade root, the rotor and stator design is unchanged, which offers the ability to retrofit existing customer engines. In addition to increased mass flow, the new blade design also offers increased compressor efficiency and operating range both in terms of pressure ratio and temperature. Particularly, improved compressor stability means that the compressor can operate with fully open inlet guide vanes at higher ambient temperatures which maintains a high mass flow and consequently increased gas turbine hot ambient power output.

The compressor is also available as a hot match version where the variable guide vanes are allowed to be fully open until 50°C. In this version the compressor stage matching has been optimized for hot ambient climate by modifying the stationary blade rows, which increases the gas turbine power output considerably at high ambient temperatures.

Combustor
The SGT-800 combustor is shown schematically in Figure 2, where the main flow paths, the flame shape and the main combustion recirculation zones are highlighted. Also the naming convention used throughout this paper is shown. The annular combustor is serial convection-cooled and almost all combustor air is used for the combustion, resulting in nearly equal flame zone
temperature and Turbine Entrance Temperature (TET). This is an advantage for DLE combustion systems, where both excellent CO performance at turndown and low NOx despite relatively high TET are obtained. The annular concept also results in a relatively smooth tangential temperature distribution and reduced cooling surface area to combustor volume ratio compared to a can system. The combustor system was enhanced by passive acoustic damping during 2007 and an improved impingement cooling system of the front panel heat shield [4].

Figure 2: Cross section of SGT-800 annular combustor: 47.5MW design including main flow paths and recirculation zones (left) and up-rated design including naming convention (right).

**Burner modifications**
The SGT-800 uses 30 3rd generation DLE burners. A burner is shown in Figure 3a, where the sections for fuel transfer, swirl generator and mixing tube are indicated. The burner uses totally five fuel lines for dual fuel capability: main gas, pilot gas, central gas, main liquid and pilot liquid, where the respective injection locations are shown in the figure.

Figure 3a: SGT-800 burner showing the sections for (A) fuel transfer, (B) swirl generator and (C) mixing tube. The fuel injection locations are shown for (1) pilot gas, (2) central gas, (3) main gas, (4) pilot liquid and (5) main liquid.

3b: SGT-800 fuel line system showing the central, main and pilot gas valve locations.
The corresponding fuel line system is shown in Figure 3b, indicating the uncomplicated fuel system using small amount of fuel lines. Few fuel lines and continuous flow in all fuel pipes in the entire load range due to the absence of fuel staging are important features for stable DLE systems, allowing for example for rapid load changes.

Recent burner investigations of the combustion stability and emissions have given a strong basis for further improvement, together with the engine operation experience. A comprehensive validation test program using experiments have been carried out in four different configurations; water rig tests for investigating the fuel-air mixing, atmospheric single can combustion tests for fundamental investigations with high optical access, high-pressure single can combustion tests to include the pressure effect and full scale engine tests for the complete effect including neighboring burners. Some of these investigations are reported in [4-6]. Corresponding Reynolds Averaged Navier-Stokes (RANS) and Large Eddy Simulations (LES) simulations have been performed to further improve the understanding of the measurement results, see for example [7-8].

The investigations have resulted in a design update for both the swirl cone area for improved manufacturing tolerances and more aerodynamically shaped swirl generators, and the pilot system for improved pilot premixing, flash back limit and life. All these design modifications have resulted in improved combustion performance, including reduction of emissions and improvement of combustion stability and fuel flexibility capabilities.

**Combustion chamber modifications**

The up-rated combustion chamber compared to the 47.5MW design is shown in Figure 2. Effort has been performed to reduce the combustor system pressure drop, which due to the serial system equals the sum of the cooling and combustion passages. Here is described the design modifications that reduce the combustion chamber cooling system effective area and pressure drop: (i) soft wall holes and modified impingement cooling layout of the front panel heat shield as described in [4], (ii) modified liner cooling channel inlet outer bypass plates and (iii) smaller liner ribs.

The bypass plates were optimized using CFD to maintain the cooling properties with reduced pressure drop, leading to a 20% loss reduction. The liner rib efficiency was investigated using LES and RANS by Lörstad [9]. Since the channel pressure loss reduces much faster with rib size compared to the heat transfer coefficient, a significant reduction in pressure loss could be achieved with a rib size reduction without significantly affecting the cooling. The cooling layout of the exit rings is designed, apart from achieving sufficient life and minimum deformation with time of the combustor exit, to achieve sufficient life of the combustor-turbine interface. This required a modification due to the reduced number of turbine guide vanes as described in next section.

The cooling system improvements were performed in conjunction with life improvements. By improving the cooling in certain hot areas with small life margin and reducing the cooling in areas with larger life margin, the total life due to thermal bond coat flaking, material oxidation and creep, is improved despite reduced cooling system pressure drop. In addition, the cyclic life is largely dependent on the thermal load variation, where the difference in thermal expansion for the various parts due to different temperatures and material thermal expansion coefficients causes life limiting thermal stresses. Since the liners and the front panel heat shield are hotter than the hood parts, this causes a difference in thermal expansion. To reduce this tension the outer cooling struts are redesigned and the difference between the designs may be seen by comparing the different designs shown in Figure 2. By increasing the distance between the hot parts and the cooling struts and by optimizing the material thickness, the cyclic life was doubled.
**Turbine**

The turbine modifications can be categorized into changes in turbine stators, aerodynamic profiles of gas channel components and cooling design of hot components, as indicated in Figure 1. In the hot gas channel, stage 1 was redesigned significantly while stage 3 was slightly adjusted.

**Turbine stator modifications**

In the 47.5MW design, the vane carrier rings consist of a few segments. The segment configuration has a risk of cooling air leakage at the locations where supply cooling air for vanes and heat shields is distributed. In addition it is not convenient to assemble these segments during the service periods. In the new enhanced design these parts are replaced by complete rings which effectively seal the cooling air passages in both stage 1 and 2. Choosing this design gives several advantages: reduced component cost, cooling air saving, better control of the tip clearance, etc. In order to fit these changes to the stator, the stator ring in stage 1 was accordingly modified.

**Turbine aerodynamic modifications**

The major aerodynamic modifications were performed for vane 1, blade 1 and blade 3. In vane 1, the total number of vanes was reduced by 24% resulting in reduced wet area in the hot gas passage and reduced Trailing Edge (TE) blockage. This has contributed to the significant cooling air saving. In addition, the reduction in vane number reduces the service cost and overhaul time. A new and thicker airfoil (see Figure 4), together with contoured platforms, were designed which have increased the turbine loading by about 33%. The design phase started with profile design programs and then through-flow calculations. Finally, 3D CFD calculations were used to check and optimize the shape of the profile, the film holes positions on the airfoil and platforms and to minimize secondary losses.

![Figure 4: SGT-800 guide vane 1 of the 47.5MW (left) and 50.5MW (right) designs.](image)

For blade 1, the total number of blades is reduced by 9.2%. Similar as vane 1, this has contributed to the reduced cooling air consumption, improved aerodynamic performance and the component initial and service cost. The airfoil profile is virtually similar and the Mach number distribution is kept. This has made the cooling design easier, because the external heat transfer coefficient is essentially the same as for the 47.5MW design. However, small adjustments were done locally for the improvement of aerodynamic performance, e.g., the reduction of the recirculation zone on the pressure side, as shown in Figure 5.

For blade 3, the new airfoil was modified based on the airfoil of the 47.5MW design but enlarged by 9%. The TE shocks in the new airfoil were reduced and thereby the shock-losses were
decreased. The tip seal was also optimized so that the tip leakage was reduced. The swirl angle was kept low at the blade TE and the uneven distribution of stagnation pressure into the diffuser was corrected. In addition the cross section area was optimized in order to avoid increased centrifugal forces and stresses.

Figure 5: Recirculation zones on pressure side tip regions of blade 1 for 47.5MW (left) and 50.5MW (middle) and continuous matrix cooling at TE (right) (Siemens patent [10]).

To check the overall performance of the whole turbine, the final aerodynamic design was carried out by three dimensional CFD calculations. Fine tuning was performed based on the complete flow fields and local aerodynamic losses.

**Turbine cooling modifications**

Cooling modifications for the up-rated SGT-800 were carried out mainly for the cooling air reduction of vane 1 and blade 1, because of their new aerodynamic profiles. For vane 1, the up-rated airfoil cooling design is very much based on the 47.5MW version, but with 3 airfoil passages instead of 2 because of the bigger airfoil. The cooling of vane airfoil and platforms were modified and optimized with cooling design tools, and APS TBC is applied on both airfoils and platforms (TBC only on platforms in the 47.5MW design). All modifications including decreased guide vane numbers have reduced the total cooling flow consumption by 20%. For blade 1, the cooling modifications mainly consist of a new TE continuous matrix (patented [10], see Figure 5) and high-performance ribs. The use of EB-PVD TBC is maintained for blade 1. The result of the blade cooling modifications plus decreased blade number is reduced cooling mass flow by 9.5% in total.

In addition, the cooling design of vane 2 and heat shield 1 were also slightly modified. For vane 2, it is due to the use of TBC on all wet surfaces, while for heat shield 1, the total number of the components was changed. Although the total cooling air consumption for these two components remains almost constant, improved life is achieved in terms of both oxidation and cyclic performance.

**VALIDATION**

To ensure a reliable gas turbine performance, Siemens always tries to validate the design at both component level and system level. The tests at the component level were often carried out under cold/hot conditions before the test at the system level. This is to detect any issues at an early stage
and in case of insufficient performance; modifications can be introduced into the complete package as early as possible. Once the complete package of the gas turbine is ready, the engine is often tested regarding the performance of aerodynamics and cooling, emission and control ability at different loads, steady and transient conditions, and even with different fuels. The final adjustments of the design are implemented on the customer engines based on the results. Even if the results from these tests completely meet the design expectation, it lays out the foundation for future improvements. The SGT-800 development has followed this well-established approach.

**Component Level**

The major validation work at the component level was conducted for the turbine components vane 1 and blade 1. This is of course because the turbine first stage was re-designed in terms of aerodynamics, cooling and coating. For vane 1, the focus was given to the aerodynamic performance and the cooling design including the external heat transfer and film cooling. For blade 1, the focus was given to the patented trailing edge design and the overall cooling performance.

**Vane 1**

For vane 1, a test rig (shown in Figure 6) was developed to study the film cooling performance for this enhanced rating [11]. The experiments adopted a transient measurement technique by using IR camera to obtain the distribution of external heat transfer coefficient. In addition to the heat transfer test, the aerodynamic performance of the full-sized vane 1 was tested at an annular sector cascade test rig located at a research partner university [12].

![Figure 6: Film cooling test rig using IR-camera.](image)

**Blade 1**

For blade 1, the component test is aimed for the new design feature introduced, i.e., continuous matrix cooling, high performance ribs, etc. UpScaled plastic models of both double matrix and continuous matrix were manufactured, and then tested by transient liquid crystal technique, at a research partner university. A typical example of the recorded liquid crystal color is shown in Figure 7, and it indicates that the new patented design outperforms the 47.5MW double matrix design in terms of heat transfer enhancement in the critical region (note that roughly the green color means high heat transfer coefficient). In addition, the full sized blade 1 (in fact blade 2 also) was tested by a calorimetric testing using zinc bath to assess the performance of the newly introduced high performance rib design and continuous matrix at the trailing edge [13].

The above tests at the component level have contributed to a great understanding of vane 1 and blade 1 in terms of aerodynamic and cooling performances. Efforts were also made to look at the possibilities to further improve the performance. This is very beneficial for the technology development as well as the future product upgrade.
Figure 7: Local distribution visualization of the 50.5MW (left) and 47.5MW (right) matrix designs by liquid crystal.

**System Level**
The full engine test for design validation included the so-called crystal test mainly for the turbine section. Since the first engine test has to cover many different operations, it used the traditional measurement technique of thermocouples for metal temperature, i.e., no thermo crystals. However, it is not possible to install sufficient thermocouples to capture the thermal gradients of hot components. Therefore, the temperature-sensitive crystal technique [14] was used to map the detailed temperature distribution with good accuracy in the crystal test. In summary, the enhanced SGT-800 has followed the same validation approach as the one used when the machine was originally introduced.

The first engine test for the complete package is to test the overall gas turbine performance, compressor performance and blade dynamics, combustor stability/emissions/thermal conditions, turbine performance and thermal conditions, etc. The test was carried out in the spring 2010 with plenty of additional instrumentation in the engine. For the compressor, combustor and turbine sections, the instrumentation included about 260 pressure taps and 400 temperature sensors across different stages including the diffuser. While the results from the pressure measurements confirmed the expectation of turbine aerodynamic performance, the temperature measurement did indicate that heat shield 1 had over-heating in certain locations. Quick action was taken immediately, and the improved design was introduced to the customer engines afterwards.

Based on the test results, the performance of the enhanced gas turbine shown in Table 2 was confirmed.

<table>
<thead>
<tr>
<th></th>
<th>50.5</th>
<th>47.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>GT output, MW</td>
<td>50.5</td>
<td>47.5</td>
</tr>
<tr>
<td>Combined cycle output, MW</td>
<td>71.4</td>
<td>66.6</td>
</tr>
<tr>
<td>Thermal efficiency, SC, %</td>
<td>38.3</td>
<td>37.7</td>
</tr>
<tr>
<td>Thermal efficiency, CC, %</td>
<td>55.1</td>
<td>53.8</td>
</tr>
<tr>
<td>Compressor pressure ratio</td>
<td>20.8</td>
<td>20.2</td>
</tr>
<tr>
<td>Exhaust mass flow, kg/sec</td>
<td>134.2</td>
<td>132.8</td>
</tr>
<tr>
<td>ISO Turbine inlet temperature, ºC</td>
<td>1230</td>
<td>1200</td>
</tr>
<tr>
<td>Exhaust temperature, ºC</td>
<td>553</td>
<td>541</td>
</tr>
<tr>
<td>Emission NOx@15%O2, ppmv</td>
<td>≤15</td>
<td>≤15</td>
</tr>
<tr>
<td>Emission CO@15%O2, ppmv</td>
<td>≤5</td>
<td>≤5</td>
</tr>
</tbody>
</table>
Compressor
The overall compressor performance was determined by measuring temperature and pressure at the diffuser inlet at four different tangential positions. Nine probes where used at each rake to accurately capture the radial profiles. The test showed, as predicted by the numerical calculations, that the compressor efficiency had been improved by about 0.3% compared to the 47.5MW design.

To validate the new blade design the compressor was instrumented with static pressure taps after each stage. In addition, to verify the optimized front stage design both radial temperature and pressure profiles were measured at the leading edge of stator 3. The results were compared with both through flow and 3D CFD calculations over the entire compressor operating range to verify the overall stage matching and radial matching and performance of the front stages. Comparison with measurements from the 47.5MW design also helped to confirm the success of the enhanced design.

Compressor stability is crucial to ensure a stable engine operation of the entire operating range. To test the limits of the compressor several different surge tests were conducted. In one of these tests the compressor speed was decreased, with the variable guide vanes in a fully open position and the bleed valves closed, until the compressor surged. This confirmed the good stability and large surge margin of the compressor at hot ambient operation.

To detect tendencies of rotating stall during start up pressure transducers were mounted in the stator along the compressor. The variable guide vane schedule could therefore be optimized to ensure a stable compressor start up behavior.

Combustion stability and emissions
The validation of combustion stability and emissions are among the most crucial parts of an engine test, due to the lack of sufficiently accurate prediction methods for those parameters and since single burner combustion tests may fail to be representative for an annular combustor.

Figure 8 shows the emissions versus load for a typical engine test when using natural gas, where the Pilot-Fuel-Ratio (PFR) is the amount of pilot fuel. As shown in the figure, the NOx emissions are low down to 50% load using standard PFR settings, and hence there is good margin to the guarantee values. CO is below 2ppm down to 30% load which verifies the benefit of this type of combustor, i.e. relatively large combustor volume and hot liners without any cooling air entering through the liners helps to keep the CO emissions down. The un-burnt hydrocarbons (UHC) follow a similar trend as CO but the values are much lower.

![Figure 8: Engine test of measured emissions versus power and PFR for natural gas (left) and liquid fuel (right).](image-url)
For liquid fuel operation the CO and UHC values are similar as compared to natural gas down to about 30% load, as shown in Figure 8, but somewhat higher for lower load. The DLE NOx emissions from this typical engine test using E10 diesel (summer quality) are about 20ppm for standard PFR settings at high loads and about 25ppm down to 50% load, but could be reduced to about 20ppm for 40-100% load by PFR adjustment. Corresponding tests with E32 diesel (winter quality) gives similar results.

The SGT-800 combustor system has shown remarkable combustion stability and especially after the introduction of the soft wall [4]. The stability was excellent during the emission tests shown in Figure 8. A flame temperature dependence test is shown in Figure 9, where the primary zone temperature was reduced by ~10% at full load conditions, which confirms the excellent margin to blow out. The values are scaled with the standard full load conditions, corresponding to the 100% load point in Figure 8 (NOx ~10ppm). As can be seen in the figure, very low NOx values may be achieved by reducing the flame zone temperature. The load level reduces somewhat with flame zone temperature, which gives the possibility of offering significantly reduced NOx emissions at high part loads. The pressure is only slightly affected, showing that the NOx reduction is not due to pressure effects. The CO emissions were below 2ppm for the whole interval.

The excellent combustion stability and the stable combustion control system using few fuel lines and the lack of fuel staging, gives the possibility of handling rapid load changes as shown in Figure 10, both sudden load increase and load rejection. The load rejection was performed from full load to 0% load (idle conditions). Another example of operation flexibility is the capability of a 10 minutes start from a cold engine to 100% load as shown in Figure 10.

![Figure 9: Engine test of emissions versus flame zone temperature at full load conditions.](image)

![Figure 10: Operational flexibility example of stable load addition and load rejection (left) and capability of 10 minutes start to 100% load (right).](image)
**Turbine crystal test**

After the first engine test, the machine was disassembled and fully inspected. The thermo-crystal instrumented components were then assembled into the machine, prepared for the crystal test. For the turbine section, a total number of 1710 thermo-crystals were used, with 1528 for metal temperature, 176 for gas inlet temperature for both stationary vanes and rotating blades, and 6 for cooling air inlet temperature of rotating blades. In addition, a limited number of thermocouples and pressure taps were used, with a few thermocouples installed on typical positions to record the transient for estimating the equivalent hold time for the thermo-crystals. Thermal paint was also used in certain locations as complement. It is of great value to measure the gas inlet temperature profile by thermo-crystals, so that the CFD aerodynamic model can be calibrated by the measurement results. The boundary conditions provided by the calibrated CFD aerodynamic model were then used for the cooling model, and the final metal temperature of the cooling model was calibrated against the thermo-crystal test results. As an example, the temperature distribution of vane 1 and blade 1, which was calibrated by the thermo-crystal results, is shown in Figure 10. The dot points in the figure are the measurement points of the thermo-crystals. With the dense usage of thermo-crystals, the distribution of thermal gradients can be assured. This work has been done for all the components along the hot gas path in the turbine section. The results have not only confirmed the expected temperature levels of turbine hot components, they also give a strong basis for future upgrade, e.g., potential cooling air saving has been identified in several areas.

During the crystal test also the combustor was tested with thermal paint measurements to complement the numerous point wise thermo-couple temperature and pressure probe measurements, to verify the predicted temperature field and especially the local variations, identifying the hottest regions and to estimate the tangential variation. Outer and inner liners and exit rings, the burners and other regions were measured with thermal paint.

In summary, the test at the system level confirmed the expectation of the enhanced design in terms of increase in power output, efficiency, life, etc.

![Figure 10: Vane #1 and blade #1 metal temperature distribution calibrated by thermo-crystals (crystal positions are shown).](image)

**Widening Fuel Capability**

The commercial SGT-800 fuel capability for both the 47.5MW and 50.5MW versions is shown in Table 3. Recent testing have shown higher capabilities for certain fuels, see for example [6,15], and work is ongoing for further improvements.
Table 3: SGT-800 gas fuel capability.

<table>
<thead>
<tr>
<th>Gas Fuel Constituents</th>
<th>Max</th>
<th>Comment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane, CH(_4)</td>
<td>mole %(^{(1)})</td>
<td>100</td>
</tr>
<tr>
<td>Ethane, C(_2)H(_6)</td>
<td>mole %(^{(1)})</td>
<td>30 Under revision based on successful burner testing up to 100% ethane</td>
</tr>
<tr>
<td>Propane, C(_3)H(_8)</td>
<td>mole %(^{(1)})</td>
<td>30 &gt;95% propane can be accepted with reduced TET</td>
</tr>
<tr>
<td>Butanes and heavier alkanes, C(_4)+</td>
<td>mole %(^{(1)})</td>
<td>15(^{(3)})</td>
</tr>
<tr>
<td>Hydrogen and carbon monoxide, H(_2) + CO</td>
<td>mole %(^{(1)})</td>
<td>10 Under revision based on successful burner testing up to 32% hydrogen</td>
</tr>
<tr>
<td>Inerts, N(_2)/CO(_2)</td>
<td>mole %(^{(2)})</td>
<td>50/40(^{(4)})</td>
</tr>
<tr>
<td>Hydrogen sulfide, H(_2)S</td>
<td>mole %</td>
<td>Evaluated case by case</td>
</tr>
<tr>
<td>Others (aromatics, alkanes, alkenes, oxygen, etc)</td>
<td>mole %</td>
<td>Evaluated case by case</td>
</tr>
</tbody>
</table>

\(^{(1)}\) % of reactant species (i.e. inerts not included)
\(^{(2)}\) % of total (inerts + reactants)
\(^{(3)}\) May be restricted to lower levels depending on the other gas constituents
\(^{(4)}\) Fuels with high inert content may require a separate fuel for ignition and start-up

As shown in the table, the hydrogen content is increased due to recent work to improve the permitted limit for both the 47.5MW and 50.5MW designs [15]. High hydrogen content as compared to pure natural gas may cause the flame to shift position which may significantly reduce the life of the burner. Also the upstream fuel line system may be affected by hydrogen embrittlement. Due to such reasons Siemens performs detailed investigations to validate any adjustment of the fuel specification.

Figure 12 shows borescope probe photos of the flame during single burner combustion tests in an annular high pressure test rig [6], for 0-32% by volume content of hydrogen mixed into natural gas (NG). In the pictures shown the test burner was operated with a PFR of 6% and the other 17 burners in the annular combustor which used pure NG were operated at the stable PFR of 10%. During natural gas operation of the test burner yellow flames are seen and there are no clear pilot flames at the pilot exits, but rather the natural gas is “swept away” and burns further downstream. As hydrogen is fed into the test burner the pilot flames appear and the fuel burns closer to the fuel exit ports. Increasing hydrogen content from 12% by volume up to the 32% does not significantly change the flame position but the light intensity increase and the flames appear whiter.

During the tests the flame position shift is controlled by flashback thermocouple reading. If the flame moves significantly upstream into the burner, i.e. flashback, this temperature would immediately increase. In all tests the increase in hydrogen content was stopped when indications of flame shifting was seen. For the burner configuration in Figure 12 (Nr 1) this was observed for the main flame. Several burner configurations were tested and for the second configuration presented here the indication of flame shifting took place in the pilot. The tests also showed that a small load reduction allows higher hydrogen content to be used.
The emissions of NOx measured after the turbine, downstream the test burner, are shown in Figure 13 versus hydrogen content. The NOx emissions are normalized to a reference case of operation on pure natural gas. For the two burner configurations mentioned above, where the first one (shown in Figure 12) showed no influence of the changed fuel composition. For the second burner configuration there is a small increase in NOx with increasing hydrogen content. For burner configuration 1 this is in agreement with the visual observation that the main flame position is not significantly changing with hydrogen and therefore the combustion situation is not affected.

Emission of CO was also measured and the values were low, showing that the trends in Figure 8 are to be obtained when applied to an SGT-800 combustor. The combustion dynamics monitored in the burner as well as in the combustor did not show any change in peak frequency or dynamic pressure in any of the frequency intervals measured, rather the hydrogen seems to have a stabilizing effect on the flames.

The temperature measured for various positions in the tip of the burner is shown in Figure 13. These are related to the temperatures measured in an SGT-800 engine in September 2011. If the temperature reading in the annular high pressure test rig environment coincide with the SGT-800 engine then the normalized temperature is 1. As seen, the normalized temperature for operation on natural gas (black crosses) deviates up to 15% between the test rig and the engine. Introducing hydrogen into the burner (red squares) has no or little influence on the temperature distribution. A small increase in some positions is seen which is consistent with the observations in Figure 12. It is therefore concluded that the burner co-fired with this concentration of hydrogen may be operated within an acceptable range for burner tip lifing. Hence there is large margin to the specified hydrogen limit in Table 3.

![Figure 12: View over test burner through borescope camera during test with increasing content of hydrogen in the natural gas (0-32% by volume H2).](image)
Figure 13: Normalized NOx emissions versus hydrogen content in combustion tests (left) and measured temperature in burner tip in combustion tests with/without hydrogen (right).

OPERATION EXPERIENCE
The first SGT-800 with 50.5MW rating was successfully operated and tested during the spring 2010, in both the first engine test and the crystal test as described above. Since then, several machines with this enhanced rate have been running at different customer sites. The engines have delivered or even exceeded the expected power output and efficiency. Totally about 20 units of the new rating are in operation, under commissioning or in delivery phase. Up to March 2013, the fleet of this new rating has accumulated >42,000 EOH, and the fleet leader has accumulated >17,000 EOH.

A planned follow-up inspection was made after 10,000 EOH by using borescope for the hot section, and it showed that all compressor, combustion chamber and turbine parts, including the redesigned guide vane 1, blade 1 and guide vane 2, were in excellent condition. Some sample pictures taken during the borescope inspection are shown in Figures 14-15 for guide vane 1, blade 1, guide vane 2 and combustion chamber, respectively, and one can see that the TBC is still in very good condition. Note that the brown patches on the platforms are due to the local cooling air distribution, and these regions have relatively low metal temperatures. Therefore, the full life time of the design target is expected to be fulfilled.

Figure 14: Guide vane #1 (left) and blade #1 (right) after 10,000 EOH.
CONCLUSION
Siemens Energy Oil & Gas has successfully validated an enhanced SGT-800 gas turbine with a power rating of 50.5MW and obtained field experience since the introduction 2010. The design modifications, validation and operation experience are described in this paper.

The major design changes in the enhanced version are for the turbine the redesign of stage 1 vanes and blades in the hot gas path and for the combustor the improved cooling system and burner redesign resulting in enhanced fuel flexibility capability. To validate the design, extensive experimental work has been carried out to verify the aerodynamic and cooling performances of vane 1 and blade 1 at the component level. At the system level, the complete package was tested in the engine test program including the so-called ‘crystal test’ for both performance test and complete mapping of the thermal state of hot components. The program included tests for combustion performance and emissions for both gaseous and liquid fuels and the combustion stability has also been shown to successfully handle sudden load changes, including load rejection from full load to idle. In conjunction with the work of the new rating, the capability of gas fuel flexibility has been improved for both the 47.5MW and 50.5MW ratings including increased content of hydrogen, ethane, propane, butane and heavier alkanes as well as inert gases.

With these tests, the design targets were confirmed or even exceeded in some areas. In addition, the test results have provided a strong basis for future upgrades, for example potential turbine cooling air savings and further fuel capability improvements have been identified.

For the operation experience up to March 2013, the fleet leader has already accumulated >17,000 EOH. A planned follow up inspection was made after 10,000 EOH by borescope for the hot section, and it showed that the combustor and turbine were in good condition.

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