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Thesis for the degree of Philosophiae Doctor

Trondheim, October 2018

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Abstract

Background
Oil and gas production at several North Sea offshore installations is limited by the gas turbine power available, and any deterioration in gas turbine performance directly affects production rates. Fouling in the compressor section of the gas turbines is the main cause of performance deterioration, and the fouling is removed by offline water wash, where the compressor performance is restored back to the baseline level. An offline wash requires the engine to stop (downtime), and it is therefore crucial to have as lengthy intervals as practically possible between offline washes to enhance production efficiency and the economy of the plant. In order to achieve lengthy intervals between offline washes, the deterioration rate must be as low as possible, by implementing the optimum combination of intake air filtration system and online water wash of the gas turbine compressor.

Objectives
Analyse gas turbine compressor deterioration mechanisms and water washing fundamentals in order to develop, validate and implement an optimum water washing system for offshore applications. In addition, analyse the effect of air intake filter system deterioration in order to develop, validate and implement optimum air intake solutions.

Research approach, activities
The basis for this work has been empirical data from eleven LM2500PE gas turbines at a North Sea offshore field. Testing and validation of online water wash at increased water rates were performed on the subject engines, as well as air intake filtration upgrades by increasing filter class and improving the filter housing design features. The PhD project work has also included onshore testing of a gas turbine in deteriorated condition, as well as gas turbine intake air filter testing in a test cell facility. The field work on this subject has spanned more than ten years and was initiated by the author prior to the PhD project.

Conclusions, contributions
An effective water wash of the axial compressor starts with an offline wash system that is capable of fully restoring the performance and baseline of the engine, i.e. restoring the recoverable deterioration (fouling). The non-recoverable deterioration rate (mechanical wear, increased tip clearance, etc.) is marginal compared with the recoverable deterioration rate. A typical figure for the recoverable deterioration rate is a 4 % drop in compressor efficiency over a 4-month operation period between offline washes, whereas the typical non-recoverable deterioration rate is only 0.5 % over 3 - 4 years (25,000 operating hours between gas generator overhaul). The main finding in this work is that a daily online wash at high flow rate is the key parameter for high effectiveness and a low deterioration rate, with a recommended water-to-air ratio for the LM2500PE of between 1.2 to 1.6 % (by mass).

An effective air intake filtration system in an offshore environment consists of two main components. An upstream vane separator that knocks out the large droplets (coalescer) followed by a downstream filter section that stops the smaller droplets. In addition, other key elements for an effective filtration system are: drainage system, water/air locks, holding frame for filter elements, anti-icing system and access doors/windows for inspection and maintenance. The main challenge for an offshore filtration system is to handle water droplets with airborne salt particles. Other solid particles (soot, sand, etc.) are not the main challenge offshore. However, the filter system must occasionally handle such particles. In foggy/high humidity conditions, it is particularly challenging for the filtration system to operate effectively due to deterioration of the filter performance. This work has tested filters at different filter grades, both offshore and in an onshore test cell facility. The main findings are that increasing the filter grade from M6 to F7 has proven to be effective, combined with a shorter filter change interval, prior to the point where filter deterioration is severe. In addition, a CFD study of the air intake system has been performed, which resulted in flow optimization in the field.

On the whole, the results of the optimized air intake filtration and online water wash at high flow rate have given longer operation intervals between offline washes and lower deterioration rates. The present standard offline wash interval is extended to six months, compared with two months in the past. The compressor efficiency deterioration rate is reduced to just 0.5 % over each 6-month operation period. This has a major impact on the production rates, production efficiency and economy of the plant. It entails lower fuel consumption and exhaust emissions and hence lower costs for fuel, CO_2 and NO_x taxes and reduced environmental impact.

In addition, procedures and algorithms for gas turbine compressor efficiency correction and air flow analysis have been developed in the project.

The major contributions of this work are presented in six papers included in the appendices.

Proposed further work

The shift in performance curves on the compressor map from clean to deteriorated condition should be further studied, as well as the effect on velocity diagrams and Reynolds number. The sensitivity in engine parameters for compressor efficiency vs. air flow should be further studied.

Acknowledgments

This project is part of a long-term initiative by Statoil and NTNU for optimized gas turbine operations. The project is financed by Statoil. The close cooperation with the Statoil operation organization over a number of years has been the basis for moving forward and improving on gas turbine performance. In this project, I have had close cooperation with a large professional network in the oil & gas industry (operators, vendors and companies), and that network has grown during this project. The combined efforts of offshore field operation, offshore field testing and validation, and onshore testing in test cell facilities, have been the key elements to achieve the results which are presented in this thesis.

In particular, I would like to thank my advisor Professor Lars E. Bakken for all his support, knowledge and contributions. All the great professional advice, combined with our shared interest in motorcycles and racing, has kept my motivation high during the project.

Statoil ASA are acknowledged for their financial assistance and for all the support I have received over the years during the follow up of rotating equipment offshore. A special thanks to the skilled Statoil turbine technicians offshore. I would like to thank Atle Aadland and Kurt Risa in the Stavanger operational group for all their support. I would like to thank Morten Løes from R&D for his great support in the first part the project, prior to his retirement. I would like to thank Håvard Nordhus from R&D for valuable HYSYS support. I would like to thank my Statoil colleagues in the Gas Turbine Pool for support on engine shop inspections and incoming engine testing. I also have to thank Olaf Brekke as the co-supervisor in this project, and Arne Ulrik Bindingsbø from R&D.

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I would like to thank my fellow PhD students and colleagues for a great work environment at NTNU. A special thanks to masters student Lena Samnøy for contributions on the HYSYS analysis.

Finally, my family have been my greatest supporters throughout the PhD work. My wife Amy and my daughters Helena and Isabella are always the greatest motivation to keep on working. They all moved with me to Trondheim in the first phase of the project, they are always there during difficult periods, and we are now back in Stavanger to build our future. Seeing the children start school and follow their development is the greatest gift in life.

Overview of Publications

The research project work has resulted in the publications listed below.

I. *Gas Turbine Operation Offshore; On-line Compressor Wash Operational Experience*
Proceedings of ASME Turbo Expo 2014; Power for Land, Sea and Air
June 16-20, 2014, Düsseldorf, Germany
Paper no. GT2014-25272

II. *Gas Turbine Operation Offshore; Increased Operating Interval and Higher Engine Performance Through Optimized Intake Air Filter System*
Proceedings of ASME Turbo Expo 2016; Turbomachinery Technical Conference and Exposition
June 13-17, 2016, Seoul, South Korea
Paper no. GT2016-56066

III. *Gas Turbine Fouling Offshore; Correction Methodology Compressor Efficiency*
Proceedings of ASME Turbo Expo 2017; Turbomachinery Technical Conference and Exposition
June 26-30, 2017, Charlotte, North Carolina, USA
Paper no. GT2017-63025

IV. *Gas Turbine Fouling Offshore; Air Intake Filtration Optimization*
Proceedings of ASME Turbo Expo 2018; Turbomachinery Technical Conference and Exposition
June 11-15, 2018, Oslo, Norway
Paper no. GT2018-75613

V. *Gas Turbine Fouling Offshore; Effective Online Water Wash Through High Water-to-Air Ratio*
Proceedings of ASME Turbo Expo 2018; Turbomachinery Technical Conference and Exposition
June 11-15, 2018, Oslo, Norway
Paper no. GT2018-75618

VI. *Gas Turbine Fouling Offshore; An Analysis of Engine Air Flow*
Proceedings of ASME 2018 Power and Energy Conference
June 24-28, 2018, Lake Buena Vista, Florida, USA
Paper no. PowerEnergy2018-7269

In addition, conference papers I and V have been published in the journals which are listed below.

VII. *Gas Turbine Operation Offshore; Online Compressor Wash Operational Experience*
Journal of Mechanics Engineering and Automation, Volume 4, Number 12, December 2014
Paper no. 4-JMEA-E20140811-1

VIII. *Gas Turbine Fouling Offshore; Effective Online Water Wash Through High Water-to-Air Ratio*
ASME Journal of Engineering for Gas Turbines and Power
Paper no. GTP-18-1296

The papers are included in the appendices.

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Nomenclature

H	Head	[J/kg or m]
h	Mass specific Enthalpy	[J/kg]
N1	GG rotor speed	[rpm]
n_v	Polytropic Volume exponent	[-]
RH	Relative humidity	[%]
P	Pressure	[mbara or bara]
PS	Pressure static	[bara]
T	Temperature	[°C or K]
v	Specific Volume	[m3/kg]
w.a.r.	Water to Air Ratio	[% by mass]
η	Efficiency	[-]
κ	Real air heat capacity ratio (kappa)	[-]

Subscripts:

c	Corrected parameter
i	Isentropic
p	Polytropic
1	Inlet conditions
2	Discharge condition

LM2500 Gas turbine station numbers:

0	Ambient condition
1	Intake plenum condition
2	HPC inlet condition
3	HPC discharge condition
5.4	LPT inlet condition
8	LPT discharge condition

Acronyms:

ASME	American Society of Mechanical Engineers
CDP	Compressor discharge pressure
CFD	Computational fluid dynamics
DLE	Dry low emission
EOS	Equation of state
EPA	Efficient particulate air filter
ESN	Engine serial number
GE	General Electric
GG	Gas generator
HEPA	High efficiency particulate air filter
HPC	High pressure compressor
HPT	High pressure turbine
ISO	International Organization for Standardization
LPT	Low pressure turbine

NGTE	National Gas Turbine Establishment
OEM	Original equipment manufacturer
PLC	Programmable logic controller
PS	Pressure side
PT	Power turbine
RR	Rolls-Royce
SAC	Standard annular combustor
SS	Suction side
ULPA	Ultra low particle air filter
VSV	Variable stator vanes
WHRU	Waste heat recovery unit

1. Introduction

Gas turbines play an important role as the main power source on offshore oil and gas installations. The high power to weight ratio is favourable in typical offshore applications, for example power generation and compressor stations. Other types of power sources, such as diesel engines which are dominant in the shipping industry, are not an option for offshore applications in most cases. In recent years, due to increased environmental requirements, onshore electrical power through sea cables has become an alternative power source on the Norwegian continental shelf on installations where this is technically and economically feasible.

For the majority of offshore installations, the gas turbine is the primary power source. Roughly 50 % of the gas turbines installed on the Norwegian continental shelf have a waste heat recovery unit to utilize the exhaust heat and increase the total plant efficiency. A high tax levy on both CO2 and NOx in Norway encourages the operators to work extensively on energy optimization. In most cases, a gas turbine package optimization yields both a plant production increase, an increase in efficiency, and a reduction in tax expenses due to the lowering of the specific fuel consumption.

Gas turbines used in oil & gas production often operate at full power and without redundancy, hence production rates are directly affected by any deterioration and engine downtime, which again has a huge impact on plant performance and availability. As an example of economy potential for one of Statoil's North Sea gas/condensate field analyzed in this study [1], an annual production gain of some 50 MSm3 of gas and some 400,000 barrels of condensate can be achieved by increasing gas turbine maintenance stop intervals from four to six months.

The focus in this project has been to study the air intake filter system and water wash system in order to find the best possible combination of offline/online water wash and air filtration system for optimum gas turbine performance. A good intake air filter system design process includes a trade-off between the inlet area, number of stages, total differential pressure, filtration effectiveness, maintainability and filter deterioration rate with the aim of obtaining the best possible gas turbine performance with low deterioration rate. A good water wash system design process includes an offline water wash system that is fully capable of restoring the baseline performance in combination with daily online water wash to reduce the deterioration rate between the offline washes. Water wash is also a trade-off between offline wash intervals, the frequency/effectiveness of online wash and acceptable engine performance. The target is for the intervals between offline washes to be as long as possible to ensure high engine availability. However, there are practical limits to consider for reliable engine operation.

The following sub-sections describe the LM2500 gas turbine, the challenges of offshore gas turbine operation, the research objectives/approach and the author's contribution to science.

1.1 Historical evolution gas turbines offshore applications

Industrial gas turbines, which were first demonstrated by Ægidius Elling in 1903 [2], are the main power source on offshore oil and gas plants. However, the biggest market for gas turbines is aircrafts. Gas turbines are also used in marine shipping applications for power generation or propulsion, industrial applications like power generator plants, combined cycle plants and compressor stations.

When the oil and gas industry developed rapidly on the Norwegian continental shelf in the 1970s, gas turbines were identified early on as the preferred primary power source. The high power to weight ratio, combined with easy access to fuel gas from the production wells, gave the gas turbine a dominant role in this type of application. The LM2500 gas turbine from General Electric (GE) was introduced in the 1970s. This is a 2-shaft aeroderivative type gas turbine, meaning it is a flight engine design (high power to weight ratio) adopted for industrial application, with a free power turbine driving the load. The LM2500 series is based on the TF39 military engine developed in the 1960s by GE. The “LM” designation refers to “land and marine” and the “2500” designation indicates 25000 hp shaft power in the first version of the engine.

Main development of the LM2500 series gas turbine:

- First built for military TF39 turbofan engine used in C-5 Galaxy aircraft in 1964.
- Developed for commercial CF6-6 turbofan engine used in McDonnell Douglas DC-10 aircraft.
- Industrial & Marine version LM2500 introduced in early 1970s.

Today, there are roughly 200 gas turbines operating on the Norwegian continental shelf. The LM series gas turbine is the leading type of engine in use, with Statoil being the leading operator of this engine type with a fleet of more than 100 engines. The LM2500 series has 23 MW shaft power in the “base” configuration, 30 MW in the “plus” configuration and 34 MW in the “plus G4” configuration. It also comes in “SAC” and “DLE” configurations, meaning standard and dry-low-emission combustion system, as well as “gas only” and “dual-fuel (gas/diesel)” configurations. Statoil also operates a few LM1600 and LM6000 engines, although the LM2500 engine is the dominant engine model. Other gas turbine manufacturers whose products are used on the Norwegian Continental Shelf are Rolls-Royce, Siemens and Kongsberg.

On the LM2500 type gas turbine, the two main sections of the engine are the Gas Generator (GG) and the Power Turbine (PT or LPT), as shown in Figure 1. The GG consists of a 16-stage axial compressor, a combustion chamber and a 2-stage axial high-pressure-turbine, whereas the PT is a 6-stage axial low-pressure-turbine driving the load (generator or compressor).

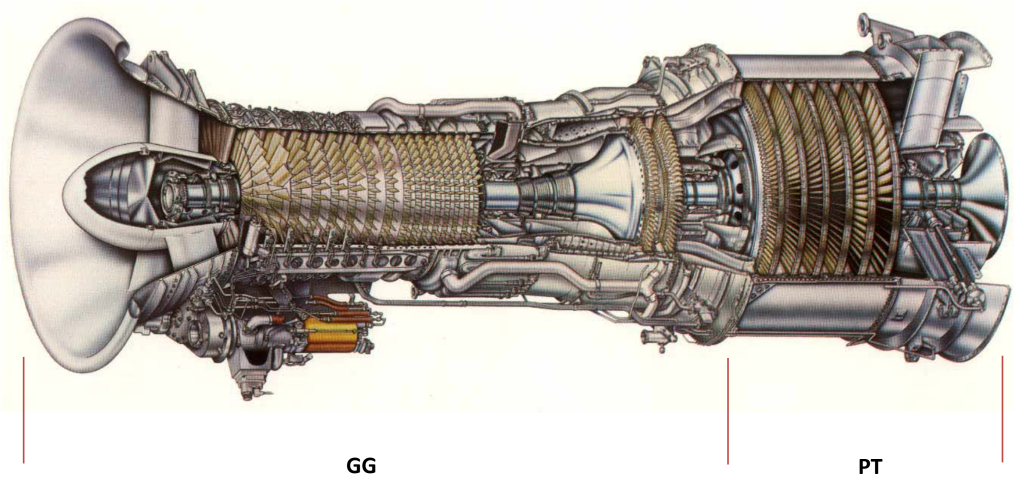

Figure 1. LM2500PE gas turbine from General Electric

The GG and PT are coupled together aerodynamically, although not mechanically since the two independent shafts run at different speeds. The GG and PT are built as modules to allow for easy change in conjunction with replacement/repair. The normal schedule for overhaul (workshop visit) is 25,000 hours for the GG and 75,000 hours for the PT. Moreover, a specific GG (ESN) has a schedule of every other workshop visit for Hot Section Repair (HSR) and Overhaul (OH). A HSR repair focuses on the combustion chamber and HPT sections with the critical components exposed to high temperatures, but a HSR does not cover a full repair of the compressor section, which is performed every 50,000 hours on OH.

Overview of the main gas turbine types operated on the Norwegian Continental Shelf:

- GE LM2500 base SAC
- GE LM2500 plus SAC
- GE LM2500 base DLE
- GE LM2500 plus DLE
- GE LM1600 SAC
- GE LM1600 DLE
- GE LM6000 SAC
- GE LM6000 DLE
- RR RB211-24G/GT
- RR Avon 200

The “LM2500 base SAC” engine in the “PE” version (with GE “6-pack” PT) is the most commonly used engine type. It is normally operated at gas-only fuel for compressor train applications, and dual-fuel (gas/diesel) for generator drive applications.

1.2 Challenges gas turbine operation offshore

All gas turbines experience a loss of performance over time. The performance loss is often referred to as “recoverable” and “non-recoverable” deterioration. Recoverable deterioration is mainly fouling in the compressor section, which is removed by water wash. Non-recoverable deterioration is mainly mechanical wear on engine components (e.g. increased tip clearance), but could also be burning damage in the combustion chamber/HPT turbine, which is (normally) not recoverable until the next engine exchange. The GG performance after a HSR is normally slightly lower than after an OH. This is illustrated in Figure 2.

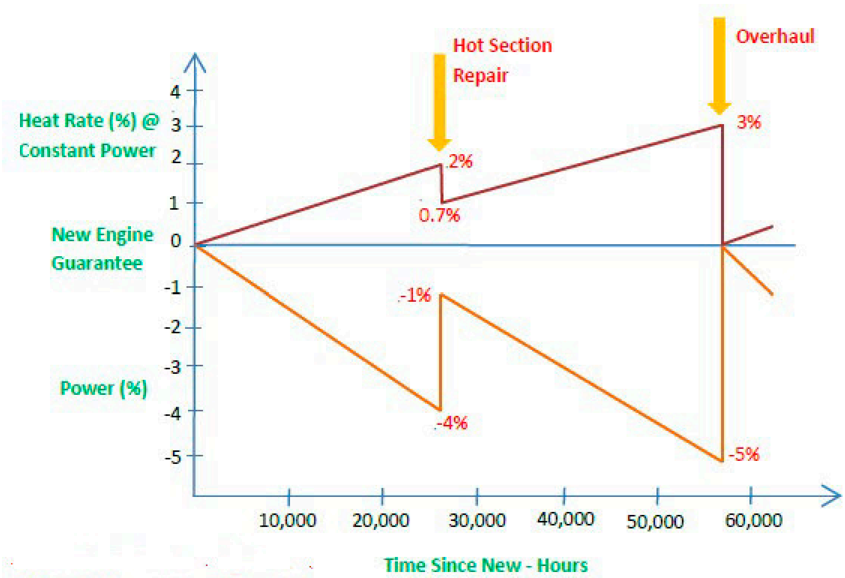

Figure 2. Gas turbine non-recoverable performance deterioration [3]

Figure 2 describes the long-term non-recoverable deterioration which is mainly mechanical wear. Changing major engine components/modules in the field is seldom performed, since it is more economical to have a pool of spare engines for quick field exchange. The vast majority of total gas turbine deterioration is related to the GG, where fouling in the compressor section is the main issue. The performance loss (non-recoverable) over the lifetime of a GG is illustrated in Figure 2, where the losses are not fully recovered until a full overhaul (OH) of the GG. In this illustration from the literature, a 1 % power loss is used as an example of difference in performance after an engine change for HSR vs. OH. Deterioration of the PT is negligible compared with the GG, and is thus not a focus in the current study. For typical North Sea gas turbine applications, the fuel used is light HC gas of good quality (minor operation time on diesel fuel on dual-fuel applications) and turbine fouling is therefore virtually non-existent. The focus area is thus recoverable deterioration and, specifically, compressor fouling.

1.2.1 Compressor deterioration

The main challenge offshore is the short-term, recoverable deterioration/compressor fouling caused by salt. Water droplets with airborne salt particles enter the intake filter system and some of the salt particles penetrate the filter system and cause compressor fouling by adhering to the compressor airfoils. Other solid particles (soot, sand, etc.) are not the main challenge offshore. However, the filter system must handle such particles occasionally to prevent engine deterioration. The recoverable deterioration is described in Figure 3, where typical operating intervals between offline wash (often combined with other maintenance) are in the range of 2,000–4,000 running hours. After an offline wash, the recoverable deterioration and performance loss are restored, indicated by the dotted lines at the left-hand side of Figure 3. By adding online wash (daily), the deterioration rate is reduced, as indicated on the right-hand side of Figure 3. A reduced deterioration rate implies higher average power available and/or improved efficiency, and can also be utilized to extend the operating intervals between offline washes, which improves the production efficiency of the plant.

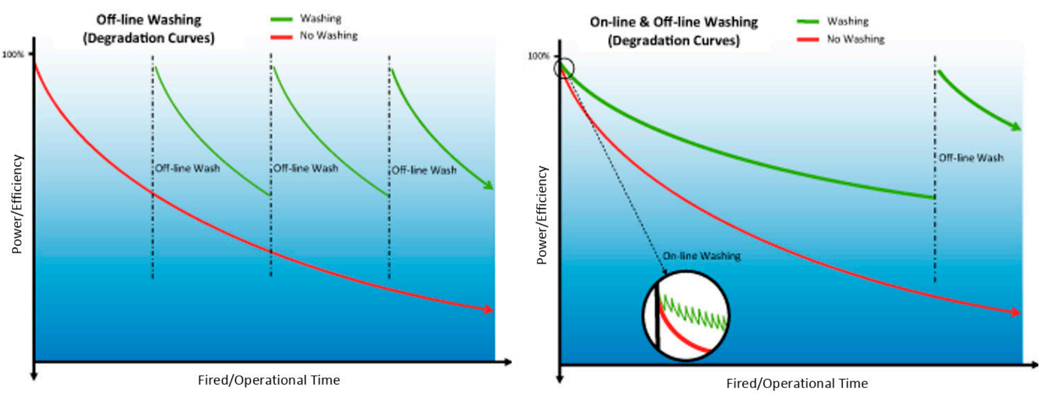

Figure 3. Gas turbine recoverable performance deterioration [4]

1.2.2 Instrumentation and condition monitoring

Instrumentation on the majority of offshore installations is often limited to the necessary instruments for machine control/protection and additional instruments for effective performance monitoring and analysis are often not available or, if installed, are less accurate.

LM2500 SAC engines are quite flexible with regard to fuel flow/composition. These values are typically programmed once in the fuel controller during commissioning and there is thus no requirement for measuring online gas-flow for fiscal quality, nor is there a need for a gas chromotograph/calorimeter to gauge online fuel-gas composition (as is required for DLE engines). Inlet depression, as well as inlet plenum temperature, pressure and humidity, are other typical parameters which are not available. Thus, inlet air flow calculation implies inaccuracy, which, in the next phase, implies inaccuracy for compressor head and efficiency calculations. In order to achieve more accurate air flow and compressor head/efficiency calculations, it is recommended to have absolute pressure transmitters for P2 (mbara) and PS3 (bara), with high resolution and as short impulse tubing lines as possible (in addition to inlet depression measurement).

Figures 4 and 5 show typical plots from condition monitoring systems used for offshore applications. These examples have compressor efficiency presented, based on an EOS. However, not all offshore applications have compressor efficiency included in the condition monitoring system. As can be seen from these trend data, the deterioration rate between offline washes is very difficult to analyze, due to data shatter in every direction. Often the offline wash effect is difficult or impossible to detect from these data, and the efficiency trend can sometimes apparently increase in the middle of an operating period, which is obviously not possible. The data shatter and inconsistent trend data are typical due to:

- Instrumentation accuracy
- Efficiency not corrected for changes in load and ambient conditions
- Periods of open compressor bleed (if used)
- Poor effectiveness of offline wash system

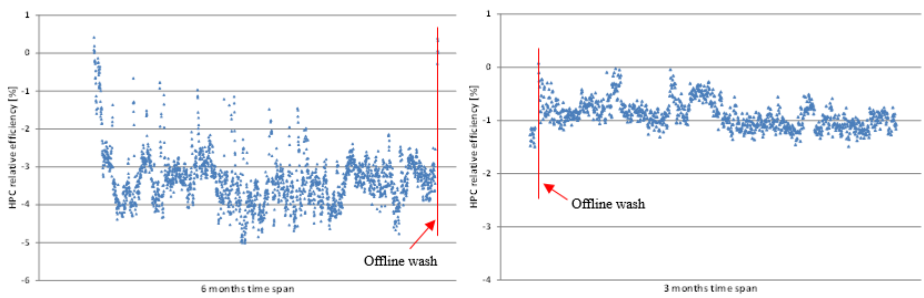

Figure 4. Compressor efficiency trend data [5]

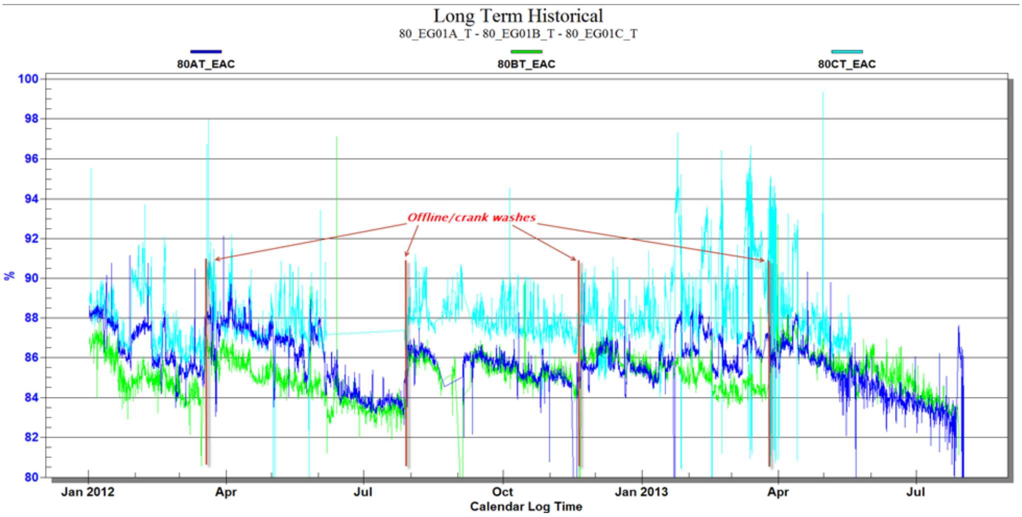

Figure 5. Compressor efficiency trend data [Statoil operating data]

Compressor bleed

Engines that use variable compressor bleed air are particularly difficult to analyze in the periods when the bleed is open. Variable bleed is typically used either for anti-icing heating (SAC engines) or emission/staging control (DLE engines). A compressor efficiency correction approach for open bleed is challenging to develop and is not included in the current work.

1.2.3 Intake filters

A gas turbine air intake filtration system typically consists of a multi-stage system with the following components:

- Upstream louvre/multihood/trash-screen
- Anti-icing manifold for icing protection
- Filter stage (one or more stages)
- Vane separator (either upstream or downstream of the filter stage)
- Silencer ducting
- Inlet plenum (with noise isolation)

If the filtration system has too many components, it yields higher differential pressure and hence lower initial gas turbine performance. This effect is illustrated in the enthalpy-entropy diagram in Figure 6, where a higher inlet loss due to high filter differential pressure has a similar effect to an inlet throttle valve. The starting point for the compression (1) process moves to the right, indicated by the red arrow, which reduces the power and efficiency of the gas turbine.

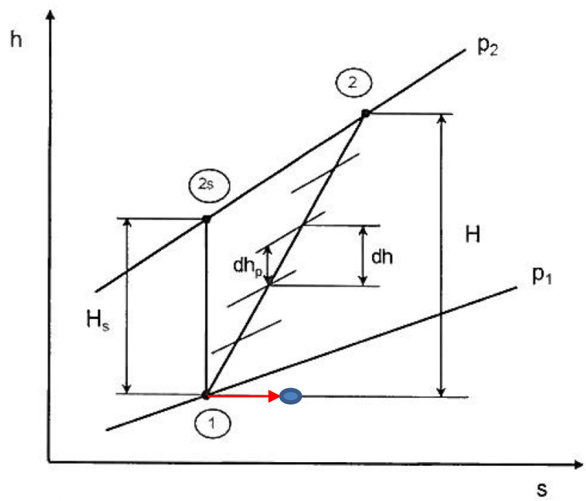

Figure 6. Compressor process - inlet loss

The number of stages, filter type and filter system configuration must be thoroughly analyzed in each project, relative to location, ambient conditions and gas turbine performance requirements. The total differential pressure must be evaluated versus the filtration effectiveness and deteriorated rate over time. Thus, a good intake air filter system design process includes a trade-off between the inlet area (which yields the design velocity), number of stages, total differential pressure, filtration effectiveness, maintainability and filter deterioration rate. Knowledge of filter deterioration over time at the subject location of the gas turbine is mandatory when selecting the operating interval between filter changes. Finally, the overall target is to obtain the best possible gas turbine performance with low compressor deterioration rate.

1.3 Research objectives

The PhD project's principal objective is: To analyse deterioration mechanisms, water washing and air intake filter fundamentals in order to develop, validate and implement optimum systems for offshore applications.

- Establish gas turbine models to analyse: performance parameter sensitivity, deterioration mechanism and effects related to different operating conditions. This includes new parameters for correcting, transposing and validating deteriorated performance.
- Analyse the phenomenological effects and establish optimum parameters for effective compressor cleaning. This includes optimum water-to-air ratio, droplet size, liquid flow rate duration time and fouling redeposit. Engine control modes and instrumentation accuracy are important parameters.
- Analyse, develop and implement water wash systems for optimum gas turbine operation. Focus is given to optimum operating interval and equivalent running hours.
- Analyse, develop and implement air intake filter systems for optimum gas turbine operation. Focus is given to the selection of optimum filter element type and filter change interval.

Limitations

It will not be a part of the scope of the PhD to evaluate the design of the LM2500(+) and LM6000 gas turbines, nor the design of water wash and air intake filter system components. There is a plan in place to cooperate with OEMs and other relevant equipment suppliers and specialist vendors for these issues. The scope of the PhD will be limited to the analysis and understanding of the phenomenological effects related to deterioration and different cleaning processes to ensure optimum gas turbine operation.

1.4 Research approach

The following methodology is used in the PhD project:

- Collect and evaluate operational data and experience from gas turbine operation at selected Statoil offshore installations.

- Performance parameter sensitivity, deterioration mechanisms and effects related to different operating conditions. This includes new parameters for correcting, transposing and validating deteriorated performance.

- Testing, operation and validation of online water wash system with high water flow rates on operating gas turbines offshore.

- Experimental gas turbine testing in OEM test cell. Incoming engine testing for performance mapping and improved deteriorated data.

- Air flow pilot instrumentation for inlet depression installed offshore on two engines. Cooperation with package OEM regarding air flow deterioration analysis.

- Air intake filter analysis:

    - Test of new air intake filter types on operating gas turbines offshore.

    - Collect and evaluate operational data and experience from intake air filtration systems at selected Statoil offshore installations.

    - Cooperation with filter OEM and a vendor with test cell facilities for inlet air filters. Testing of used filters from offshore operation and comparison with new filters with different filter grades and intervals.

    - Cooperation with a specialist vendor for CFD analysis on intake air filter system.

- Evaluate performance and results for best possible solutions of water wash and air intake filter systems on gas turbines offshore.

1.5 Author's contribution to science

The author has contributed the following to science through full scale validation on 23 MW gas turbines operated offshore:

- Online water wash optimized at high water-to-air ratio, above the OEM recommendations.
- Air intake filtration optimization through development of new filter type for offshore gas turbine applications.
- Compressor efficiency correction algorithm development, improved condition monitoring parameter.
- Compressor air flow, field implementation. Validation of methodology, improved condition monitoring parameter.

2. Operation of Aeroderivative Gas Turbines

This chapter describes the LM2500 gas turbine, the basic design features, the engine characteristics, performance and deterioration mechanisms.

2.1 Definitions

The LM2500 has the following station numbers / terminology for the monitored parameters, shown in Figure 7:

- T2 Compressor Inlet Temperature (or CIT)
- P2 Compressor Inlet Total Pressure
- T3 Compressor Discharge Temperature (or CDT)
- PS3 Compressor Discharge Static Pressure (or CDP)
- T5.4 (4.8) Power Turbine Inlet Temperature (or TIT)
- P5.4 (4.8) Power Turbine Inlet Pressure
- T8 Power Turbine Discharge Temperature

Forward direction is the air intake side, i.e. left-hand side of Figure 7. Aft side is the exhaust side, i.e. right-hand side of Figure 7.

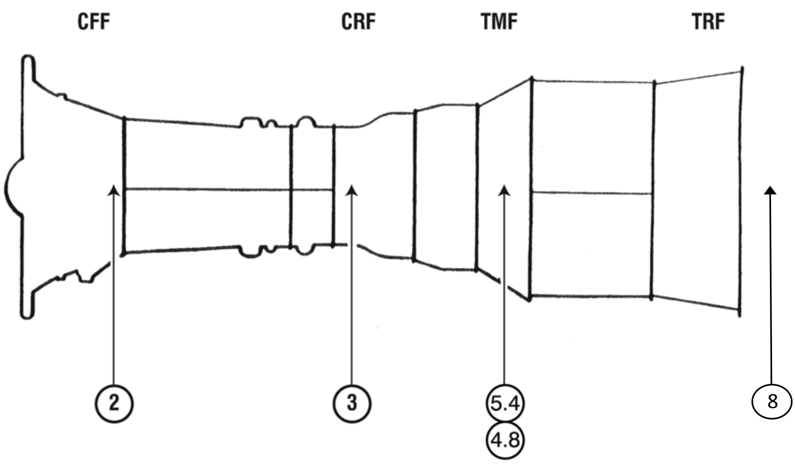

Figure 7. LM2500 station numbers and frames

The LM2500 engine has four frames as shown in Figure 7:

- Compressor Front Frame (CFF)
- Compressor Rear Frame (CRF)
- Turbine Mid Frame (TMF) – Aft flange of TMF is the connection between GG and PT
- Turbine Rear Frame (TRF)

2.2 HPC compressor

The forward section of the GG is the high-pressure-compressor (HPC shown in Figure 8) of the gas turbine, where air is compressed up to the required level for a given engine load. The HPC has a pressure ratio of 18:1 at baseload / ISO conditions. The compressor is driven by the HPT turbine.

The axial compressor of the LM2500:

- *16 compressor stages*
- *IGV*
- *6 VSV*
- *c_a remains constant*

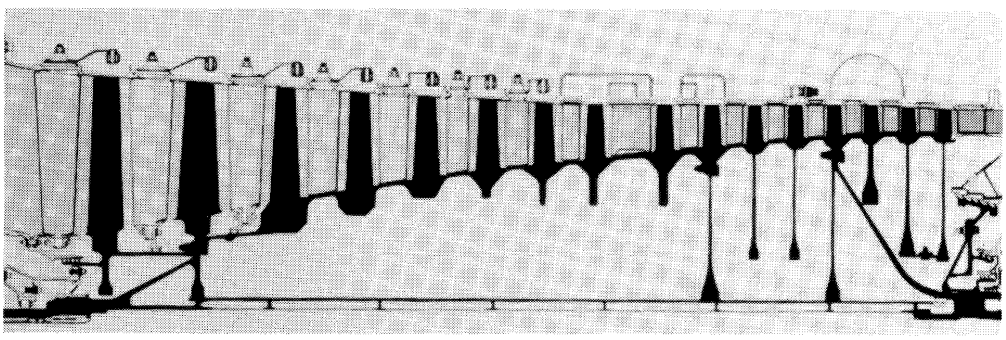

Figure 8. LM2500 16 stage axial compressor [6]

The Inlet Guide Vanes (IGVs) on the HPC compressor are next to six stages of vanes, the Variable Stator Vanes, or VSVs. These vanes change their angular pitch in response to a change in compressor inlet temperature or a change in gas generator speed. The purpose of this is to provide stall-free operation of the compressor through-out a wide range of speed and inlet temperatures. The VSVs are illustrated in Figure 8.

2.2.1 Gas turbine theory axial compressor

Gas turbine theory and simulation software tools are extensively used for both design and performance calculations (power and efficiency). This section describes the basis for compressor stage velocity triangles, the power and efficiency equations commonly used, and the performance characteristics of the compressor, shown in Figures 9 and 10.

Stage velocity triangles axial compressor:

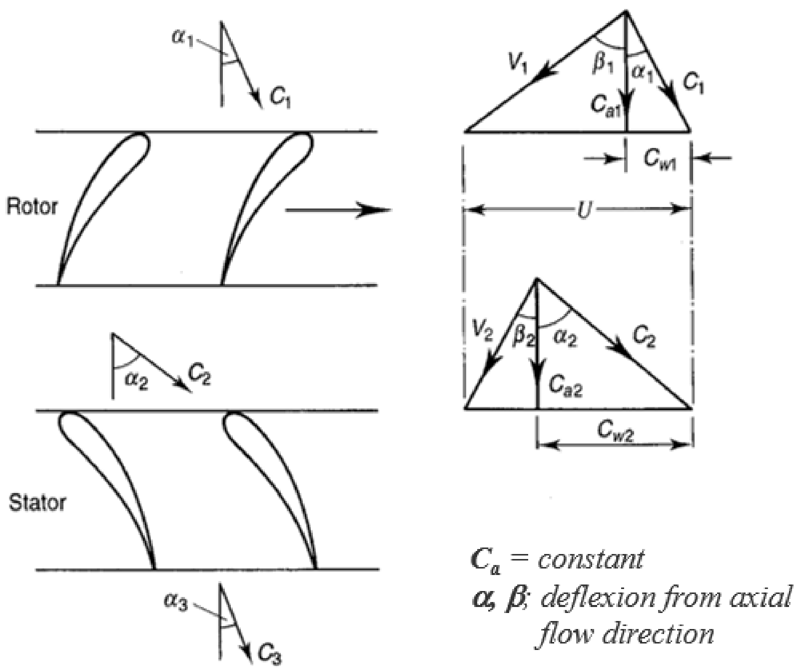

Figure 9. Axial compressor stage [6]

Power input *(thermal);*

$W = m\, c_p\, (T_{02} - T_{01})$

Related to stage velocity triangles;

$W = m\, u\, (c_{w2} - c_{w1})$

$W = m\, u\, c_a\, (\tan b_1 - \tan b_2)$

Power input;

Absorbed input energy which:
- raises fluid pressure
- overcomes frictional losses

Giving a rise in fluid stagnation temperature for the stage:

$DT_{0s} = T_{03} - T_{01} = T_{02} - T_{01} = (u\, c_a\, (\tan b_1 - \tan b_2)) / c_p$

Utilizing **isentropic efficiency**:

$h_s = (T'_{03} - T_{01}) / (T_{03} - T_{01})$

The stage **pressure ratio** becomes:

$$\frac{p_{03}}{p_{01}} = \left[1 + \frac{\eta_S \Delta T_{0S}}{T_{01}}\right]^{\gamma/(\gamma-1)}$$

Performance characteristics

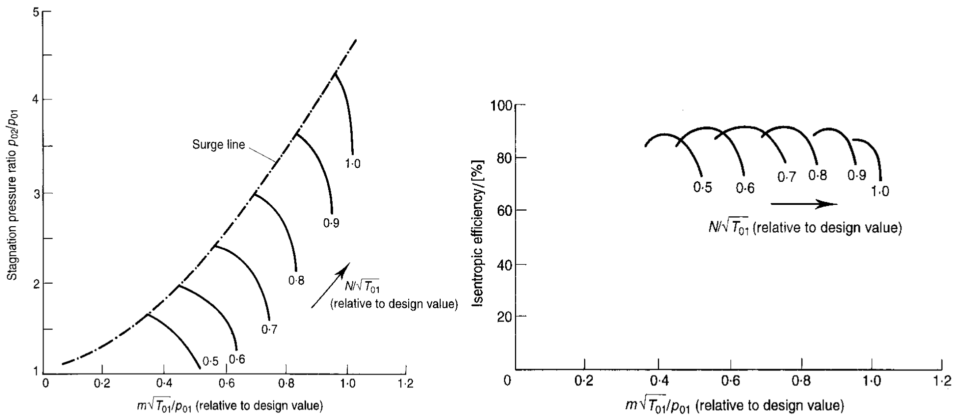

Figure 10. Axial compressor characteristics [6]

2.2.2 Compressor map

The full compressor map for the LM2500 engine is not available due to OEM confidentiality. An available map from the literature [7] has been used as a basis, with additional adjustments of the curves from available field data. The assumed/predicted compressor map is shown in Figure 11.

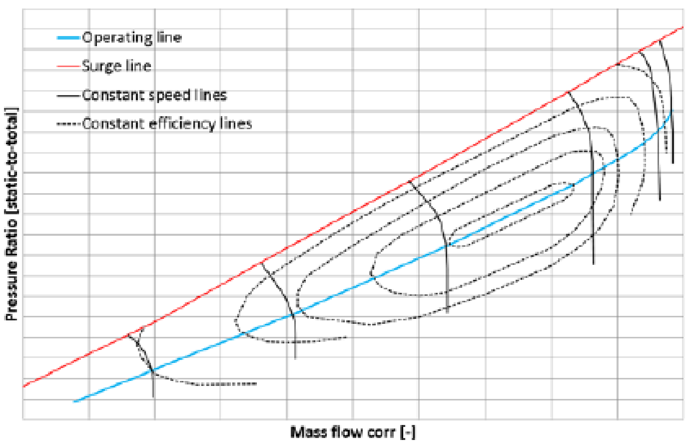

Figure 11. Compressor map [8]

The map is presented with the traditional axis of pressure ratio (static-to-total) vs. mass flow (corrected), with constant speed lines, constant efficiency lines, expected surge line and operating line, all indicated in the same map. Increased speed/load is up to the right of the map.

The constant speed lines (corrected speed N1c) become steeper at higher load, and the constant efficiency lines denote the highest efficiency at the innermost circle.

2.2.3 Performance and deterioration mechanisms

The historical compressor efficiency degradation on the subject engines (prior to modification of filter and water wash system in the 1990s), was in the range of 1.0 - 1.2 % per month of operation [9], as illustrated in Figure 12. This yields an equivalent of 6.0 - 6.5 % over a 6-month period between each offline wash/maintenance stop. However, six months between offline washes was not possible to achieve in the 1990s and 2000s due to a rapid fouling rate. Four months was the maximum operating interval until recent years [1].

As illustrated in Figure 12, the improvement in the deterioration rate after the modification/optimization of the water wash system and filter system has been significant, with a figure of 0.5 % degradation today. It is estimated that, of the total performance (efficiency) gain of 6 %, the contribution is equally shared between improved filter and online wash system, i.e. 3 % respectively. To obtain this level of performance, the water-to- air ratio has been operated at 1.4 % (by mass) during the online wash sequence, significantly above the OEM limit.

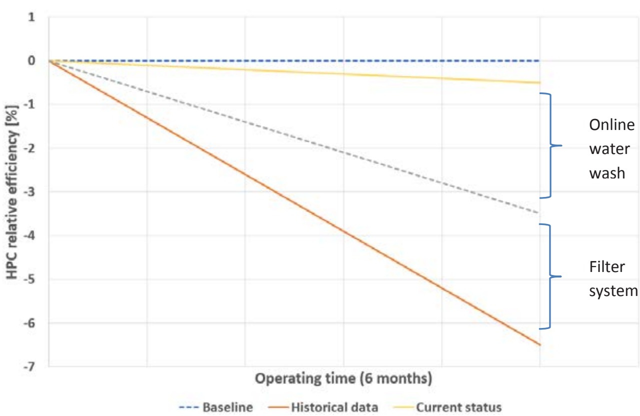

Figure 12. Typical figures of HPC degradation [9]

Field particle analysis

Particles have been collected from the floor of the filter housing in the plenum area during maintenance stops. In order to obtain an understanding of what type of particles penetrate the filters, it is assumed that the particles in the plenum area are representative of the particles following the air stream to the engine inlet and causing compressor deterioration.

In the plenum area, two main components are found: NaCl (salt) and Al (aluminum). Aluminum is likely to originate from structure corrosion at the silencer section of the intake system.

Salt remains the main component from the air stream penetrating the filters and, as proven by earlier research [10], is the main contributor to compressor deterioration.

Shift in performance in fouled condition

Fouling in the compressor results in higher friction losses and incident losses, and can lead to compressor stall.

The main observation from engine workshop visit inspections after offshore operation is that stage 1 rotor blade has some deposits on the suction side, most dominant at the LE. The PS of stage 1 rotor blade is clean, as well as the clean airfoils downstream of stage 1. This is typically the case (prior to offline wash) when the engine is operated at the best filter configuration and highest online water wash flow rate.

Thus, the main challenge is the stage 1 rotor blade suction side deposit. The assumed shift in velocity triangles is illustrated in Figure 13, where the dotted lines represent fouled condition. The velocity vectors are shifted to the right and down, i.e. axial velocity (mean) is also reduced in fouled condition.

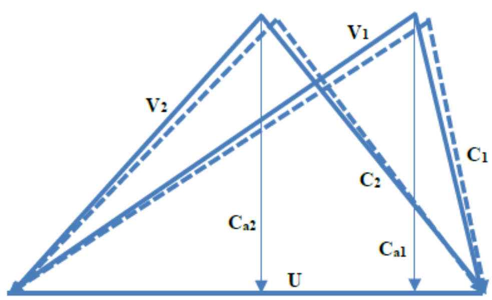

Figure 13. Velocity triangle, performance shift [9]

The observed deposit on stage 1 SS of the airfoils is illustrated in Figure 14, with the red line on the suction side of the airfoil. Thus, the velocity profile on the suction side will change, which results in performance loss.

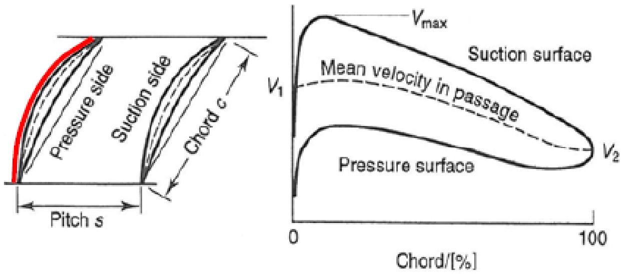

Figure 14. Velocity profile, performance shift [9]

The adverse pressure gradient and increased boundary layer due to annulus fouling have an impact on the velocity profile through the compressor. The shift in velocity profile is illustrated in Figure 15. As flow area between root and tip decreases, the axial velocity profile through the stages becomes more sharpened and stage energy transfer is reduced. The shift in velocity profile and related reduction in energy transfer are linked to the work-done mechanism (ref. “work-done factor” or “blockage factor”).

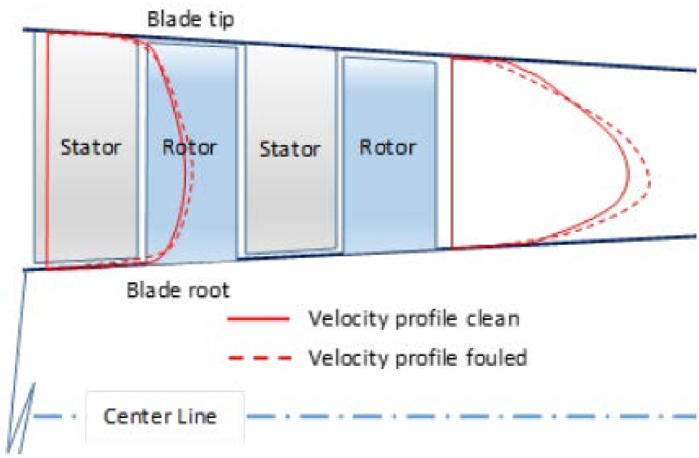

Figure 15. Velocity profile, performance shift [9]

2.3 HPT/LPT turbines

The HPT section of the LM2500 engine is a 2-stage axial turbine consisting of four main components:

- Stage 1 nozzles
- Stage 1 rotor
- Stage 2 nozzles
- Stage 2 rotor

These components are exposed to high temperatures, and advanced materials are used to be able to operate with high TIT and hence high engine performance and efficiency. These components are the most expensive on the engine. However, if the fuel gas quality is good and the engine is operated at gas only, there is seldom a high degree of deterioration on the HPT section.

The T5.4 temperature probes are located downstream of the HPT section at the TMF, where the T5.4 maximum control limit is the limiting factor for the gas turbine power output to protect the materials in the HPT section.

The LPT turbine (or PT) is a 6-stage axial turbine ("6-pack") to drive the load. The PT operates at 3,600 rpm for 60 Hz generator applications, and variable speed (normal range 3,000-3,800 rpm) for compressor applications where a speed-increasing gearbox is used to adapt to the required process gas compressor speed (normal range 5,000-15,000 rpm).

2.4 Instrumentation - condition monitoring

Most offshore gas turbine packages have the minimum amount of instrumentation necessary for engine control and protection. Additional instrumentation for effective condition monitoring is typically not available.

On a LM2500PE gas turbine, which is a twin-shaft GT with a free power turbine, it is very challenging to perform accurate performance and deterioration analysis due to the fact that the engine adjusts itself to any change in ambient condition or deteriorated condition. The GG reaches an equilibrium setting (output gas power vs. HPT/HPC power balance) based on the power and speed requirement from the PT. In addition, each individual GGs has slightly different performance characteristics in both new and deteriorated condition. Thus, many engine parameters are inter-connected, which makes condition monitoring challenging.

The standard LM2500 instrumentation related to compressor analysis is:

- Compressor inlet total pressure P2 [mbara]
- Compressor inlet temperature T2 [degrees C]
- Compressor discharge static pressure PS3 [bara]
- Compressor discharge temperature T3 [degrees C]
- Compressor speed N1 [rpm]
- VSV position [degrees]
- PT inlet temperature T5.4 [degrees C]

Such standard instrumentation for performance analysis and condition monitoring, often results in poor and inaccurate performance analysis. Typically, P2 and PS3 have insufficient resolution and response due to long impulse tubing lines.

The following instrumentation has been upgraded (new transmitter, shorter impulse tubing line) for improved accuracy:

- Compressor inlet total pressure P2 [mbara]
- Compressor discharge static pressure PS3 [bara]

In addition, the following instrumentation has been added for improved accuracy and air flow computation:

- Ambient pressure P0 [mbara]
- Ambient temperature T0 [degrees C]
- Ambient humidity RH0 [%]
- Inlet plenum temperature T1 [degrees C]
- Inlet plenum humidity RH1 [%]
- Compressor inlet static pressure PS2 [mbara]

After these upgrades, the GT condition monitoring system attains improved accuracy. However, for effective HPC efficiency analysis, a correction approach is also necessary to obtain reasonable results when analyzing deterioration trend data due to fouling. This is an approach that corrects and transposes the efficiency to a parameter where operating points can be compared over time independent of ambient conditions and engine load. This correction approach is described in the next chapter. In addition, air flow deterioration, including a correction approach, is an important parameter to add to the analysis, which is covered in chapter 6.

3. Performance analysis - simulations

The main performance parameter used on LM2500 engines to describe the baseline performance and deterioration rate is the compressor efficiency. The GT thermal efficiency is sometimes also used, but this parameter requires accurate fuel gas composition and flow measurements. This chapter focuses on the compressor efficiency.

The performance evaluation of the gas turbine compressor section is based on the following equations:

Polytropic exponent:

$$n_{\mathrm{v}} = \frac{\ln\left(\frac{\mathrm{p}_2}{\mathrm{p}_1}\right)}{\ln\left(\frac{\mathrm{v}_1}{\mathrm{v}_2}\right)}$$

Polytropic head:

$$H_{\mathrm{p}} = \frac{n_{\mathrm{v}}}{n_{\mathrm{v}} - 1}\left[\mathrm{p}_2 v_2 - \mathrm{p}_1 v_1\right]$$

Total head:

$$H = h_2 - h_1$$

Polytropic efficiency:

$$\eta_{\mathrm{p}} = \frac{H_p}{H}$$

The given polytropic approach is valid and is also utilized during wet operating conditions, e.g. following the wet performance variation during a water wash sequence or “continuous” water injection/fogging to improve power outage for a given timeframe. If required, a similar approach is used for isentropic analysis. The efficiency trends analyzed are based on relative change from a baseline, since the absolute value can vary from one engine to another and is thus not of significant interest. The baseline is defined as a clean engine after offline/crank wash and is set at 0 %.

3.1 HYSYS model compressor

A process simulation model of the LM2500 compressor is established in Aspen HYSYS as illustrated in Figure 16, with a humid air inlet to simulate real inlet air conditions offshore, as well as added water for online washing. The process simulation analysis is performed to compute HPC efficiency and validate the efficiency computed by the field condition monitoring system.

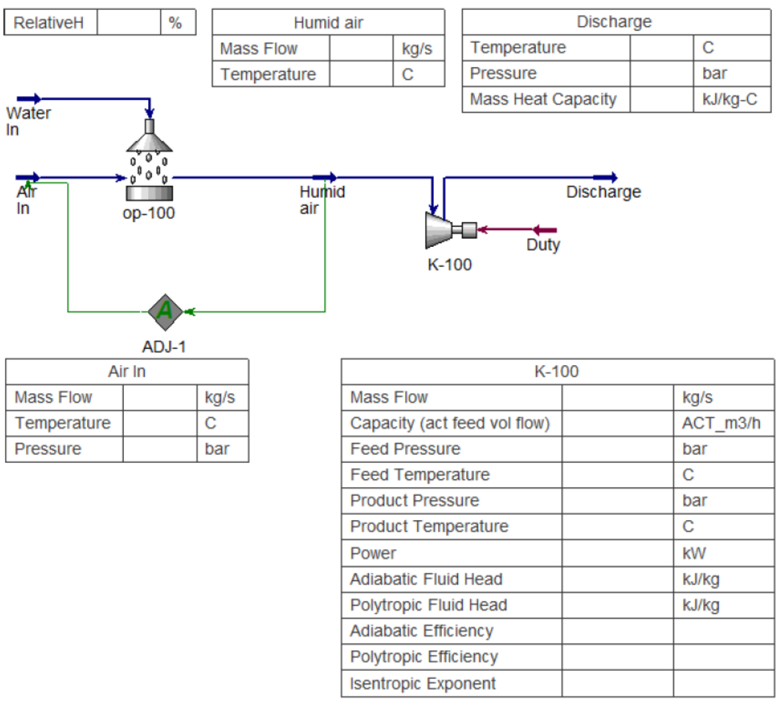

Figure 16. HYSYS model compressor

In addition, the simulation is performed to validate the compressor response in order to compute the response when the water-to-air ratio is further increased (up to 3 %, future target based on previous research [11]) and inlet conditions vary (T2, P2, RH). The analysis is even more important for DLE-engines, with a more sophisticated fuel and control system, thus risking disturbance of the burning-modes/staging during water ingestion.

Empirical cases are simulated in steady state condition in order to compare results of how the HPC discharge condition is affected during water ingestion. The EOS (equation of state) used is the SRK (Soavo-Redlich-Kwong), with further real air heat capacity ratio (κ) and correction factors derived using the ASME and Schultz methods [12]. Large empirical datasets are processed in Aspen Simulation Workbook for computation of HPC efficiency.

The main effect on compressor efficiency for reduction in single parameter values is as follows:

- Reduced inlet temperature: lower efficiency
- Reduced inlet pressure: higher efficiency
- Reduced inlet humidity (RH): higher efficiency
- Reduced discharge temperature: higher efficiency
- Reduced discharge pressure: lower efficiency

3.2 Efficiency correction algorithm

The correction methodology in the current study is based on the development of baseline compressor efficiency curves and to correct HPC efficiency to the baseline curve. The focus in this study has been to use N1c (corrected GG speed) vs. HPC polytropic efficiency, since there is a strong correlation between these parameters. Definition of N1c:

$$N_{1C} = \frac{N_1}{\sqrt{\frac{(T_2 + 273.15)}{288.15}}}$$

(T_2 in degrees C and speed in rpm)

This definition of N1c disregards inlet pressure (P2). However, for industrial gas turbines operated at sea level, the common industry practice is not to take pressure into account for corrected GG speed. N1c is further represented as a normalized value, i.e. 1.0 is the base load speed at ISO inlet conditions (15 degrees C, 60 % RH). When HPC efficiency is corrected vs. N1c, both the GG speed and inlet temperature T2 are taken into account, which yields a good representation of the engine load and operating condition.

Other parameters such as inlet air humidity and compressor pressure ratio have also been evaluated, although these parameters are not seen as important factors in this context. The correction method is performed through the following steps:

1. Establish a baseline curve for engines in new/overhauled condition.
2. Validate the shape of the baseline curve vs data in deteriorated condition.
3. Construct an artificial efficiency line (horizontal) at the top of the baseline curve (HPC eff. = 0) i.e. at BEP (best efficiency point).
4. Represent the distance between the baseline curve and the artificial horizontal line using an algorithm, i.e. a polynomial as a function of N1c.
5. Calculate the correction factor, i.e. use the polynomial function on all engine data points (actual N1c).
6. Apply the correction factor to the calculated HPC efficiency points.

To find the most suitable algorithm, a trial and error approach has been applied. The testing of various polynomial functions (2^{nd} to 6^{th} order) on operating data sets has been performed, in particular data sets that have a significant change in N1c over the operating period between offline washes. By applying the appropriate algorithm/correction methodology, all HPC efficiency data in the time domain can be compared as absolute values, i.e. attain a corrected efficiency parameter independent of where the engine is located on the operating map. Thus, the efficiency data can be transposed to useful deterioration trends.

Figure 17 shows the calculated efficiency points for four different engines in new/overhauled condition. The focus is given to the N1c range of 0.80 to 1.05, where most operating data falls within, which is why other part-load points are not shown in the figure. An average/representative baseline curve is constructed where most efficiency points match the curve shape.

Some engines are above the general baseline curve, but the shape of these points follows the baseline curve quite well, which is the most important criterion in the current methodology. Above this range of N1c, the efficiency varies by 5 %.

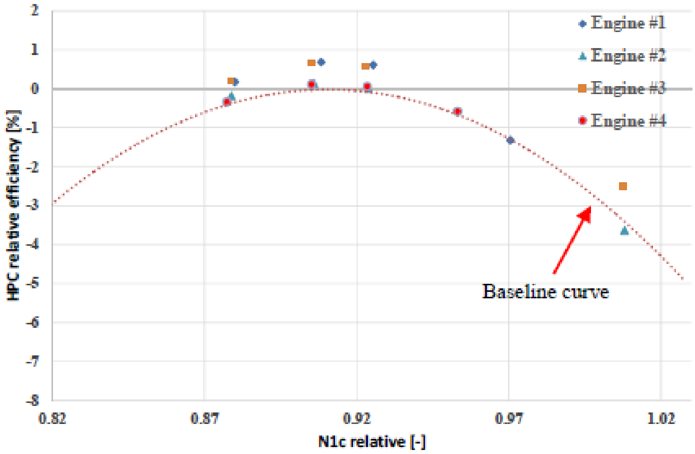

Figure 17. Baseline curve efficiency [5]

The next step is to validate the shape of the baseline curve vs. data in deteriorated condition. This is shown in Figure 18, where the blue line is the efficiency when an engine is due for overhaul (after 25,000 operating hours), and the green line is for the same engine after a "simplified" offline water wash (use of high pressure hose). As can be seen from the figure, the efficiency line is 2 - 2.5 % below the baseline curve, and is lifted ~1 % after the offline wash. When performing a "full" offline wash, the efficiency gain for this engine will typically be 2 %. This yields very good consistency of the baseline curve for this engine type.

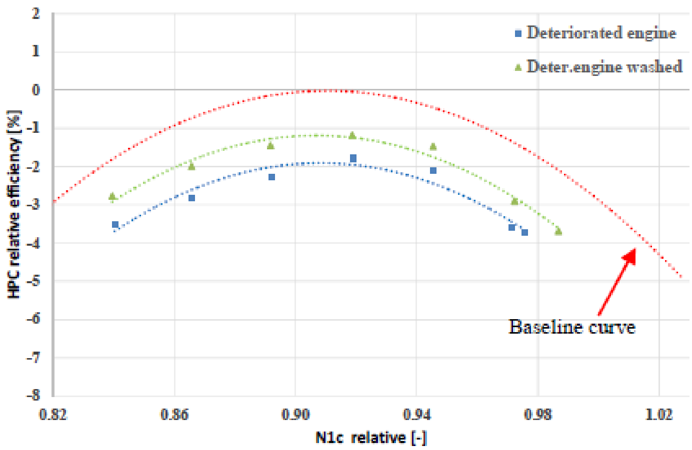

Figure 18. Validation baseline curve vs. deteriorated engine [5]

3.3 Correction results

A case of corrected GG speed vs. HPC efficiency is shown in Figure 19 for a 6-month operating period. It is evident from this trend that there is a strong correlation between N1c and HPC efficiency.

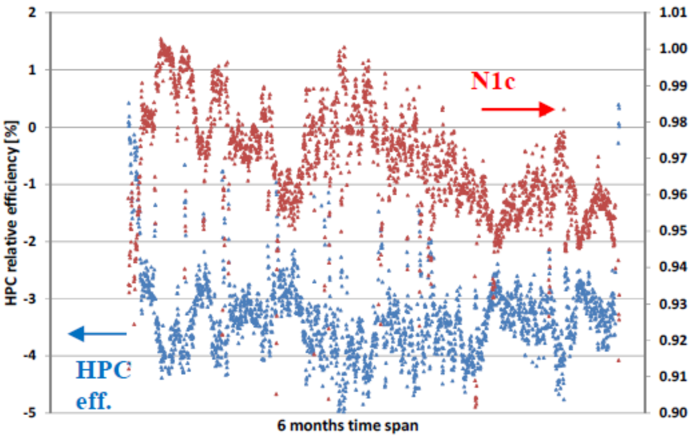

Figure 19. Correlation HPC efficiency and corr.speed [5]

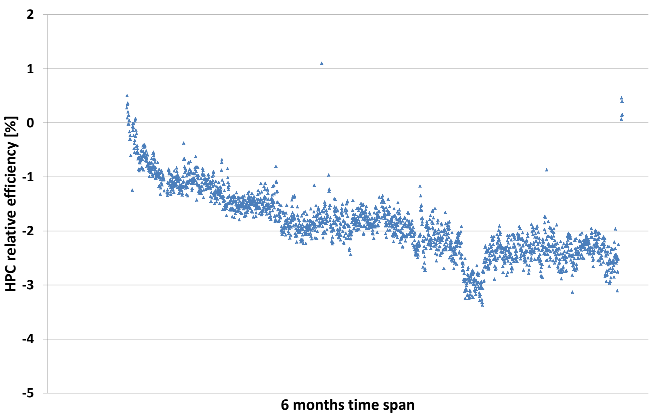

Figure 20. Corrected HPC efficiency [5]

The corrected GG speed oscillates substantially, with an average decline from ~1.0 to ~0.95. The HPC efficiency is, on average, almost flat. The operating point is drifting towards a lower N1c over time. Based on the shape of the baseline curve for this engine type from Figure 17 and 18 it is evident that a lower N1c on this section of the baseline curve yields a higher initial HPC efficiency. The real HPC efficiency deterioration is not possible to detect from Figure 18, since the real deterioration is counterbalanced by a lower N1c, and thus the HPC efficiency curve appears flat.

After applying the correction factor to this dataset, the HPC corrected efficiency is presented in Figure 20. Now the degradation is clearly visible with a near linear smooth deterioration of 2 - 2.5 % over a 6-month operating period. The efficiency dip in Figure 20 is due to a period of heavy fog, which affects the performance of the intake air filters and, ultimately, the HPC efficiency.

Another case of corrected GG speed vs. HPC efficiency is shown in Figure 21 for two operating periods of three months each. The operating point is drifting towards a lower N1c over time for each operating period. A lower N1c on this section of the baseline curve yields a higher initial HPC efficiency. It is not possible to detect the real HPC efficiency deterioration from Figure 21, since the real deterioration is counterbalanced by a lower N1c, and thus the HPC efficiency curve appears flat.

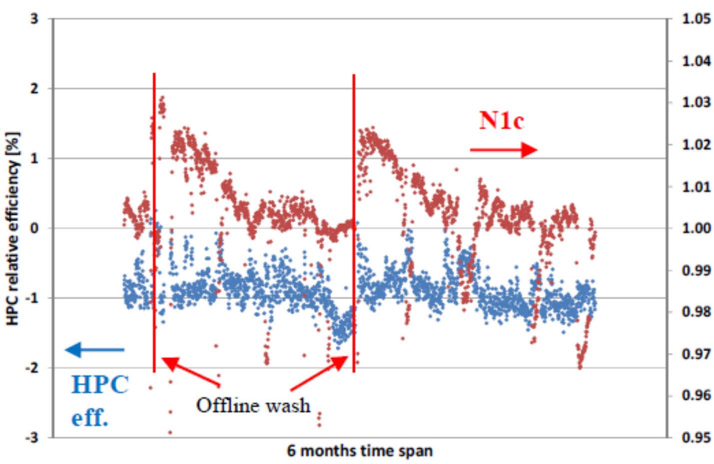

Figure 21. Correlation HPC efficiency and corrected speed [5]

After applying the correction factor to this dataset, the HPC corrected efficiency is presented in Figure 22. Now the degradation is clearly visible with a near linear smooth deterioration of 1 % over each 3-month operating period.

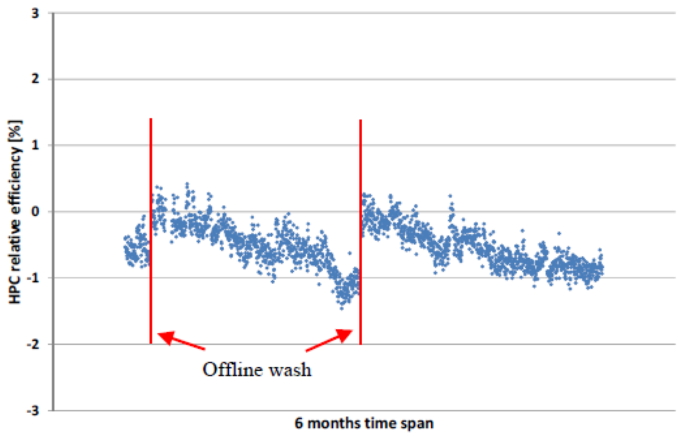

Figure 22. Corrected HPC efficiency [5]

4. Water Wash of Gas Turbines

4.1 Definitions

Water wash system

Water supply is taken from platform fresh water distribution, which is produced from seawater by evaporators. Furthermore, the water is led through a set of DI-filters (de-ionization or de-mineralization) and a final particle filter before entering the water wash skid, in order to achieve the OEM water quality specification for online water wash. The layout is shown in Figure 23 below.

The water wash skid has two tanks (each with a 200 litre capacity), one tank for clean water (used for online wash and offline cleaning after crank soak wash) and one tank for detergent/water mix (only used for offline wash). Thus, the risk of contamination of the clean water tank is avoided. The water is further pre-heated to 60 degrees Celsius, before ingestion to the engine from the pressure outlet connection to the water wash nozzles assembled at the engine's Bellmouth. One water wash skid is typically used for several engines in the same module; which is why a sequence of logic for solenoid valves, heaters and pump is programmed into the unit's PLC. The unit is operated semi-automatically (local manual triggering of the sequence), in order to minimize the time consumption for platform operators to operate the system. A fully integrated system is also possible with the platform process control system, i.e. one which enables remote system operation from the platform control room, or by a programmed timer function. But a fully integrated system adds complexity to the system, as well as the risk of malfunction, and the operators lose hands-on operation of the system.

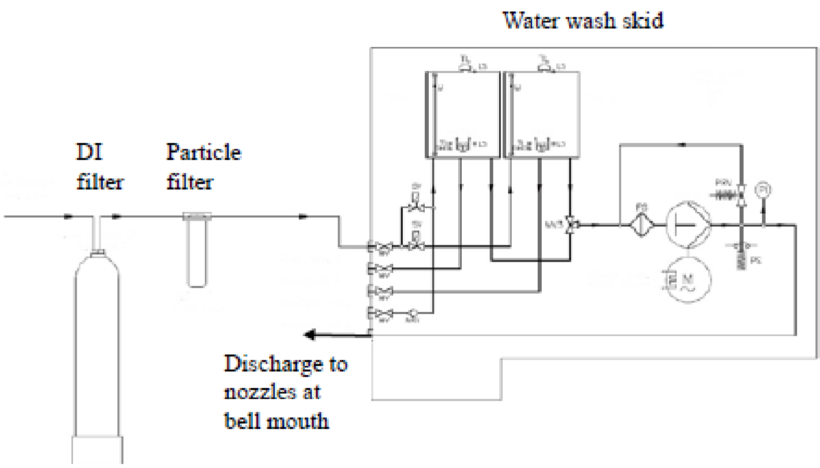

Figure 23. Water wash skid with inlet filters [1]

The nozzles are located at the Bellmouth as shown in Figure 24, mounted in pairs of two nozzles (10 nozzle configuration) between each strut at the CFF. This is done to prevent the water spray from hitting the struts, and the spray angle is selected to have full coverage around the circumference at the engine inlet area. The spray is aimed at the IGV hub (in offline wash mode), however, the spray will be somewhat bent towards the mid-section of the IGV in online wash mode due to the forces from the air flow at full engine load.

Figure 24. Bellmouth with water wash nozzles [9]

Offline wash

Offline water wash is defined as washing the compressor section on crank / motoring when the engine is stopped. The normal method of performing an offline wash consist of the following steps:

- One 3-minute motoring sequence by injecting a water / detergent mixture.
- Stop motoring for about 10 minutes, in order to make the water/detergent mixture rest on the rotor/static/airfoil surfaces.

- Two 3-minute motoring sequences by injecting water only.
- One 3-minute motoring sequence without water injection for drying / purging.

During offline wash, it is important that the low point pressure transmitter impulse tubings are disconnected, to avoid liquid accumulation in the lines and possible incorrect readings and start-up issues. The lines for PS3 and P5.4 are normally the most important to disconnect. It is obviously important to re-connect all these lines after the offline wash is finished, as well as being careful not to damage any fittings/connections during this operation. To achieve an effective offline water wash, the key element is high pressure/high temperature injection of detergent/water with open VSV at low-speed crank motoring (typically 1,200 – 1,500 rpm). An efficient detergent solution must be used and mixed correctly with water to be effective. The liquid flow rate is of less importance for the effectiveness of the offline wash.

Online wash

Online water wash is defined as washing the compressor section on full engine power (actual operational load). Only clean water is used for online washing. The key element for effective online washing is high pressure/high temperature injection of water at high flow rate. In addition, proper water spray coverage of the axial compressor inlet area (avoid struts), spray direction towards the IGV hub and de-ionized water treatment have been proven to be effective. Online water wash has been operated daily at full engine load, with a flow rate of three times the maximum permitted flow rate stipulated by the OEM without any negative findings for the engine during operation, maintenance or overhaul.

Idle wash

Idle water wash is defined as washing the compressor section on core idle engine speed, typically 5,000 – 6,800 rpm GG-speed for the LM2500. Idle washing implies downtime not only during the wash sequence itself, but also for a time-consuming period during the increasing and decreasing of the load on the train and production wells. Depending on the train/process/well configuration, this may take several hours, especially on compressor trains. However, for installations that run frequent offline wash intervals and/or have redundancy, idle washing may represent a preferable and economical washing regime in addition to offline washing. Statoil has some operational experience of idle washing from test periods offshore, with typical intervals of 1,000 running hours (offline/crank-wash interval 3,000 hours). These tests have demonstrated the potential for power recovery since the water-to-air ratio is high (fixed water flow rate). However, due to the disadvantage of engine downtime, idle washing has not been taken further to offshore operational routines in Statoil, and has thus not been included in this PhD work.

4.2 Online wash, offshore field testing, engine response

Three sets of water rates have been used, which are listed in Table 1. Starting at the 17 l/min rate some 10 years ago, the water rate was gradually increased to the present level of 50 l/min based on operating experience. For each water rate, the pump rate is slightly higher than the actual rate given by the nozzle capacity, and the extra water is recycled to the tank via a combined PRV/PSV (pressure regulator/safety valve). The corresponding water-to-air ratio (by mass fraction) for each flow configuration is consequently a function of the engine load, since the water rate is fixed. For example, for the 50 l/min water rate, the water-to-air ratio is 1.2 % at engine base load, and 1.6 % at 50 % engine load.

The main water wash parameters are given in Table 1, as well as accumulated operating hours for each of the water flow rates. For the highest flow rate of 50 l/min, 55,000 engine operating hours are accumulated.

Table 1. Online water wash test matrix [9]

	Single flow config	**Double flow config**	**Triple flow config**
Water rate (actual)	17 l/min	30 l/min	50 l/min
Pump rate	30 l/min	38 l/min	54 l/min
Water-to-air ratio (engine load)	0.4–0.5% by mass (100–50% engine load)	0.7–0.9% by mass (100–50% engine load)	1.2–1.6% by mass (100–50% engine load)
Pressure	60 bar	60 bar	60 bar
Temperature	60 ℃	60 ℃	60 ℃
No of nozzles	5	10	10
Droplet diam.	150 um	150 um	150 um
Engine hours	120,000 *	260,000 *	55,000 *

*Operating hours as per Sept 2018

Parameters such as water droplet size, injection pressure and temperature are kept constant in all the tests conducted. The wash effectiveness impacts of variations in these parameters are not covered in the current work. The main parameter that is varied is the water flow rate.

Engine transients

An instance of engine transients during the online wash sequence is seen in Figures 25 and 26, for an engine operated at part-load. N1-speed increases during the wash sequence, but stabilizes close to the initial condition after the water is shut off. T3 decreases, PS3 increases slightly and T5.4 decreases during water ingestion. The higher the water ratio, the higher the temperature drop of T3 and T5.4.

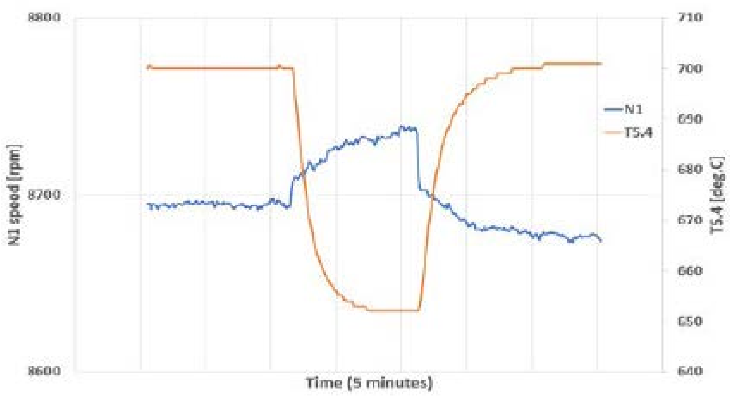

Figure 25. Engine response to water ingestion, N1 / T5.4 [9]

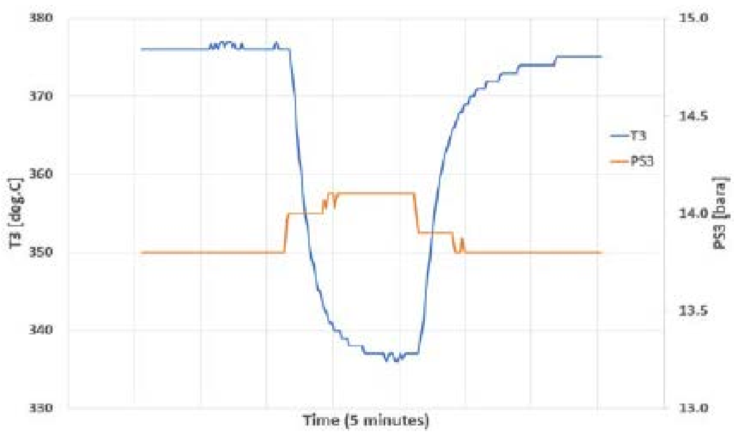

Figure 26. Engine response to water ingestion, T3 / PS3 [9]

The T3 and T5.4 temperature drop is in line with the thermodynamic theory, since the energy to evaporate the water is extracted from the air through the compressor. Thus, the PS3 increases due to higher mass flow through the compressor. When engine load is high, the impact of temperature-drop (particularly T3) is less than in part-load operation.

When the engine is running at T5.4 control (T5.4 maximum) prior to water ingestion and the T5.4 drops during water wash, the control system demands more power from the turbine, hence T5.4 will increase to compensate for the initial drop. Depending on the control system (TCP), the T5.4 response during water ingestion is somewhat different between old and new systems. Figures 27 and 28 show the response with an old TCP, where the T5.4 control limit is not restored until the water injection is stopped.

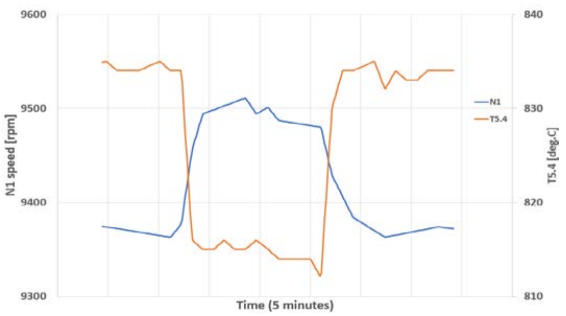

Figure 27. Engine response to water ingestion, N1 / T5.4 [9]

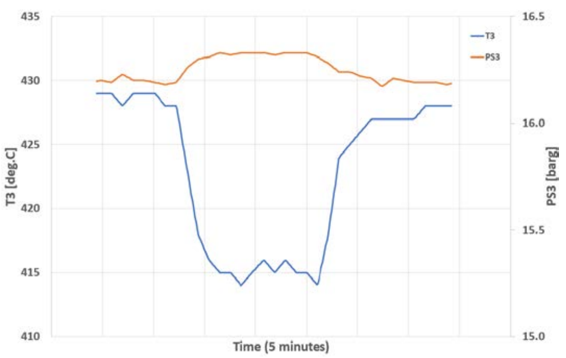

Figure 28. Engine response to water ingestion, T3 / PS3 [9]

With a new TCP shown in Figure 29 and 30, T5.4 very quickly restores the control settings after the water is injected, followed by a stable value until the water is stopped and a peak is seen before again restoring the T5.4 control level. Other than the difference in T5.4 response, the other parameters are very similar for all cases.

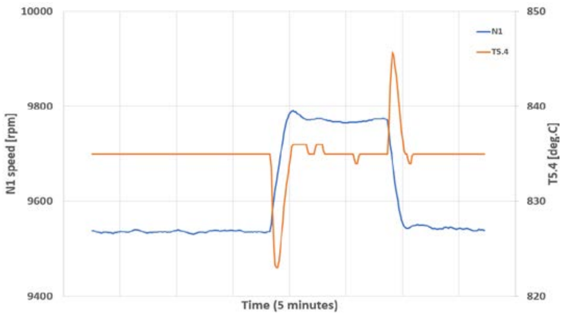

Figure 29. Engine response to water ingestion, N1 / T5.4 [9]

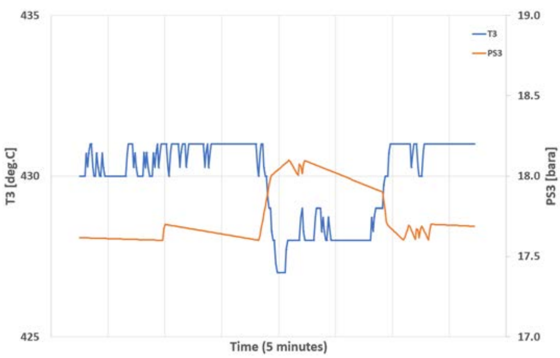

Figure 30. Engine response to water ingestion, T3 / PS3 [9]

4.3 Performance analysis

HPC efficiency trend data is used as the main performance deterioration parameter for the different online water wash rates. With no online wash, the deterioration was 4.5 % over a 4-month operating interval between offline washes, as indicated in Figure 31.

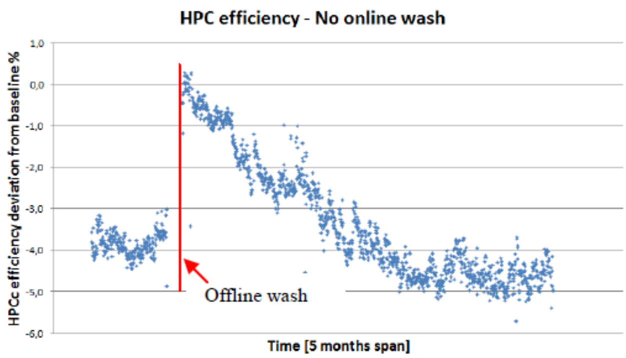

Figure 31. HPC deterioration with no online wash [1]

When starting online wash at 17 l/min flow rate, the deterioration was reduced to 3 % over a 4-month operating interval, as indicated in Figure 32.

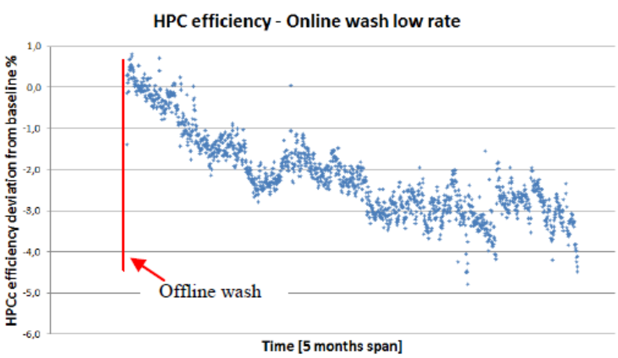

Figure 32. HPC deterioration with online wash 17 l/min [1]

When increasing the online wash flow rate to 30 l/min, the deterioration was reduced to 2 % over a 4-month operating interval, as indicated on Figure 33.

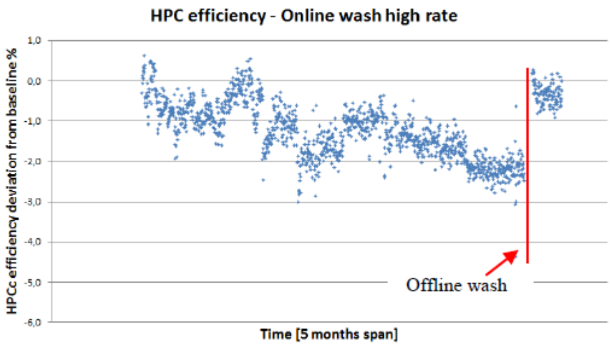

Figure 33. HPC deterioration with online wash 30 l/min [1]

Prior to a further increase in online wash flow rate, the intake air filters were upgraded from M6 to F7 filter class, and there was also an instrumentation upgrade. After this upgrade, the operating interval was extended from four to six months between offline washes. A HPC deterioration of 1 % over six months was achieved, as indicated in Figure 34. This trend data set is very smooth and clear, with a close to linear deterioration rate of 1 % between offline washes, and a further lift of 0.5 % in efficiency baseline after GG change (due to non-recoverable deterioration).

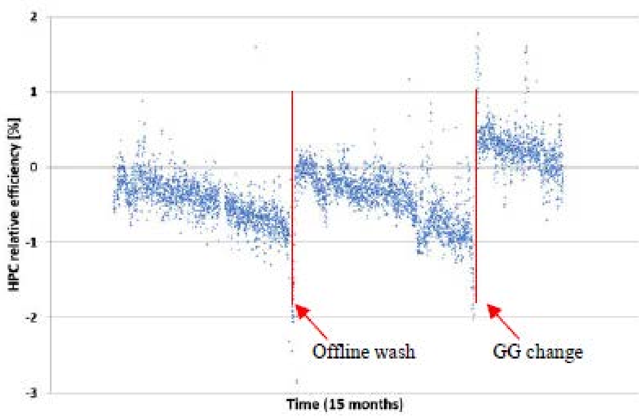

Figure 34. HPC deterioration with online wash 30 l/min and F7 filters [5]

Finally, when increasing the online wash flow rate to 50 l/min, the deterioration was reduced to 0.5 % over an operating interval of six months, as indicated on Figure 35.

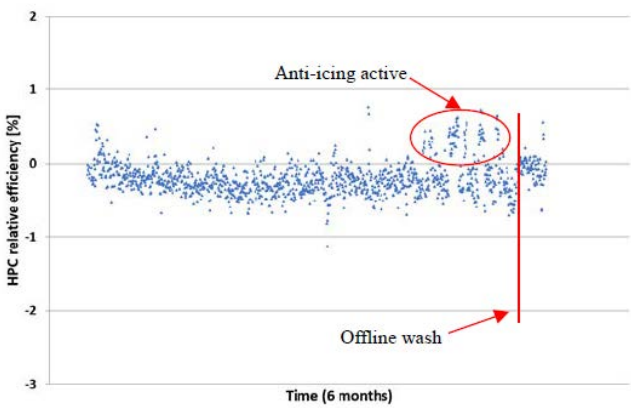

Figure 35. HPC deterioration with online wash 50 l/min [9]

These long-term trends demonstrate that a high water-to-air ratio is advantageous for achieving as low a compressor degradation as possible, and thus the potential to extend the shut-down interval for offline/crank wash and improve the overall performance and efficiency. A summary is presented in Table 2, with combinations of filter and water wash parameters.

Table 2. Air filter and water wash parameters, engine data [9]

	Config #1	**Config #2**	**Config #3**
Intake air filter type, change interval	M6 (12 months)	F7 (6 months)	F7 (6 months)
Online water wash rate	30 l/min	30 l/min	50 l/min
HPC efficiency degradation, operating interval	2–3% (6 months between offline wash)	~ 1% (6 months between offline wash)	~ 0.5% (6 months between offline wash)
Normal engine load, engine application	Part load (~75%) Compr.drive	Base load (100%) Compr.drive	Part load (~ 75%) Gen.set

4.4 Onshore engine shop inspections

Airfoil and engine inspection have been thoroughly analyzed in relation to the online water wash project at the subject offshore field. The engines have been subject to borescope inspections of the HPC, combustion chamber and HPT every six months. More detailed inspections have been performed when the engines are withdrawn for overhaul (HSR or OH), where a closer inspection can be conducted in the workshop. Each time the online water wash rate has been increased (ref. Table 1), this has been done on a pilot engine in the final period of the GG life prior to a planned overhaul, and an engine with medium criticality for production has been selected. In this way, a good assessment of the effect of online water wash and an upfront risk mitigation have been conducted to reveal any possible downsides (e.g. erosion, corrosion), prior to the extension of higher water rates to engines with high production criticality.

Figure 36 consists of pictures from an engine running with “configuration #3” as shown in Table 2. The high 50 l/min water rate was operated during the final year prior to withdrawing the engine for planned overhaul. No offline wash was performed during the last six months.

The main observation is that all stages (rotor and stator vanes) are in good condition without any abnormal wear. The stage 1 rotor blade has some deposits on the suction side, most dominant at the LE. The PS of the stage 1 rotor blade is very clean, except some deposit spots at TE near the tip.

Figure 36. HPC stage 1 rotor blade SS (left), stage 1 rotor blade PS (right) [9]

A comparison of online wash flow rates is shown in Figure 37 for the stage 7 rotor blade. The left-hand photo is from an engine running with “configuration #3” as shown in Table 2 vs. an engine running with “configuration #1” on the right-hand side. For the engine with “configuration #1”, the lower 30 l/min water rate was operated during the final eight months of operation prior to withdrawing the engine for planned overhaul and no offline washing during the last four months. Here, the deposits at PS are dominant near the hub and tip of the blade. It should also be pointed out that this engine was operated with a lower air inlet filter class and lower operating intervals (between offline washes) than the engine on the left-hand side of Figure 37, which may affect the deposit characteristics.

 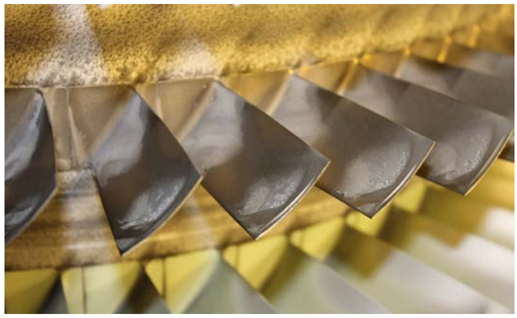

Figure 37. HPC stage 7 rotor blade PS, left side 50 l/min, right side 30 l/min water rate [9]

4.5 Conclusions

Online water washing at high flow rate is the key to keeping turbine performance high and minimizing compressor deterioration. It is therefore possible to increase offline wash intervals without running the compressor into a severe fouling condition.

The main findings in the current work:

- A high water-to-air ratio (1.4 % by mass) is the key parameter for achieving increased power recovery and reducing long-term deterioration.

- The HPC efficiency deterioration is reduced from 1.0 % to 0.5 % over each 6-month operating interval by operating online wash daily at 1.4 % water-to-air ratio versus the earlier level of 0.8 % w.a.r. (similar air inlet filter class).

Considerations:

- Running at an excessive water flow rate may cause operational issues and long-term effects such as:
    - Cooled HPC casing, risk of rub
    - Flame out
    - Icing at HPC inlet (cold ambient conditions)
    - Long-term effects of corrosion/erosion
    - Staging/burning mode change (DLE engines)

- No such negative effects have been observed at the water rates analyzed in this work.

- The LM2500 base engine has a sufficient surge margin to operate with this online water wash rate. Other engine types (as well as engines with water injection, e.g. for emission control), may operate much closer to the surge line, which means that online water rates must be thoroughly studied and assessed.

5. Air Intake Filter System

5.1 Definitions

Filter system

The filter system's main components are described in Figure 38. Upstream of the filtration section is an anti-icing system which is either a manifold with a heating medium from the WHRU system or a manifold with hot air supply from engine CDP bleed. The filtration section consists of a vane separator stage followed by a filter stage. Downstream of the filtration section is a silencer ducting, and finally the plenum in front of the engine inlet. The entire intake system is arranged in a horizontal configuration.

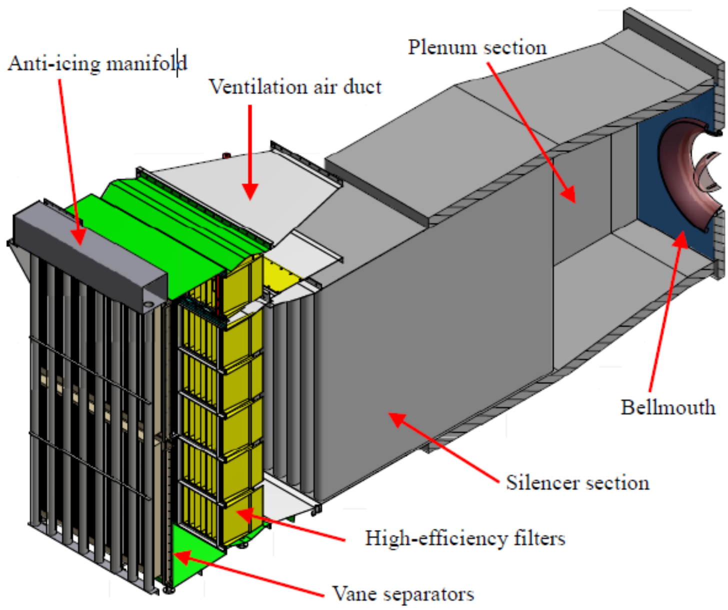

Figure 38. 3D view of 2-stage gas turbine intake system [courtesy of Nymo]

Filter classification

The filters analyzed in the current work are promoted with an M6 and F7 rating in accordance with the European EN 779:2012 standard shown in Table 3. In Table 3, the EN 779 and EN 1822 standards are combined in order to cover the whole range of filter classification groups from Coarse to ULPA.

Table 3. European EN filter classification [13]

Group	EN Filter Class	EN 779 : 2012		EN 1822 : 2009	
		Average arrestance (A_m) of synthetic dust (%)	Average efficiency (E_m) of 0,4 µm particles (%)	Total filtration separation efficiency (%)	Local filtration separation efficiency (%)
Coarse	G1	$50 \leq Am < 65$			
	G2	$65 \leq Am < 80$			
	G3	$80 \leq Am < 90$			
	G4	$90 \leq Am$			
Medium	M5		$40 \leq Em < 60$		
	M6		$60 \leq Em < 80$		
Fine	F7		$80 \leq Em < 90$		
	F8		$90 \leq Em < 95$		
	F9		$95 \leq Em$		
EPA	E10			85	
	E11			95	
	E12			99.5	
HEPA	H13			99.95	99.75
	H14			99.995	99.975
ULPA	U15			99.9995	99.9975
	U16			99.99995	99.99975
	U17			99.999995	99.9999

Old filter housing design

The subject gas turbine packages initially had a 3-stage high velocity system as shown in Figure 39. The filter system had vane separator stages both upstream (often referred to as louvre) and downstream (often referred to as separator) of the filter stage. The theory behind such a design is that the filter should be able to let some water through on the downstream side, which will eventually be drained out in the final vane separator. However, in real operating conditions in harsh offshore environments, the final vane separator does not have the desired function (designed to separate large droplets, whilst the majority of droplets passing the filters are sub-micron particles) and the typical filters used in such applications are soaking wet after a short operating time. The filters then rapidly lose performance as well as mechanical strength and let high amounts of salt and water through to the downstream side.

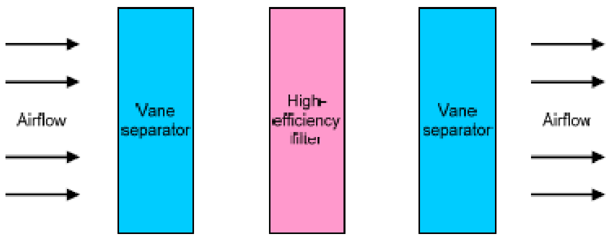

Figure 39. Layout of 3-stage gas turbine intake air system (single filter system) [14]

Upgraded filter housing design

The engines analyzed in this work have developed from an original filter configuration as illustrated in Figure 39 with a louvre upstream and a vane separator downstream of the filter, to an upgraded and simplified 2-stage system. The layout of the upgraded 2-stage gas turbine intake air system is illustrated in Figure 40 (only the forward part of the air intake filter housing is upgraded i.e. vane separator and filter section), where the combustion air and package ventilation air have a common intake system and the two ducts are divided downstream of the filter (as illustrated in Figure 38).

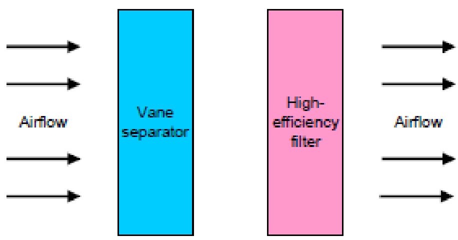

Figure 40. Layout of 2-stage gas turbine intake air system (single filter system) [14]

These systems rely on an efficient upstream vane separator to remove the vast majority of water from the airflow before it reaches the high-efficiency filters. The high-efficiency filters are especially designed to withstand moisture, and seals on the bottom of each filter bag are intended to allow for water that reaches the filters to be drained out on the upstream side of the filter elements. These filters constitute the final barrier before the airflow enters the gas turbine compressor.

5.2 Development of new filters for the offshore environment

After the filter housing upgrade, the filters had a M6 rating, and the system saw a major improvement in air filtration effectiveness compared with the old design. The major advantage was to have a more efficient vane separator on the upstream side of the filters, more efficient filters with better hydrophobic media properties and built-in drainage function, as well as an improved filter housing drainage system. However, further optimization of filtration effectiveness was targeted by looking for even better hydrophobic media properties in the filter elements. This was done through the development and testing of two new additional filter types. These were an intermediate type designated “M6+” and eventually a filter type with a F7 rating.

5.3 Offshore field testing - performance analysis

The deterioration rate of HPC efficiency was compared between the three filter types (M6, M6+ and F7). Other parameters such as operating intervals and online water wash rates were kept constant for consistent comparison of the filters. When changing to M6+ filters, there was no impact on the deterioration trend compared with M6, as can be seen on Figure 41. This was quite unexpected, since the M6+ had better efficiency from the filter test standard vs. M6. However, when changing to F7 filters, the deterioration trend was significantly improved by a reduction in deterioration from approximately 2.5 % to 1 % over six months.

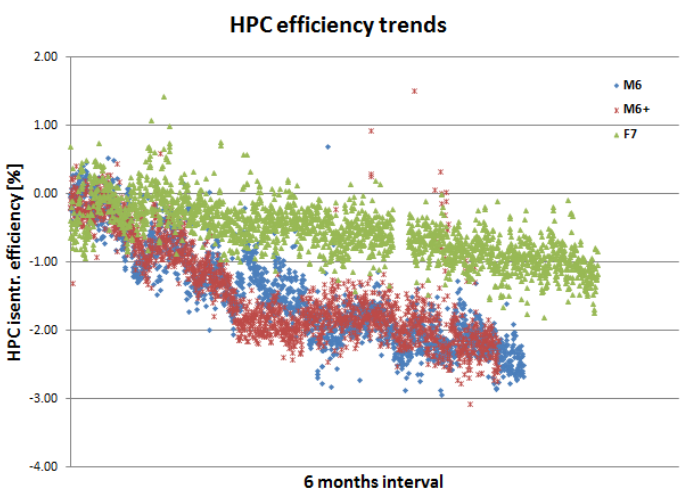

Figure 41. HPC efficiency trends with various filter types [14]

5.4 Onshore test cell data

The test rig is utilized for testing both new and used filter elements of various filter grades (M6, M6+, F7) and used filters with various operating hours offshore. The principal objective is to reveal the hydrophobic properties and performance of the filters, as well as documenting filter degradation characteristics over time. This is performed by running a salt ingestion program and measuring “brine” (saltwater droplets) downstream of the filter, as well as running a dp profile.

As expected, a higher filter class yields an increase in dp in new condition, which is illustrated in Figure 42.

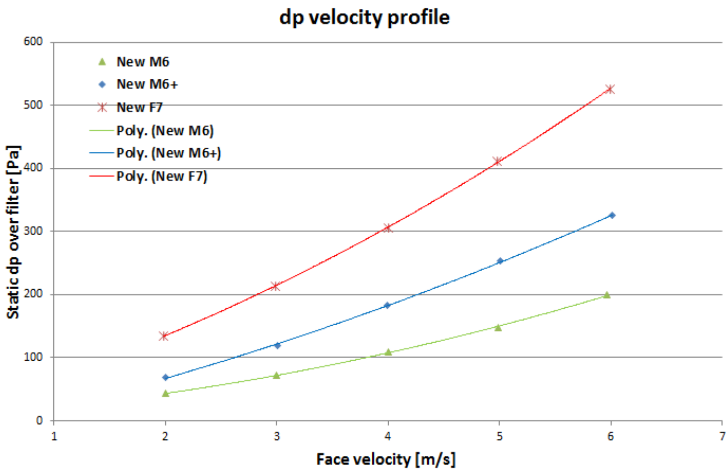

Figure 42. Dp-velocity profile for new filters [14]

The salt water ingestion test data are presented in Figure 43. The main result from the tests is that the M6 filters keep the performance close to new condition after six months of operation. However after twelve months of operation, the degradation is visible on most test points, but the performance is still not very different from the new condition. This means that the M6 filters yield a minor improvement in HPC degradation if changed at six month intervals vs. twelve months.

For the M6+ filters, the initial performance is clearly much better than the M6 filters. However after six months of operation, the degradation is significant and the levels are near the M6 on most test points. This means that the overall effect of changing from M6 to M6+ is marginal, and the degradation over time is rapid for this filter type.

For the F7 filters, the initial performance is clearly the best out of the three different filter types. After six months of operation, the degradation is visible on all test points, but still significantly better than both M6 and M6+ in used condition.

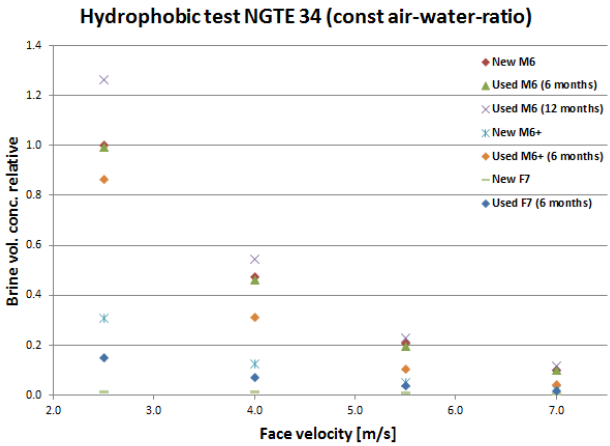

Figure 43. NGTE 34 test results [14]

The main conclusion from the test program is that the F7 filters demonstrate a significant improvement in performance in both new and used condition vs the other filter types. A six month operating interval between filter changes yields a very good overall result. These results are also supported by the testing of used filters performed at the filter OEM, where filter efficiency is measured as well as destructive testing and analysis of the filter layers.

The data from the M6+ filter is a typical example that proves that even if the initial efficiency looks good, the rapid deterioration of such a filter yields no improvements in the field. It is easy to only consider filter class and think that a higher filter class improves the performance. However, the filter's hydrophobic media properties are of vital importance in an offshore environment, to ensure performance over the installed period for the filter.

Subsequent operational test periods with a twelve month filter change interval for F7 filters documented the sensitivity for high humidity/fog if these filters were operated for excessive periods. In such cases, the dp increase was significant, which has to be compensated by use of anti-ice heating to dry up the filters in these conditions. A steam test in the test cell facility also documented this effect. This is shown in Figure 44.

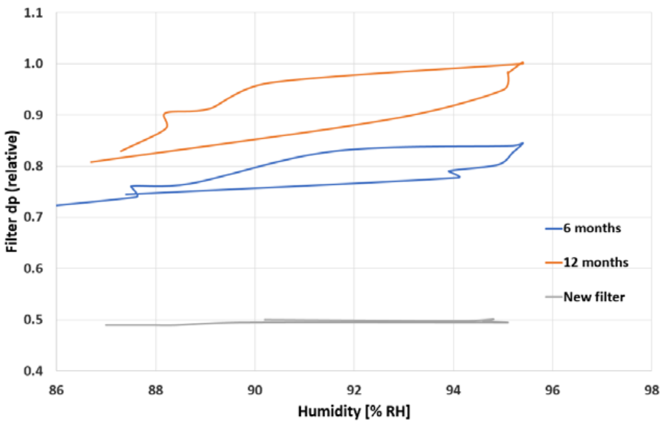

Figure 44. Steam test data [15]

5.5 CFD analysis air intake system

The main findings from the CFD analysis are the difference in velocity distribution between bleed-air and radiator anti-ice manifold systems. Velocity profile at the plane immediately upstream of the filter section is shown in Figure 45, where the velocity for the radiator is significantly higher than the bleed-air manifold. In addition, the velocities are well above the design value (5.5 m/s) in the majority of the filter area.

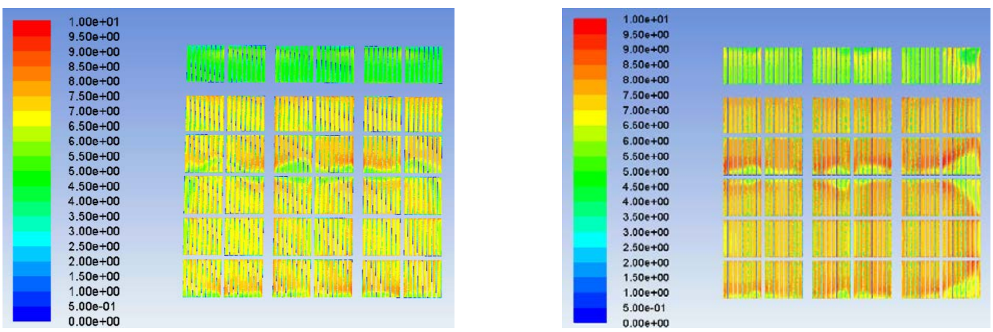

Figure 45. Velocity (m/s) at the plane upstream filters, bleed air left, radiator right [14]

The anti-icing radiator yields an overall higher velocity in the whole combustion air area due to the flow restrictions/blockage on the structural geometry. The flow effect is shown from a top-down view in Figure 46, where the lower left area is most affected due to a large blockage area. This effect can, in turn, be seen on the right-hand side of Figure 45, where local areas for combustion air have low velocities, which leads to high velocities for the majority of the combustion air area.

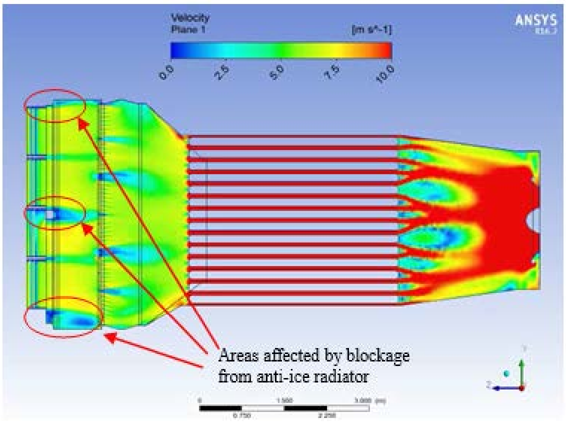

Figure 46. Velocity (m/s) for the entire intake system, radiator [14]

CFD field results

- Plate/cover on heating medium radiator removed (seen in Figure 47).
- Approximately 0.8 m2 increased inlet area.
- In future projects: design better symmetry between radiator and vane separator structural elements.

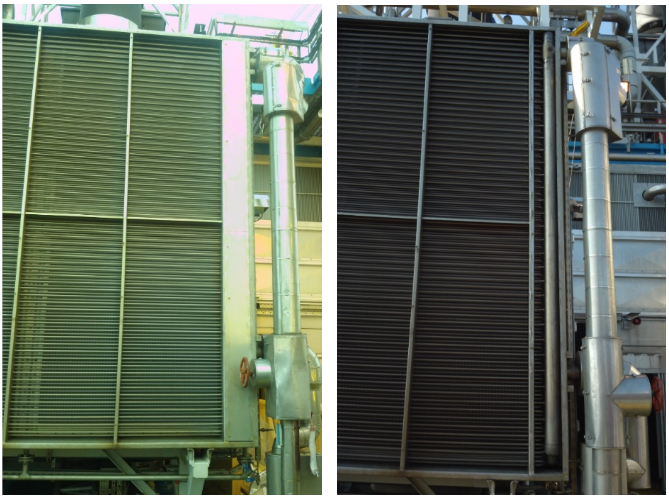

Figure 47. Radiator cover mounted (left) and dismantled (right)

Table 4. CFD set up Ansys [14]

Software:	Ansys Workbench & Fluent v.16.2
Material properties:	Air @ 10 deg.C / 1 atm; density 1.225 kg/m3, viscosity 0.017894 cP
Boundary conditions:	Air inlet: air enters the domain to balance the outflow to the gas turbine and the vent outlet. The reference pressure has been set to equal the pressure at the air inlet. Turbine inlet: 69.8 kg/s of air flow into the gas turbine. Vent outlet: 11.25 kg/s of air leave via the vent outlet.
Meshing:	Polyhedral meshes have been used with a cell count of approx. 20 million
Turbulence model:	Realizable k-epsilon
Other conditions:	The model is solved with a steady-state solver. Hot air from bleed-air anti-ice manifold not accounted for. Vane separator and filter pressure drop is modelled as viscous resistance as a function of air velocity.

5.6 Field improvements air intake system

In addition to the improvements by running a F7 filter at a 6-month interval and the flow optimization on the radiator cover described in the previous chapter, other filter housing features have also been improved.

Summary of field optimization:

- Holding frame reinforced with structural elements to avoid bending. Improved sealing (new gasket material) to avoid leakages
- Improved drainage system: piping slope without water traps, i.e. dry air-lock check valves (avoiding low points where water and dirt can accumulate)
- Improved access door design and ladder for easy access to the upper filter rows
- Gas detectors and lighting moved to the outside walls: i.e. limit restrictions on the air flow path within the filter housing, as well as improved maintenance access

5.7 Conclusions

Optimizing the inlet air filter system and selecting the correct filter system components to handle harsh offshore weather conditions have provided significant improvements in gas turbine compressor deterioration. Thus, longer operating periods (up to six months) can be achieved with better average power output.

The testing and operational experience analyzed in this work documents that the F7 filters yield a significant improvement in reducing long-term deterioration. A 6-month filter change interval is considered an optimum interval for the current F7 filters as the best compromise between performance, deterioration and cost.

6. Gas Turbine Air Flow Analysis

6.1 Definitions

In the current study, two pilot compressor drive engines are instrumented with a new static P2 probe (shown in Figure 48) in order to achieve an inlet depression measurement and thus be able to monitor compressor air flow. The purpose of this is to compare compressor air flow vs. efficiency degradation for these operating LM2500 engines, in order to document whether air flow degradation is a more viable parameter when monitoring compressor fouling vs. the established compressor efficiency monitoring.

6.2 Intake depression - new instrumentation

The intake depression is the difference in measured static pressure at the Bellmouth throat from the stagnation pressure at the Bellmouth intake (plenum stagnation pressure). The Bernoulli equation states the correlation between static pressure and air velocity, and hence the air mass flow rate in the engine Bellmouth. The intake depression is nondimensionalized by the plenum stagnation pressure. The deterioration in inlet mass flow rate is monitored by comparing the intake depression to the corrected engine shaft speed (N1c). A baseline curve is established for the engine after an offline water wash, which is used as reference point for future comparison.

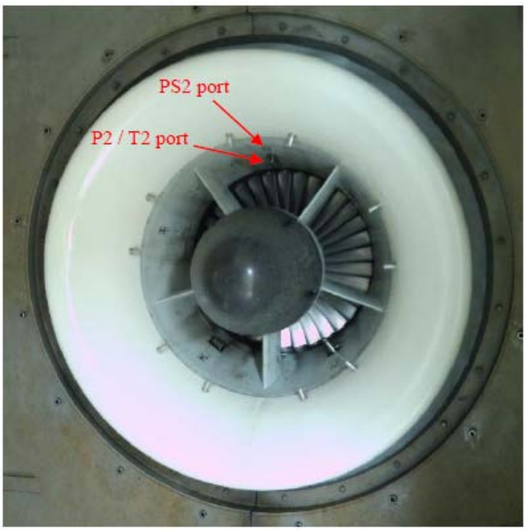

Figure 48. Engine inlet / Bellmouth with PS2 probe [8]

Thus, compressor air flow is calculated / measured by means of the engine intake depression, by using the pilot static pressure probe PS2 and total pressure probe P2 as the basis for the calculation. The properties of atmospheric air, such as gamma, gas constant and density, can be calculated by an appropriate EOS based on the actual temperature, pressure and humidity. For reasons of simplicity, in the current study, a spreadsheet methodology has been used for calculating the air property parameters.

Compressor inlet air mass flow is calculated by the following relations using Bernoulli's equation:

Mach number:

$$Ma = \sqrt{\frac{2}{\gamma - 1}\left[\left(\frac{PS2}{P2}\right)^{-\frac{(\gamma-1)}{\gamma}} - 1\right]}$$

Axial velocity engine inlet:

$$Ca = Ma\sqrt{\gamma\, R\, T2}$$

Volume flow engine inlet:

$$Q = Ca\, A$$

Mass flow engine inlet:

$$\dot{m} = Q\, \rho$$

A gas turbine axial compressor can be regarded as a "constant volume flow machine" at a given speed (in clean condition). The volume flow is constant, although mass flow increases at a given speed when the inlet temperature decreases.

Selected flow coefficient definitions:

$$\phi = \frac{Q}{N}$$

$$\phi = \sqrt{\frac{P2 - PS2}{\rho}}$$

6.3 Offshore field testing, data collection

A baseline curve is created from field data in clean engine condition. The engine operating points in clean condition are shown in Figure 49, where there is a spread of data points throughout the operating range of the engine. A baseline curve is constructed based on the data points with a polynominal fit.

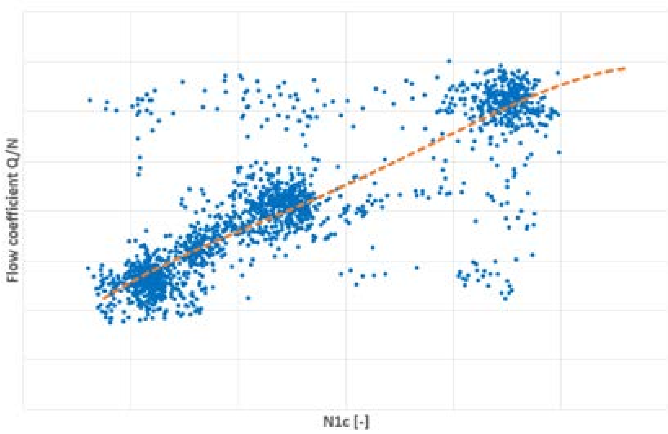

Figure 49. Baseline curve flow coefficient [8]

6.4 Performance analysis

The trend data for flow coefficient are shown in Figure 50 for a 6-month operating period, including two offline washes. Flow coefficient deviation is defined as the difference between the operating point and the baseline shown in Figure 49.

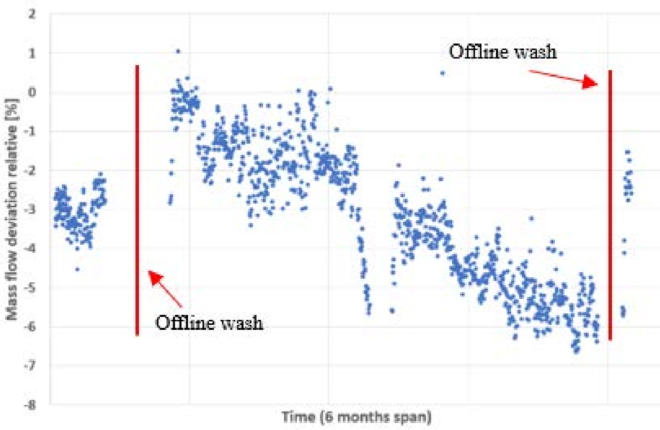

Figure 50. Mass flow deterioration [8]

Hysteresis effect of ramping-up/down

The hysteresis effect of ramping-up vs. ramping-down can be clearly seen in Figure 51. The "ramp-up" and "ramp-down" points are taken at similar N1c, which means the two points are following the near vertical speed line in this operating range of the map. This implies that the operating point is closer to surge at the ramping down sequence. Operational experience shows that there is little drift-off of the VSV schedule over the lifetime of a typical engine if the setting is optimum when the GG is installed, the engine is kept clean with low deterioration through its lifetime (25 - 30k operating hours) and proper maintenance routines are applied. This has been confirmed through an incoming performance test in a test cell facility, where the VSV schedule was close to the stall line, i.e. at the optimum point.

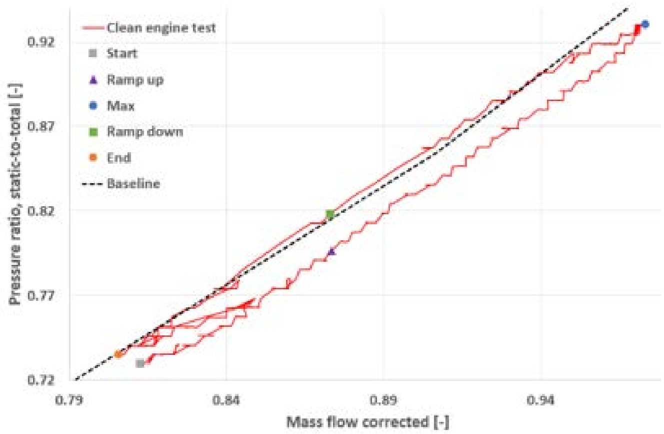

Figure 51. Performance test - hysteresis clean engine [8]

Performance shift

Figure 52 shows a comparison of clean and deteriorated condition (pre/post offline wash) which demonstrates that the deteriorated curve is shifted towards reduced pressure ratio and decreased corrected mass flow rate in the map vs. the clean engine, i.e. reduced surge margin.

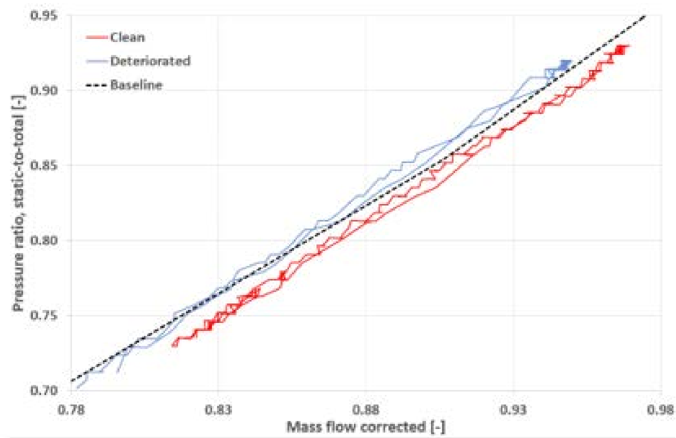

Figure 52. Performance test clean vs. deteriorated engine [8]

6.5 Correlation air flow vs. efficiency

By comparing compressor efficiency and air flow on this basis, we find that the air flow has twice the deterioration rate on a percentage basis (shown in Figure 53). However, the profile of the trend is almost identical, meaning that both the efficiency and air flow parameters have a good correlation.

Figure 53. HPC efficiency vs air flow trend data [8]

6.6 Conclusion

The main conclusion from this work is that air flow has twice the deterioration rate on a percentage basis vs. HPC efficiency, as shown in the combined plot in Figure 53. However, the profile of the trend data of these two parameters is almost identical, meaning that efficiency and air flow have a good correlation. Both parameters can be used for effective condition monitoring of gas turbine compressor degradation, given that the appropriate correction of the parameters is applied. Finally, the shift in performance curves from clean to deteriorated condition is documented, as well as the hysteresis effect related to load changes.

7. Conclusions

7.1 Project findings

Main project findings:

- Online water wash: high water-to-air ratio (1.4 % by mass) is the key parameter for achieving increased power recovery and reducing long-term deterioration.
- Air intake filter: upgrade of inlet air filter housing design and an upgrade from M6 to F7 filters yields a significant improvement in reducing long-term deterioration.
- Compressor efficiency correction algorithm developed for improved condition monitoring.
- Compressor air flow - new parameter developed for improved condition monitoring: twice the deterioration rate on a percentage basis vs. efficiency.

7.2 Improvement in deterioration rate and engine performance

The combined effect of improved online wash parameters and improvements to the intake air filter system is an efficiency improvement of some 6 % over a 6-month operating interval between offline washes. This is illustrated in Figure 54.

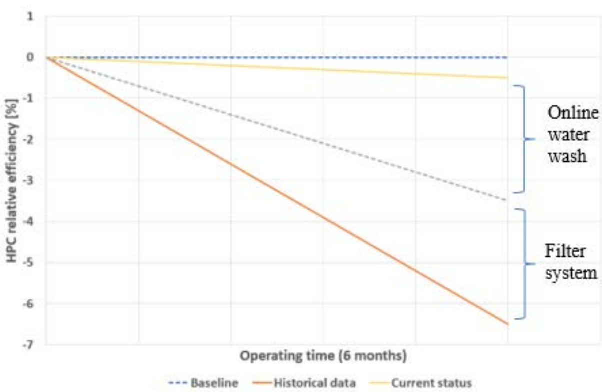

Figure 54. Comparison of HPC degradation - historical and current [9]

Comparison before and after

A visual impression of the long-term improvements, can be seen in engine inspection photos from workshop visits when engines are scheduled for HSR/OH. The left-hand side of Figure 55 shows the historical condition of the HPC casing of one engine from the subject field, where large amounts of salt/soap have accumulated on the aft stages of the compressor. This was prior to air intake filter upgrades, with old low-pressure offline water wash system (no online wash), and with shorter operating intervals (two months). It can be observed that the aft compressor section is significantly deteriorated, due to an ineffective air filtration system, as well as an ineffective offline wash system which only moved the deposits to the aft section of the compressor where it accumulated.

After the upgrade of the water wash system and intake air filter system, the right-hand side of Figure 55 shows a significant improvement from an engine from the same field, with a very clean HPC casing, even after longer operating intervals (six months) between offline washes (no offline wash performed prior to GG change).

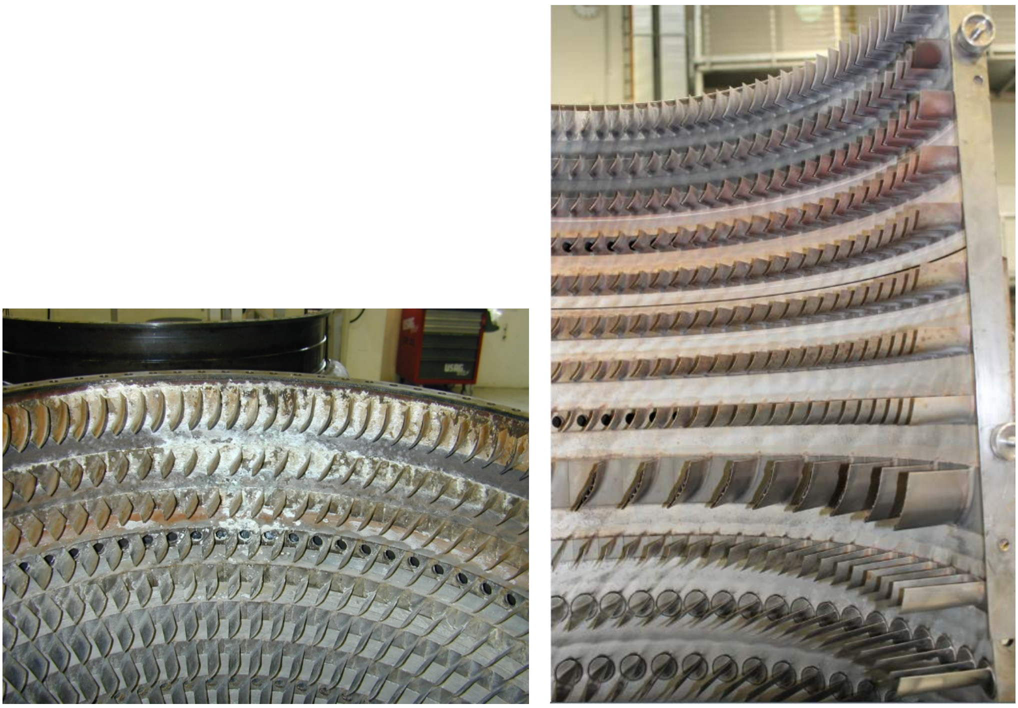

Figure 55. Engine condition following shop visit

7.3 Summary

The combined impact of an improved water wash system and improvements in the intake air filter system, has been a significant improvement in the gas turbine deterioration rate.

The best possible results are obtained largely through finding the optimum combination of offline and online water wash (at high water rate) combined with an optimum inlet air filter system.

Overall, the objective of this PhD has been achieved: i.e. to obtain the highest possible engine performance, availability and efficiency.

8. Further Work

The calibration and optimization of the offshore instrumentation and condition monitoring system is necessary to obtain accurate measurements and good trend data. The shift in performance on the compressor map from clean to deteriorated engine should be further studied, in particular with regard to an analysis of whether the speed line shapes have shifted. The same applies for the effect on velocity diagrams and Reynolds number effects. The sensitivity in engine parameters for HPC efficiency vs. air flow calculations should be further studied, e.g. the effect/sensitivity on performance due to inlet humidity which has been emphasized in other such studies [16].

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Appendices

Appendix I

Gas Turbine Operation Offshore; On-line Compressor Wash Operational Experience
Proceedings of ASME Turbo Expo 2014; Power for Land, Sea and Air
June 16-20, 2014, Düsseldorf, Germany
Paper no. GT2014-25272

Is not included due to copyright

Appendix II

Gas Turbine Operation Offshore; Increased Operating Interval and Higher Engine Performance Through Optimized Intake Air Filter System
Proceedings of ASME Turbo Expo 2016; Turbomachinery Technical Conference and Exposition
June 13-17, 2016, Seoul, South Korea
Paper no. GT2016-56066

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Appendix III

Gas Turbine Fouling Offshore; Correction Methodology Compressor Efficiency
Proceedings of ASME Turbo Expo 2017; Turbomachinery Technical Conference and Exposition
June 26-30, 2017, Charlotte, North Carolina, USA
Paper no. GT2017-63025

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Appendix IV

Gas Turbine Fouling Offshore; Air Intake Filtration Optimization
Proceedings of ASME Turbo Expo 2018; Turbomachinery Technical Conference and Exposition
June 11-15, 2018, Oslo, Norway
Paper no. GT2018-75613

Errata:

Reference [6] should read:
Madsen, S., Bakken, L. E., 2018, “Gas Turbine Fouling offshore; effective online water wash through high water-to-air ratio”, proceedings of ASME Turbo Expo 2018, Oslo, Norway, ASME GT2018-75618.

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Appendix V

Gas Turbine Fouling Offshore; Effective Online Water Wash Through High Water-to-Air Ratio
Proceedings of ASME Turbo Expo 2018; Turbomachinery Technical Conference and Exposition
June 11-15, 2018, Oslo, Norway
Paper no. GT2018-75618

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Appendix VI

Gas Turbine Fouling Offshore; An Analysis of Engine Air Flow
Proceedings of ASME 2018 Power and Energy Conference
June 24-28, 2018, Lake Buena Vista, Florida, USA
Paper no. PowerEnergy2018-7269

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Appendix VII

Gas Turbine Operation Offshore; Online Compressor Wash Operational Experience
Journal of Mechanics Engineering and Automation, Volume 4, Number 12, December 2014
Paper no. 4-JMEA-E20140811-1

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Gas Turbine Operation Offshore: Online Compressor Wash Operational Experience

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Abstract: Online compressor wash for six GE LM2500PE engines at a Statoil North Sea offshore field is analyzed. Three engines are generator drivers whilst three engines are compressor drivers. Two of the compressor drive engines are running at peak load ($T5.4$-control), hence production rate is limited by the available power from these engines. All the six engines analyzed run continuously without redundancy, hence gas turbine uptime is critical for the field's production and economy. The performance and operational experience with on-line wash at different water-to-air ratios and engine loads, as well as economy potentials related to successful on-line wash are given. This work is based on long-term operation with on-line wash, where operational data are collected and performance analyzed, over a 4-5 year period. All engines are operated with four-month intervals between maintenance stops, where off-line crank-wash is performed as well as other necessary maintenance and repairs. On-line wash is performed daily between the maintenance stops at full load (i.e., normal operating load for the subject engine). To keep the engine as clean as possible and reduce degradation between maintenance stops, both an effective on-line water wash system as well as effective air intake filter system, are critical factors. The overall target is to maintain high engine performance, and extend the interval between maintenance stops through effective on-line wash. It is of vital importance to understand the gas turbine performance deterioration. The trending of its deviation from the engine baseline facilitates load-independent monitoring of the gas turbine's condition. Engine response to water injection at different loads and water-to-air ratios, as well as engine response to compressor deterioration is documented and analyzed. Instrument resolution and repeatability are key factors required in order to obtain reliable performance analysis results. Offshore instrumentation on older installations is often limited to the necessary instruments for machine control/protection, and additional instruments for effective performance monitoring and analysis are often missing or, if installed, have less accuracy. As a result of these analyses, a set of monitoring parameters is proposed for effective diagnosis of compressor degradation. Avenues for further research and development are proposed in order to further increase the understanding of the deterioration mechanisms and the gas turbine performance and response.

Key words: Gas turbine performance, water wash, inlet air filter system.

Nomenclature

N1	GG rotor speed (rpm)
N2	LPT rotor speed (rpm)
P	Pressure (mbar or bar)
PS	Pressure static (bar)
T	Temperature (°C or K)
RH	Relative humidity (%)

Subscripts

c	Corrected parameter
i	Isentropic
p	Polytropic
η	Efficiency
κ	Real air heat capacity ratio (kappa)
h	Mass specific enthalpy

LM2500 Gas Turbine Station Numbers

0	Ambient condition
1	Intake plenum condition
2	HPC inlet condition
3	HPC discharge condition
5.4	LPT inlet condition
8	LPT discharge condition

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Acronyms

GG	Gas generator
HPC	High pressure compressor
HPT	High pressure turbine
LPT	Low pressure turbine
PT	Power turbine
CDP	Compressor discharge pressure
GE	General Electric
RR	Rolls-Royce
OEM	Original equipment manufacturer
R-MC	Rivenæs motor-clean
VSV	Variable stator vanes
ISO	International Organization for Standardization
WHRU	Waste heat recovery unit
SAC	Standard annular combustor
DLE	Dry low emission
MCS	Maximum continuous speed
PLC	Programmable logic controller
ASME	American Society of Mechanical Engineers

1. Introduction

Oil and gas production on several Statoil North Sea offshore installations is limited by the gas turbine power available, and any deterioration in gas turbine performance directly affects production rates. Fouling in the compressor section of the gas turbines is the main cause of performance deterioration, and the fouling is removed by water wash.

The most important finding from previous research work [1-3] is that the water-to-air ratio (by mass) during online washing should be increased compared to current online water wash systems. Water-to-air ratios of up to 3% (mass fraction) have been tested for online water wash on a GE J85-13 jet engine, and the tests revealed that online water wash at such high water-to-air ratios gave a significant increase in the power recovery after online water wash compared to online water wash at lower water-to-air ratios. Efficient online water wash will reduce the gas turbine performance deterioration caused by contaminants in the intake air, hence the power available and the efficiency of the gas turbine will increase. The reduction in performance deterioration will allow for extended intervals between stoppages for offline water wash. Efficient online water wash will increase the production in installations where the oil and gas production rates are limited by power, by increasing the average power available. The benefit from efficient online water wash in installations with 100% gas turbine redundancy will mainly be increased efficiency which implies lower fuel consumption and exhaust emissions and hence lower costs for fuel, CO_2 and NO_x taxes and reduced environmental impact.

In order to minimize compressor fouling, it is important that both intake air system and water wash are evaluated as equal important factors. Previous research [4, 5] on offshore intake air filters has shown large variations in efficiency for various filter systems as currently used in the North Sea. This demonstrates the importance of an efficient intake air filter system, in terms of optimum design, operation and maintenance routines to minimize compressor fouling and degradation.

The paper is organized as follows: Section 2 is the review of Statoil online wash experience; Section 3 is the engine description; Section 4 describes water wash system; Section 5 describes engine response to deterioration and water ingestion; Section 6 introduces process simulations; Section 7 states the performance trends/analysis; Section 8 depicts instrumentation; Section 9 introduces the economical aspects; Section 10 gives conclusions; and Section 11 gives the recommendations for further work.

2. Review of Statoil Online Wash Experience

Statoil has extensive operating experience of both offline and online water wash regimes. Offline wash is generally very efficient in terms of regaining efficiency, if the engine does not have any mechanical damage (e.g., blade tip rub) that reduces performance. The performance gain can be clearly seen on HPC efficiency trends. Typically a detergent solution (R-MC), mixed with water, is used. It is injected at a 3 min low-speed crank-sequence, followed by one to two sequences with clean water, and a final drying sequence. Both the detergent/water mix and the clean water are pre-heated to approximate 60 °C before injection. Pre-heating of the water to this temperature level is carried out for both offline and online water wash, and is based on experience. Pre-heated water yields a much more efficient compressor cleaning compared to cold/ambient water temperatures. The disadvantage of frequent offline/crank-washing is obviously that production has to be halted for several hours, if operating without redundancy. With redundancy, the engine down-time for offline wash is of less importance. An offline wash is normally conducted in combination with other necessary maintenance tasks, such as bore scope inspection, intake air filter change, exhaust duct inspection.

Online water wash discussed in this paper is defined as water injection at full load (i.e., normal operating load for the subject engine), and not idle wash (typically 5,000 rpm GG-speed for LM2500), which is another online wash regime. The advantages of online wash at full load are that there is no impact on engine downtime and production can therefore be maintained. Idle washing implies downtime not only during the wash sequence itself, but also for a time-consuming period during the ramping-up and -down of the load on the train and production wells. Depending on the train/process/well configuration, this may take several hours, especially on compressor trains. But for installations that run frequent offline wash intervals and/or have redundancy, idle washing might represent a preferable and economical washing regime in addition to offline washing. Statoil has some operational experience of idle washing from test periods offshore, with typical intervals of 1,000 running hours (offline/crank-wash interval 3,000 h). These tests have shown a potential for power recovery since the water-to-air ratio is high (fixed water flow rate). However, due to the disadvantage of engine downtime, idle washing has not been taken further to offshore operational routines in Statoil.

Online water wash is not widely used on Statoil North Sea offshore fields, but a couple of installations have extensive operating experience of online wash at full load on GE LM2500PE and RR RB211-24G engines, respectively. The operating hours accumulated by online wash are to date well above 100,000 running hours, and the LM2500 engines by far have the highest portion of these operating hours. The water-to-air ratios have been 0.5%-1.0% (mass fraction), which is in the range of maximum recommended water from the OEM (GE/RR) guidelines. The general improvement has been cleaner engine on bore scope inspections and during visual observations of the engine intake. The performance gain has been documented by overall compressor efficiency trends in some operating periods, particularly when the load and other parameters are stable (e.g., no anti-icing bleed). But when these factors vary, it is more challenging to evaluate compressor efficiency and to obtain repeatable results between operation intervals. In the absence of conclusive results from offshore online water wash, Statoil has funded extensive research and experimental testing to determine the fundamental mechanisms of axial compressor performance deterioration and recovery through online washing.

3. Engine Description

All engines analyzed are two-shaft LM2500PE type, with GE six-pack LPT. Depending on the load, the GG operates in the speed range 8,500-9,500 rpm, and LPT in the speed range 3,300-3,700 rpm.

The generator drive engines, have a direct drive configuration from the LPT to generator, and operate at 3,600 rpm. These engines use compressor bleed air (CDP 16stg compressor air) for anti-icing protection. The bleed air for anti-icing is controlled by an on/off valve controlled by ambient temperature and humidity. Moreover, an orifice is used in order to minimize the bleed air consumption. The bleed air is finally routed to an anti-icing manifold upstream of the intake air filter vane separators. The generator engines seldom run in $T5.4$ control, only occasionally when droop mode is used ($T5.4$ is set manually at a fixed value for one engine, while the other engine(s) handle load variation). Otherwise, isochronous-mode is the normal operating condition with equal load-sharing between the generators.

The compressor drive engines have a speed-increasing gearbox between the LPT and compressor. The MCS and over-speed limit for the process compressor is normally set at the LPT shaft speed control ($N2$-control compensated for gear ratio), thus a margin is set for the LPT speed limit when choosing gear ratio in order to obtain MCS at the process compressor before the LPT speed limit occurs. These engines use the heating medium from the WHRU-system for anti-icing protection, a much more energy-efficient solution than that used in the generator engines described above.

3.1 Available Power and Control Modes

The power available from the engine is highly dependent on ambient temperature. At low temperatures, the air density increases, hence the air mass flow increases. This again leads to more air feed to the combustion chamber and more power output is generated from the turbine.

At very low temperatures, however, shaft power will decrease slightly. Anti-icing influences the curve when ambient temperature is below 4.4 °C and air humidity is high (Fig. 1). The typical power characteristics shown in Fig. 2 do not include anti-icing. Anti-icing will increase $T2$ and therefore reduce power output. Thus, the most energy-efficient operation point for North Sea ambient temperature is just above the upper anti-icing limit, since average relative humidity is typically above 70 %.

The characteristics given in Figs. 3-5 for $PS3$, $N1_c$ and $T5.4$, respectively, are all at engine base load condition.

Control limit for compressor discharge pressure ($PS3$), shown in Fig. 3, only occurs at very low ambient temperatures, below the typical annual ambient temperature range in the North Sea.

Control limit for $N1_c$ speed (corrected GG speed) is shown in Fig. 4. This typically occurs at the same ambient temperature (and lower) as maximum PT shaft power. At very low ambient temperatures, the $N1c$ speed will drop below the control limit.

Control limit for $T5.4$ shown in Fig. 5, is the most common engine limitation for the typical annual ambient temperature range in the North Sea. At very low ambient temperature, $T5.4$ is not a limitation. However, at higher ambient temperature, $T5.4$ is the limiting parameter for the engine. The control limit of $T5.4$ is typically set in the range of 832-840 °C.

For ambient temperature and operational conditions in the North Sea, the most common control limits are:

- $T5.4$ control (835 °C);
- $N1$-control (GG speed);
- $N2$-control (MCS process compressor speed).

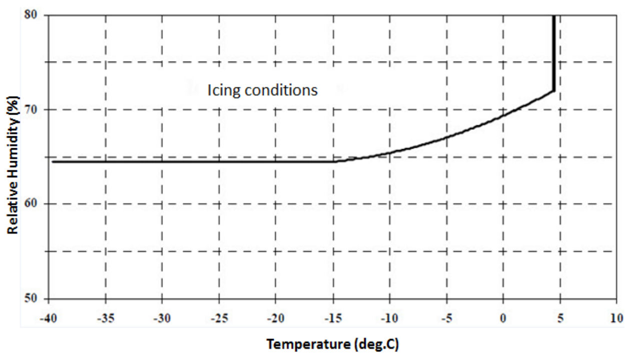

Fig. 1 Icing conditions, function of ambient temperature and relative humidity.

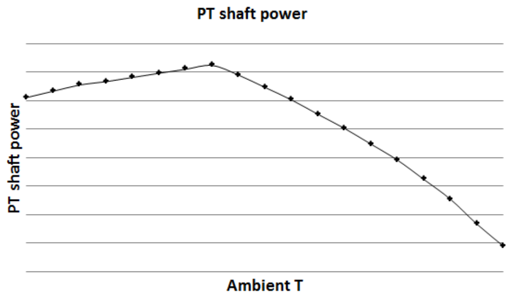

Fig. 2 PT shaft power, function of ambient temperature.

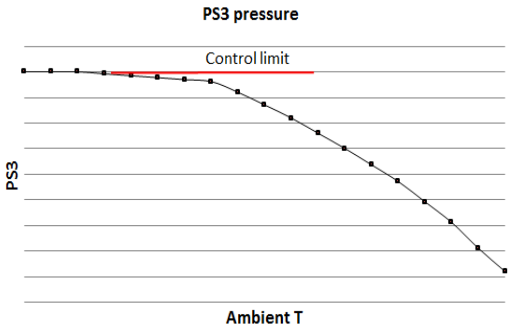

Fig. 3 PS3, function of ambient temperature.

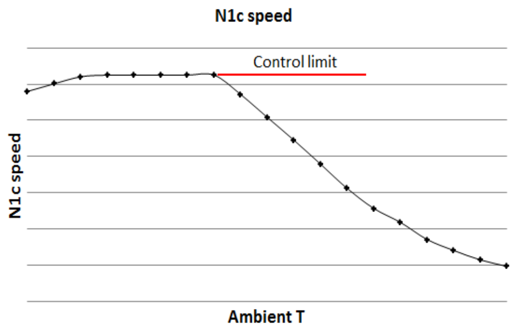

Fig. 4 Corrected GG speed, function of ambient temperature.

3.2 Inlet Air Filter Configuration

The layout of the gas turbine intake air system is illustrated in Fig. 6. These systems rely on an efficient vane separator to remove the vast majority of water and humidity from the airflow before it reaches the high-efficiency filters. The high-efficiency filters are especially designed to withstand moisture, and seals on the bottom of each filter bag are intended to allow

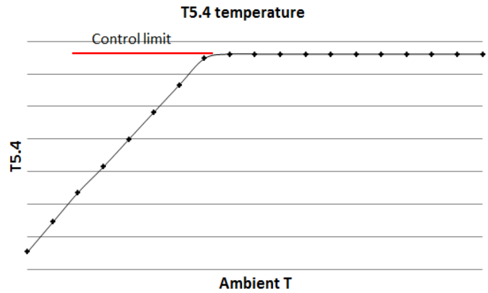

Fig. 5 T5.4, function of ambient temperature.

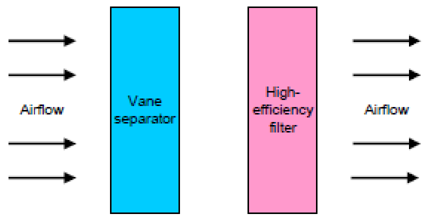

Fig. 6 Layout of the gas turbine intake air system.

for water that reaches the filters to be drained out on the upstream side of the filter elements. These filters are the final barrier before the airflow enters the gas turbine compressor and are promoted with an M6 rating in accordance with the European EN-779 standard [6].

The gas turbines have 30 filter elements arranged in five rows of six elements for combustion air and an additional row of six filter elements for cooling air to the gas turbine enclosure.

4. Water Wash System

Water supply is taken from platform fresh water distribution, which is produced from seawater by evaporators. Further, the water is led through a set of DI-filters (de-ionization or de-mineralization) and a final particle filter before entering the water wash skid, in order to achieve the OEM water quality specification for online water wash. The layout is shown in Fig. 7.

The water wash skid has two tanks (each with a 200 liter capacity), one tank for clean water (used for online wash and offline cleaning after crank soak

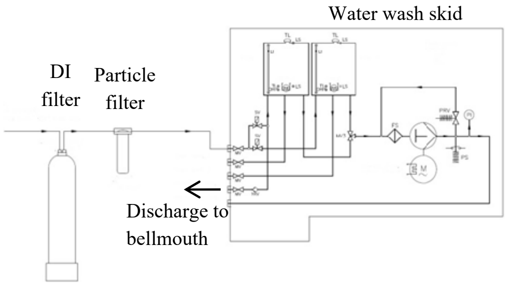

Fig. 7 Layout of the water wash system.

wash) and one tank for detergent/water mix (only used for offline wash). Thus, the risk of contamination of the clean water tank is avoided. The water is further pre-heated to 60 °C, before ingestion to the engine from the pressure outlet connection to the water wash nozzles assembled at the engine's bell mouth. One water wash skid is typically used for several engines in the same platform module; hence a sequence of logic for solenoid valves, heaters and pump is programmed into the unit's PLC. The unit is operated semi-automatically (local manual trigging of the sequence), in order to minimize the time consumption for platform operators to operate the system. A fully integrated system is also possible with the platform process control system, i.e., one which enables remote system operation from the platform control room, or by a programmed timer function. But a fully integrated system adds complexity to the system, as well as the risk of malfunction, and the operators lose hands-on operation of the system.

5. Engine Respons to Deterioration and Water Ingestion

The control modes discussed previously in this paper are valid when the engine is in new condition (i.e., overhauled/after offline water wash cleaning). When the compressor section becomes deteriorated by deposits, the control modes will change. Engines typically compensate for a deteriorated compressor by increasing *N*1-speed to keep *N*2-speed/power output at the same level. In particular engines initially running in *T*5.4-control (*T*5.4 maximum), will climb up against the *N*1-control limit. Thus the N1-control limit can occur before *T*5.4 maximum. This is typical when engines are not operated with online water wash. Thus, the target for online wash is to avoid such cases and run at *T*5.4-control limit continuously. However, if an *N*1-control limit occurs, some power margin can normally be gained by adjusting the VSV schedule to reach the maximum *T*5.4 limit again (if the VSV schedule is close to the overspeed side). This requires an adjustment of the VSV micro-adjust (or bracket in some cases) which is an on-engine manual adjustment (for base SAC engines). Thus, this operation requires ramping down to idle speed, or shutting down the engine, in order to enter the turbine enclosure compartment, which implies downtime. If the engine is stopped, an offline wash can be done simultaneously. If the engine is not stopped, the VSV adjustment is just a temporary compensation until the next offline water wash.

Risks to be considered when running online wash are:

- Cooled HPC casing, risk of rub;
- Flame out;
- Icing at HPC inlet (cold ambient conditions);
- Long-term effects corrosion/erosion;
- Staging/burning mode change (DLE engines).

Four cases with actual operating data are analyzed:

- Case #1: low water rate/part-load operation;
- Case #2: low water rate/peak-load operation;
- Case #3: high water rate/part-load operation;
- Case #4: high water rate/peak-load operation.

Two sets of water wash nozzles are used, one for low rate and one for high rate. High water rate is 30 liter/min which yields approximate 0.75% water-to-air ratio (by mass fraction) at engine base load. Corresponding ratio at low water rate is approximate 0.45%, by water rate of 18 liter/min. The water rate is fixed through the pump/nozzle configuration; hence, the water-to-air ratio will increase at part-load operation. Thus, the highest water-to-air ratio at approximate 1.00% is for case #3.

Discussion

The operational experience with online wash of LM2500 Base SAC engines at these water rates has not shown any disadvantageous impact on the operation of the engine/train. The transient conditions described in the analyzed cases, are handled well by the control system and do not influence the output $N2$-speed or power demand in a manner that poses a risk for the operation (e.g., engine trip). When water ingestion is shut off, the engine stabilizes at the initial condition as before the online wash sequence. Online wash is operated without any restrictions (e.g., fixed parameters) in the control system. However, as a precaution, the trip level for $T5.4$ has been increased, to avoid trips in transient conditions during online water wash, in particular when the water is shut off.

$N1$-speed tends to increase during the wash sequence, ref. Fig. 8 below, but stabilizes close to initial condition after the water is shut off. The same transient pattern is observed for all four cases (Figs. 9-20); $T3$ decreases, $PS3$ increases slightly and $T5.4$ decreases during water ingestion. The higher the water ratio, the higher the temperature drops of $T3$ and $T5.4$. The $T3$ and $T5.4$ temperature drop is in accordance with the thermodynamic theory, since the energy to evaporate the water is extracted from the air through the compressor. Thus, the $PS3$ increases due to higher mass flow through the compressor. When engine load is high, the impact of temperature-drop (particularly $T3$) is less than in part-load operation. When the engine is running at $T5.4$ control ($T5.4$ maximum) prior to water ingestion (case #2), and the $T5.4$ drops during water wash, the control system demands more power from the turbine, hence, $T5.4$ will increase slightly to compensate for the initial drop.

6. Process Simulations

A process simulation model of the LM2500 compressor is established in Aspen HYSYS, with humid air inlet to simulate real inlet air conditions offshore, as well as added water for online washing. The process simulation analysis is performed to validate the compressor response in order to compute the response when the water-to-air ratio is further increased (up to 3%, future target based on previous research [1-3]) and inlet conditions vary ($T2$, $P2$, RH). The analysis is even more important for DLE engines, with a more sophisticated fuel and control system, thus the risk of distribution of the burning-modes/staging during water ingestion.

Each of the empirical cases in the previous section in this paper is simulated in steady state condition, in order to compare results of how the HPC discharge condition is affected during water ingestion. The EOS (equation of state) used is the SRK (Soavo-Redlich-Kwong), further, real air heat capacity ratio (κ) and correction factors are derived using the ASME and Schultz methods [7].

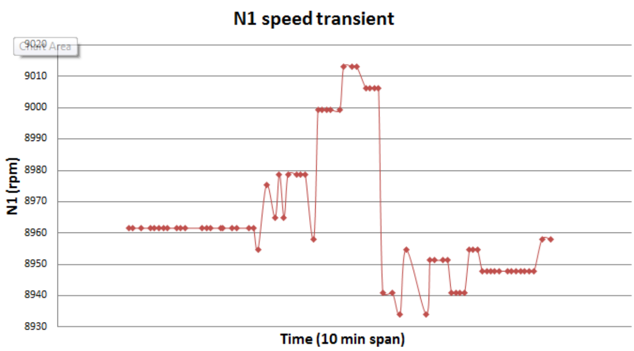

Fig. 8 Variation of N1-speed during water ingestion.

Case #1

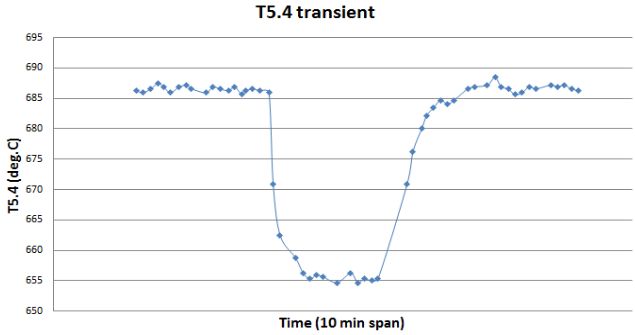

Fig. 9 Variation of T5.4 during water ingestion.

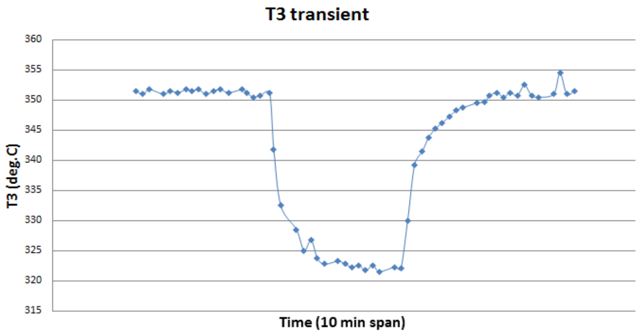

Fig. 10 Variation of T3 during water ingestion.

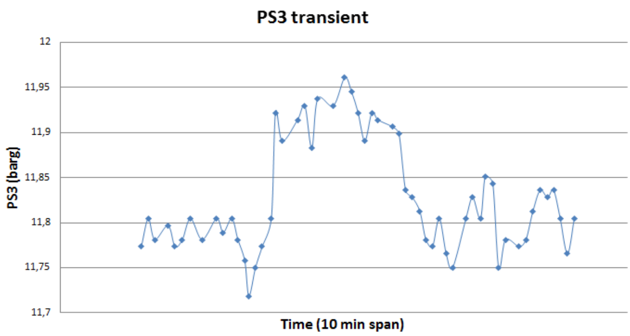

Fig. 11 Variation of PS3 during water ingestion.

Table 1 Engine parameters and water wash parameters.

Water injection	Engine data	Change
T: 60 °C	T2: 10.8 °C	T5.4: ~30 °C
P: 60 bar	P2: 1,001 mbar	T3: ~30 °C
Rate: 18 liter/min	RH: 87.6%	PS3: ~0.15 bar
Duration: 3 min	T3: 351 °C	
	PS3: 11.7 bar	
	T5.4: 686 °C	
	N1: 8,474 rpm	

Case #2

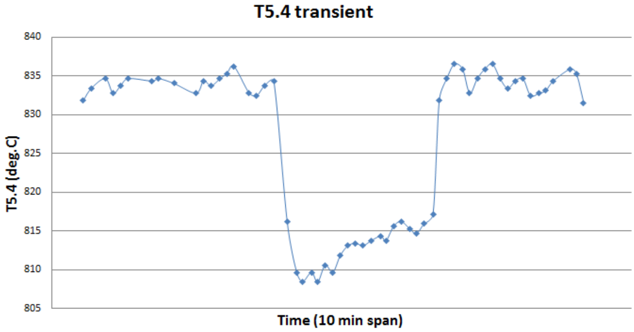

Fig. 12 Variation of T5.4 during water ingestion.

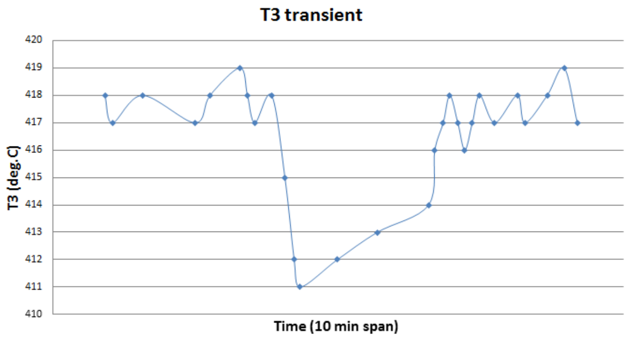

Fig. 13 Variation of T3 during water ingestion.

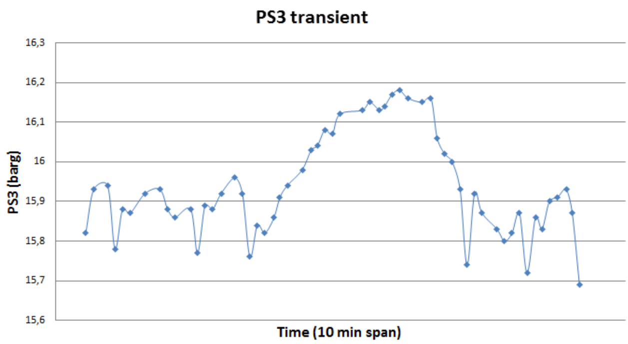

Fig. 14 Variation of PS3 during water ingestion.

Table 2 Engine parameters and water wash parameters.

Water injection	Engine data	Change
T: 60 °C	T2: 11.1 °C	T5.4: ~25 C°
P: 60 bar	P2: 978 mbar	T3: ~7 °C
Rate: 18 liter/min	RH: 78.1%	PS3: ~0.40 bar
Duration: 3 min	T3: 417 °C	
	PS3: 15.9 bar	
	T5.4: 834 °C	
	N1: 8,986 rpm	

Case #3

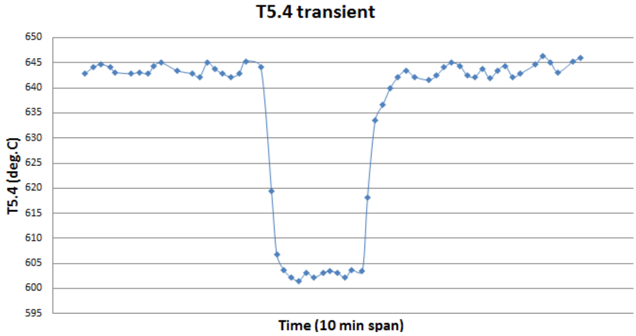

Fig. 15 Variation of T5.4 during water ingestion.

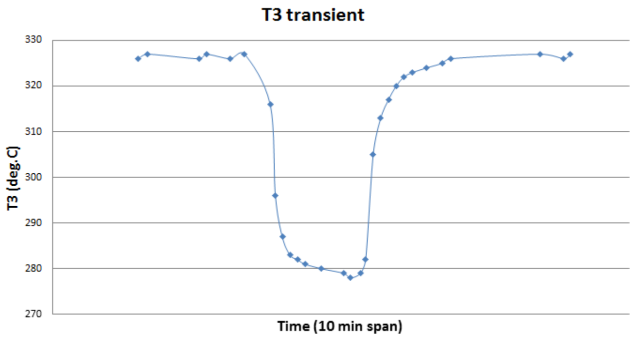

Fig. 16 Variation of T3 during water ingestion.

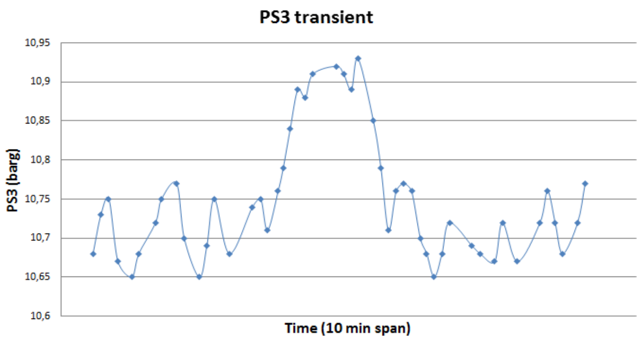

Fig. 17 Variation of PS3 during water ingestion.

Table 3 Engine parameters and water wash parameters.

Water injection	Engine data	Change
T: 60 °C	T2: 14.0 °C	T5.4: ~40 °C
P: 60 bar	P2: 1,000 mbar	T3: ~45 °C
Rate: 30 liter/min	RH: 74.3%	PS3: ~0.20 bar
Duration: 2 min	T3: 326 °C	
	PS3: 10.7 bar	
	T5.4: 645 °C	
	N1: 8,389 rpm	

Case #4

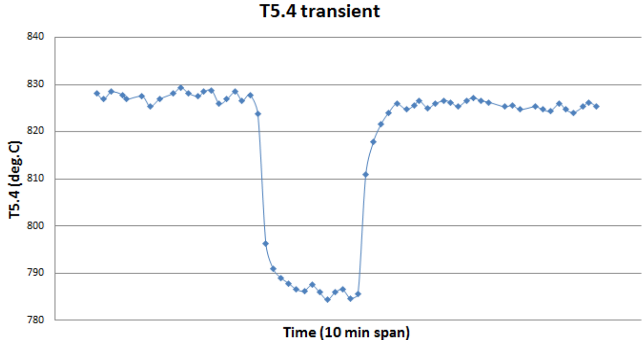

Fig. 18 Variation of T5.4 during water ingestion.

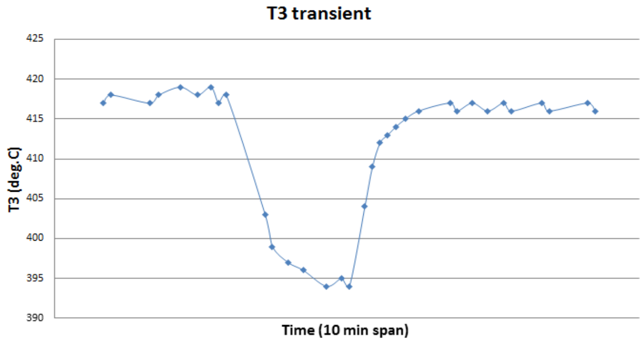

Fig. 19 Variation of T3 during water ingestion.

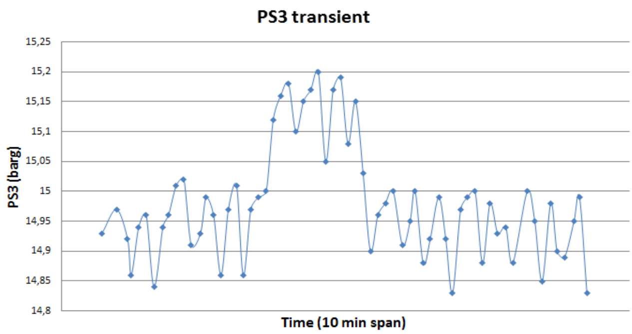

Fig. 20 Variation of PS3 during water ingestion.

Table 4 Engine parameters and water wash parameters.

Water injection	Engine data	Change
T: 60 °C	T2: 20.0 °C	T5.4: ~40 °C
P: 60 bar	P2: 1,006 mbar	T3: ~25 °C
Rate: 30 liter/min	RH: 93.8%	PS3: ~0.20 bar
Duration: 2 min	T3: 418 °C PS3: 14.9 bar T5.4: 828 °C N1: 8,986 rpm	

Summary of simulation results are shown in Table 5. An iterative method have been performed in order to optimize and tune the model to compensate for the added water and speed variations during the water sequence. A single performance point is established based on operating data for each cases, further, the performance point above and below (at fixed speed) is set based on experience and iterations to established a performance map around the operating point. The *T*3 simulated values are within 0.7% deviation of for all cases. *PS*3 values are slightly over-predicted in the simulations, except for case #3 (which has the highest water-to-air ratio).

Sensitivity Analysis

Compressor air flow is a calculated value based on the energy balance of the GG (iterative method), from the performance software used on the subject offshore field. In the simulations, inlet air humidity is assumed to be equal to ambient condition (in the absence of an inlet plenum RH measurement), hence the inlet air filter system (vane separators/filters) is assumed to remove large droplets and not change the humidity significantly. Furthermore, compressor head and efficiency are calculated using empirical data input to Aspen HYSYS in steady-state condition without water ingestion, and a compressor operating point/map is established. In the water ingestion simulations, the HPC discharge pressure and temperature are calculated values based on the given compressor map, thus an error in air flow and humidity implies uncertainties in the HPC discharge calculations. Furthermore, the operating point/map will change to some degree when water is added during water ingestion.

Since the above mentioned parameters introduce some uncertainties to the simulation model, a sensitivity analysis is performed for inlet air flow and humidity for case #1. A deviation of ±1% for inlet air flow and ±10% for inlet RH, is applied in the simulation model to determine the effect on HPC discharge condition (*T*3, *PS*3). As shown in Figs. 21 and 22, deviation of air flow implies the largest effects on HPC discharge condition, in particular for *T*3.

Table 5 Summary of simulation results.

Case	T3 sim. (°C)	T3 dev (%)	PS3 sim. (bar)	PS3 dev. (%)
1	320	-0.6%	13.5	+4.5%
2	409	-0.5%	17.8	+3.5%
3	282	+0.7%	11.4	-4.2%
4	394	-0.3%	16.5	+1.9%

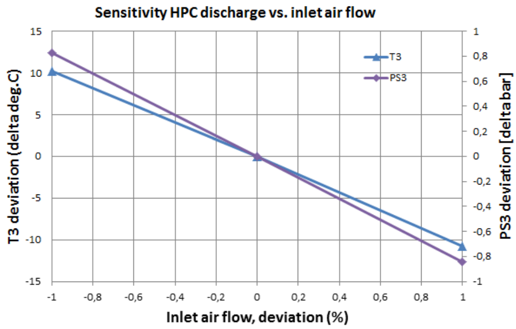

Fig. 21 Variation of T3/PS3 vs. air flow.

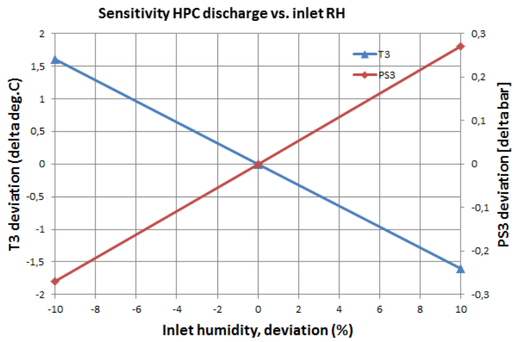

Fig. 22 Variation of T3/PS3 vs. air humidity.

7. Performance Trends/Analysis

In the performance software tools used in the subject offshore field, gas turbine compressor efficiencies are derived from the following relations:

Compressor Polytropic Efficiency

$$\eta_{.p} = \frac{\ln\left(\frac{T3i}{T2}\right)}{\ln\left(\frac{T3}{T2}\right)} = \frac{\ln\left[\left(\frac{P3}{P2}\right)^{\frac{(\kappa-1)}{\kappa}}\right]}{\ln\left(\frac{T3}{T2}\right)}$$

Compressor Isentropic Efficiency

$$\eta_{.i} = \frac{(h3_i - h2)}{(h3 - h2)}$$

Isentropic T3-value

$$T3_i = T2 \cdot \left(\frac{P3}{P2}\right)^{\frac{(\kappa - 1)}{\kappa}}$$

Isentropic enthalpie ($h3_i$) is derived as a function of the $T3_i$-value, furthermore, kappa (κ) for air is calculated using the mean temperature over the compressor ($T2$, $T3$).

The efficiency trends analyzed are based on relative change from a baseline, since the absolute value can vary from one engine to another and thus, are not of such interest. The baseline is defined as a clean engine after offline/crank wash, and is set at 0. Further, the HPC efficiency is ISO-corrected against $T2$. However, this type of ISO-correction does not result in any significant difference compared to the HPC efficiency without the correction, thus, this parameter is not very useful in this kind of analysis [8].

Long Term Trend Data

Fig. 23 shows the trend data for HPC efficiency from an engine operated without online wash. The degradation over an operational period of four months between offline washes is approximate 4.5% from the baseline. The efficiency gain of approximate 4% after offline wash can be clearly seen in the figure as well.

Fig. 24 shows the trend data for HPC efficiency from an engine operated with a daily online wash at low water rate over a four-month period between offline washes. The degradation is approximate 3%.

Fig. 25 shows the trend data for HPC efficiency from an engine operated with daily online wash at high water rate over a four-month period among offline washes. The degradation is approximate 2%. Again, the efficiency gain of approximate 2% after offline wash can be clearly seen in the figure.

These long-term trends document that a high water-to-air ratio is advantageous for achieving

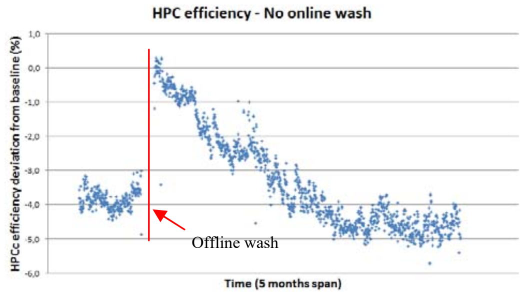

Fig. 23 HPC efficiency—no online wash.

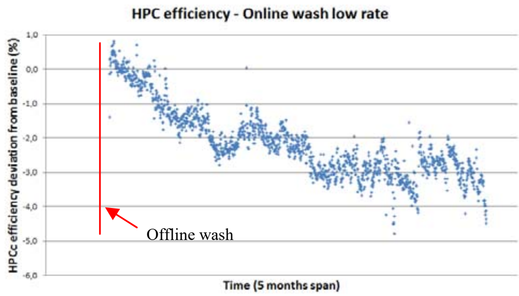

Fig. 24 HPC efficiency—online wash low rate.

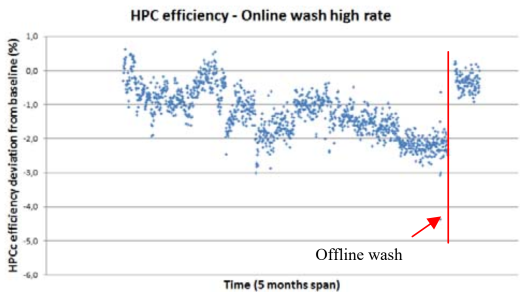

Fig. 25 HPC efficiency—online wash high rate.

compressor degradation as low as possible, hence, the potential to extend shut-down interval for offline/crank wash. This is in accordance with previous research [1-3]. However, not all operational periods (four-month intervals) yield conclusive and repeatable results. In particular when engine load and ambient conditions vary, it is challenging to evaluate compressor efficiency and obtain repeatable results between operation intervals. Load-independent correction factors/algorithms should be further investigated, as well as correction factors for bleed air from the compressor (e.g., anti-icing, rig air and engine cooling). The traditional ISO correction for standard ambient conditions is not sufficient for this type of performance analysis [8].

8. Instrumentation

Instrumentation on older offshore installations is often limited to the necessary instruments for machine control/protection, and additional instruments for effective performance monitoring and analysis are often missing or, if installed, have less accuracy.

LM2500 SAC engines are quite flexible regarding fuel flow/composition. These values are typically programmed once in the fuel controller during commissioning, thus, there is no requirement for measuring online gas-flow for fiscal quality, nor is there a need for a gas chromotograph/calorimeter to gauge online fuel-gas composition (as is required for DLE engines). Inlet depression as well as inlet plenum temperature, pressure and humidity are other typical parameters which are not available. Thus, inlet air flow calculation implies inaccuracy, which in the next phase implies inaccuracy for compressor head and efficiency calculations. In order to achieve more accurate air flow and compressor head calculations, it is recommended to have absolute pressure transmitters for $P2$ (mbar) and $PS3$ (bar), with high resolution and as short impulse tubing lines as possible (in addition to inlet depression measurement).

The above mentioned monitoring parameters are suggested for effective diagnosis of compressor degradation.

9. Economical Aspects

The economic potential of running online water wash is dependent on several factors, such as field layout, engine/train configuration, the number of engines running at peak load, whether the performance gain is utilized by longer intervals between maintenance stops (offline wash), or if the performance gain alone is utilized by keeping the same maintenance stop interval.

For the typical North Sea gas/condensate field analyzed in this paper, an annual production gain of some 50 MSm3 of gas and some 400,000 barrels of condensate can be achieved, just by increasing maintenance stop intervals from four to six months. With a current EU gas sale price of 40 US cents/Sm3 and USD 110/barrel of condensate, this implies some USD 64 million in annual savings. This is considered to be a conservative estimate, as maintenance cost savings (labor hours/spare parts) and daily production gains through higher power availability are not included in the calculations, nor are fuel and CO_2/NO_x tax savings.

10. Conclusions

Online water wash has shown great potential in keeping turbine performance high and increasing compressor efficiency. It is, therefore, possible to increase offline wash intervals without running the compressor into a severe fouling condition. However, great care has to be taken when choosing the water wash system design, particularly the water flow rate, but also the water temperature and pressure as well as the system design which must include proper freezing protection for North Sea conditions.

Inlet air filter systems, as well as operation and maintenance routines, are of equal importance in achieving the best possible results in keeping the engine clean.

The operational experience analyzed in this paper, documents that a high water flow rate (water-to-air ratio) is a key parameter to achieve increased power recovery and reduce long-term deterioration. However, running at too high rates may cause operational issues, as well as long-term effects such as erosion/corrosion to the engine components. At the water rates analyzed in this paper, no such negative long-term effects have been seen.

Finally, the correct monitoring parameters and performance software tools, as used to analyze the engines, are of vital importance to obtain the correct overview of engine condition and evaluate the effect of online washing.

11. Recommendations for Further Work

Process simulation software, such as Aspen HYSYS, is a very useful analytical tool in the evaluation and understanding of the fundamental mechanisms of axial compressor performance deterioration and recovery through online washing. The simulation model used for analysis in this paper, should be further developed to include the HPT and LPT turbines (and possibly the complete train), in order to obtain a better understanding for evaluating and optimize online water wash systems. In addition, each particular engine should be individually tuned in the simulation model.

Online water wash testing at higher water-to-air ratios (above 1%) is recommended in order to document the engine response and ascertain whether increased power recovery can be achieved without compromising engine integrity. But long-term effects such as erosion/corrosion issues due to the high water-rate, in particular hot corrosion (HPT blades/nozzles and combustion chamber), need to be further evaluated. Load-independent correction factors/algorithms should be further investigated, as well as correction factors for bleed air from the compressor (e.g., anti-icing, rig air and engine cooling).

Air inlet filter systems should be further evaluated and optimized. An upgrade to a higher filter class (F7 rating), is currently being evaluated for the subject offshore field analyzed in this paper, with future plans for field testing and comparison with the current M6 filter class.

Acknowledgments

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Appendix VIII

Gas Turbine Fouling Offshore; Effective Online Water Wash Through High Water-to-Air Ratio
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