NTNU
Norwegian University of
Science and Technology

Gas Turbine Optimum Operation

Synnøve Mangerud Flesland

Master of Science in Product Design and Manufacturing
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Supervisor: Lars Erik Bakken, EPT
Co-supervisor: Stian-Mikael Madsen, Statoil
Olaf Brekke, Statoil

Norwegian University of Science and Technology
Department of Energy and Process Engineering

Problem Description

1. Validere TurboWatch rutiner for oppfølging av degradering av gassturbinen. Kompressordelen vektlegges spesielt.
2.Analysere og dokumentere degradering av kompressordelen til maskiner i drift. Valgte maskiner bør ha samme last og ulike vedlikeholdstiltak og vannvaskeintervaller.
3.Analysere og dokumentere endringer og sammenhenger mellom kompressor utløpstrykk (CDP), akseleffekt og virkningsgrad. Forholdet vil muliggjøre bedre nøyaktighet i beregninger av økonomi, optimaliseringsbehov og ustlipp.

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MASTEROPPGAVE

for

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Optimal drift av gassturbin
Gas Turbine Optimum Operation

Bakgrunn

Gassturbiner står i dag for en vesentlig andel av verdens kraftforsyning. Nye anlegg både på land og offshore møter høye krav til ytelser og miljø. Forholdet inkluderer pålitelighet og regularitet, samt stringente krav til miljøutslipp av NO_x og CO_2.

I de senere år er det spesielt fokusert på utslippene av CO_2 og dens innvirkning på global oppvarming. Optimal ytelse, både relatert til design og drift er derfor avgjørende. Ved drift av gassturbiner vil ytelsen gradvis degraderes. Spesielt gjelder det kompressordelen av maskinen hvor belegg på rotor- og statorskovler er dominerende. Degraderingen reduserer anleggets effektivitet og øker miljøutslippene.

For å redusere utslippene av miljøgasser er det et stort incitament i å begrense degraderingen av gassturbinen og gjenvinne ytelsen, samt dokumentere teknisk underlag for optimal drift og vedlikehold. Herunder inngår erfaringer fra anlegg som er utsatt for slitasje og degraderingseffekter på selve maskinen og tilhørende systemer.

Mål

Oppgavens mål er å kartlegge og dokumentere degradering av gassturbiner med tilhørende inntakssystemer, samt dokumentere teknisk underlag for overvåking. En sentral del av oppgaven er relatert til kompressordegradering og systemer for gjenvinning av ytelsen. Prioriterte GE-LM2500 på Sleipner benyttes i analysene. Oppgaven gjennomføres i samarbeid med Statoil.

Oppgaven bearbeides ut fra følgende punkter

1. Validere TurboWatch rutiner for oppfølging av degradering av gassturbinen. Kompressoren vektlegges spesielt.
2. Analysere og dokumentere degradering av kompressordelen til maskiner i drift. Valgte maskiner bør ha samme last og ulike vedlikeholdstiltak og vannvaskeintervaller.
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" - "

Senest 14 dager etter utlevering av oppgaven skal kandidaten levere/sende instituttet en detaljert fremdrift- og eventuelt forsøksplan for oppgaven til evaluering og eventuelt diskusjon med faglig ansvarlig/veiledere. Detaljer ved eventuell utførelse av dataprogrammer skal avtales nærmere i samråd med faglig ansvarlig.

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NTNU, Institutt for energi- og prosessteknikk, 3. august 2010

Olav Bolland
Instituttleder

L. Bakken

Lars E Bakken
Faglig ansvarlig/veileder

Medveiledere;
Stian-Mikael Madsen, Statoil
Olaf Brekke, NTNU/Statoil

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Synnøve Mangerud Flesland

Abstract

Many offshore installations are dependent on power generated by gas turbines and a critical issue is that these experience performance deterioration over time. Performance deterioration causes reduced plant efficiency and power output as well as increased environmental emissions. It is therefore of highest importance to detect and control recoverable losses in order to reduce their effect. This thesis project was therefore initiated to evaluate parameters for detecting performance deterioration in addition to document different aspects of gas turbine degradation and performance recovery. Compressor fouling is the largest contributor to performance deterioration. Investigating fouling was therefore the main focus of this study.

In the present study the deterioration rates of four different gas turbines were evaluated. When choosing gas turbines it was emphasised to select gas turbines operating under equal conditions but with different washing procedures. In addition to offline washing two of the gas turbines had daily online washing routines and one of the gas turbines run idle wash every 1000 hour between each offline wash. Data was extracted from the monitoring software, TurboWatch, and loaded into Excel files. MATLAB scripts were created to handle the large amount of data and visualize performance trends. Series of two parameters were plotted against each other and the graphs were evaluated.

The evaluation showed that an overall trend was that the gas turbine that had been running with online washing continuously over a long period of time had higher performance than the reference engine. For the second gas turbine a daily online washing procedure has recently started. The advantage with the evaluation of this gas turbine was that a good reference engine was available. The two engines were operating under quite similar conditions at the same location in addition to having equal filter systems. Some deterioration trends were possible to detect. For the first period both engines seemed to have quite equal deterioration trends. During the second period no clear trends were seen in corrected CDP and corrected EGT when evaluated for constant GG speed. The compressor efficiency had decreasing trends for both engines during the second period as well, but the compressor efficiency for machine 1 was overall higher during the period with online washing than the previous period. The borescope pictures taken after the first period with online washing showed good visual results. However, it is too premature to make a final decision regarding the exact performance gain of online washing. At the time the study was performed the engine had only been running online washing for one operating interval, and more investigation over longer time is recommended.

For the engine running with idle wash it was not possible to conclude on the basis of the collected data. No clear deterioration trends were detected and investigations over longer time and several operating intervals are recommended. It is also important to be aware of the fact that the performance gain of idle wash needs to be much higher than for online washing in order for idle wash to be economically profitable.

There are several uncertainties related to performance trends. These include inaccuracy in instrumentation, monitoring software, calibration etc. Due to the fact that all the gas turbines evaluated in this study only have standard instrumentation it caused additional uncertainty in the performance trends. One suggestion for further study is to initiate a test instrumented gas turbine into operation with sensors for measuring inlet pressure depression.

Abstract in Norwegian

Mange offshore installasjoner er avhengig av kraft som genereres av gassturbiner. Alle gassturbiner degraderes over tid, noe som bidrar til redusert anleggsvirkningsgrad og effekt i tillegg til økte miljøutslipp. Dette prosjektet ble utført for å evaluere parametre som kan påvise degradering, samt dokumentere ulike aspekter ved ytelsestap og metoder som kan gjenvinne tapt ytelse. Begroing (fouling) er den største bidragsyteren til redusert ytelse og er derfor hovedfokus i denne rapporten.

I denne rapporten er degraderingsraten for fire ulike gassturbiner evaluert. Ved valg av gassturbiner ble det vektlagt å finne gassturbiner som opererte under like forhold, men med ulike vaskerutiner. I tillegg til offline vasking hadde to av gassturbinene daglig online vasking og en av gassturbinene kjørte idle vasking hver 1000 time mellom hver offline vask. Data ble hentet fra programvaren, TurboWatch, og lastet inn i Excel filer. MATLAB ble brukt til å enkelt kunne behandle den store datamengden samt visualisere trender for ytelse. Serier med to parametre ble plottet mot hverandre, og grafene ble deretter evaluert.

Analysen viste at en generell trend var at gassturbinen, som hadde kjørt online vannvask kontinuerlig over en lengre periode, hadde høyere ytelse enn referansemaskinen. For den andre gassturbinen hadde online vasking nettopp startet. Fordelen med denne gassturbinen var at en god referansemaskin var tilgjengelig. De to maskinene opererer under like forhold, er lokalisert på samme sted i tillegg til at de har identiske filter system. Enkelte degraderingstrender ble påvist. For den første perioden hadde begge maskinene relativt like degraderingstendenser. For den andre perioden ble det ikke observert noen klare trender i korrigert kompressor utløpstrykk (CDP) eller korrigert eksosgasstemperatur (EGT) for noen av gassturbinene. Kompressor virkningsgrad avtok for begge maskinene i løpet av den andre driftsperioden, men kompressor virkningsgraden for maskin 1 var generelt høyere for driftsperioden med online vasking enn perioden før. Borecope bilder tatt etter driftsperioden med online vasking viste også gode synlige resultater. Det er i midlertidig for tidlig å konkludere med det eksakte utbytte av online vasking. Når dette studiet ble gjennomført, hadde online vasking kun blitt kjørt for en driftsperiode, så videre utredelse over lengre tid anbefales.

For maskinen som ble kjørt med idle vasking var det ikke mulig å konkludere på bakgrunn av innsamlede data. Ingen tydelige degraderingstrender ble påvist, og videre utforskning over flere driftsintervaller anbefales også for denne maskinen. Det er også viktig å være klar over at gevinsten i ytelse for idle vasking må være større enn gevinsten for online vasking for at idle vasking skal være økonomisk lønnsomt.

Det er flere usikkerheter knyttet til ytelsestrender. Disse inkluderer unøyaktighet i instrumentering, programvare, kalibrering osv. Alle gassturbinene evaluert i dette studiet har kun standard instrumentering, og et forslag for videre arbeid er å installere en testinstrumentert gassturbin i drift, for eksempel med sensorer som måler trykktap over innløp.

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Nomenclature

Symbol	*Descriptions*	*Unit*
h	Enthalpy	[kJ/kg]
$\dot{m}$	Mass flow	[kg/s]
N	Shaft speed	[rpm]
P	Power	[W]
p	Pressure	[bar]
pr	Pressure Ratio	[-]
s	Entrophy	[kJ/kgK]
T	Temperature	[K]
X	Random value	[-]

Greek Symbols

Symbol	*Descriptions*	*Unit*
Θ	Temperature correction factor	-
δ	Pressure correction factor	-
κ	Adiabactic exponent	-
n	Polytropic exponent	-
η	Efficiency	-

Accronyms

Symbol	*Descriptions*
Machine 1	Main power generator 1
Machine 2	Main power generator 2
Machine 3	Main power generator 3
Machine 4	Mechanical drive 4
ASME	American Society of Mechanical Engineers
CDP	Compressor discharge pressure
DI	Deionization filter
DOD	Domestic object damage
EGT	Exhaust gas temperature
FOD	Foreign object damage
GE	General Electrics
GG	Gas generator
HP	High pressure
HPC	High pressure compressor
HPT	High pressure turbine
HR	Heat rate
LP	Low pressure
LPT	Low pressure turbine
NTNU	Norwegian University of Science and Technology
PT	Power turbine
SPPT	Shaft Power Power Turbine

Software	*Description*
MATLAB	MATrix LABoratory – data handling
Microsoft Office Excel	Spreadsheet application
TIKS	Gas Turbine Monitoring
TurboWatch	Gas Turbine Monitoring

Subscripts

Symbol	*Descriptions*
a	Air
c	Corrected parameter
comp	Compressor
comb	Combustor chamber
exp	Expected value
f	Fuel
i	Isentropic
p	Polytropic
r	Relative value
ref	Reference value
S	Shaft
T	Total conditions
t	Turbine
th	Thermal

Superscript

Symbol	*Descriptions*
a	Exponent for temperature correction factor
b	Exponent for pressure correction factor

Gas turbine Numbering System

Symbol	*Descriptions*
0	Ambient condition
2	Inlet compressor section
3	Outlet compressor section
4	Outlet high pressure turbine
54	Outlet gas generator/ inlet power turbine
8	Outlet power turbine

1 Introduction

1.1 Background

Offshore installations are dependent on power and most installations get their power supplied by gas turbines. New plants, both onshore and offshore, need to meet high requirements regarding steady performance and environmental emissions. Reliability and regularity is therefore an important focus area in order to maximize the efficiency and the profit. The focus on CO_2 emission and its environmental impact has increased over the last years. Optimized performance, both regarding operation and design is therefore crucial.

All gas turbines experience performance loss over time. Performance deterioration results in reduced plant efficiency and power output in addition to increased fuel consumption. Increased fuel consumption leads to higher operating costs and increased emissions.

Compressor fouling constitutes the greatest part of performance deterioration in gas turbines. Fouling is caused by air contaminants adhering to the internal surface of the compressor section, and the rate of fouling is highly site specific. Filter systems are installed at the gas turbine inlet to protect the engine from contaminants from the surrounding air and to decrease the deterioration rate.

The outer environment at offshore installations is complex and causes great challenges for the inlet filtration systems and the design of these. The filter systems are exposed to, and therefore need to handle high levels of humidity, salt particles, sand and dust from drilling in addition to exhaust gas. The filter systems are also expected to operate in dry, humid and freezing conditions.

In order to reduce the emissions to the environment it is an incentive in limiting the deterioration rate and regaining the performance. Fouling is classified as a recoverable loss, and can be regained by wet cleaning i.e. by injection of fluid into the engine intake. There are currently two different methods for wet compressor wash; online and offline washing.

1.2 Scope of work

The scope of this study was to map and document deterioration rates for gas turbines in order to investigate the potential for improving efficiency rates through online and idle washing by monitoring several parameters. A literature study was initially performed in order to get an overview of the state-of-art understanding of deterioration mechanisms.

The Sleipner field was a selected case. The Sleipner field has 11 gas turbines in operation which are providing the field with power. All the gas turbines are in the LM 2500 PE series. Offline wash is performed approximately every 3000 hours on all the gas turbines. In addition machine 1 and 3 run daily online washing and machine 4 run idle wash approximately every 1000 hours between each offline wash.

The four gas turbines were selected together with supervisors and after request from Statoil. When choosing the engines emphasis was put on comparing equal engines operating under equal conditions but with different washing procedures.

Methods used to detect the performance losses are discussed. Due to the great variation in ambient conditions correcting methods have been applied to perform evaluation on the gas turbines on equal terms. The monitoring software, TurboWatch, is further validated with focus on the routines for monitoring compressor degradation and water washing. Calculations and equations applied in the software are evaluated and discussed in order to get an understanding of the mathematical relations on which TurboWatch is based on.

1.3 Report structure

Chapter 2 briefly presents basic gas turbine theory.

Chapter 3 presents gas turbine performance deterioration, the offshore environment and discuss which parameters can be used for detecting compressor fouling.

Chapter 4 presents filter systems used in order to decrease the deterioration rate. Different washing methods are also discussed.

Chapter 5 gives a brief overview of data handling and processing. Correction formulas are discussed, and a description of how the data is processed is also included.

Chapter 6 presents the validation of the monitoring software TurboWatch.

Chapter 7 includes the deterioration trends for four selected gas turbines at Sleipner.

Chapter 8 briefly discusses the effect of performance deterioration.

Chapter 9 discusses the overall results from the report.

Chapter 10 gives a conclusion of the thesis.

Chapter 11 presents suggestions for future work.

2 Gas turbines

The underlying section gives a brief description to basic gas turbine theory, including gas turbines sensors.

2.1 Gas turbines

Gas turbines are one of the main power suppliers at offshore installations. All the gas turbines investigated in this study are in the General Electric LM 2500 PE series and are two-shafted simple cycle gas turbines as shown in figure 2-2.

The function of a gas turbine is dependent on many components. A two-shaft gas turbine consists of an air compressor, a combustor, a gas generator turbine and a power turbine. The gas generator in the LM2500 consists of an axial compressor with 16 stages, a combustion chamber and a 2 stage high pressure turbine. The compressor section consists of several airfoils circumferentially positioned on a rotor which is driven by the high pressure turbine (HPT). The power turbine (PT) is aerodynamically coupled to the gas generator, and consists of a low pressure turbine with 6 stages.

The compressor generates air at high pressure. The air is fed into the combustor chamber where the fuel is burned. The combustion products and excess air leaves the combustor at high pressure and temperature. The gas is further expended in the high pressure turbine in order to run the compressor. The gas leaving the high pressure turbine still has high pressure and temperature, and is further expanded trough the low pressure turbine (LPT). The power turbine is connected to the driven equipment which can be a compressor, a pump or a generator.

The gas generator is controlled by the amount of fuel supplied to the compressor. The firing temperature and the maximum shaft speed are the gas generators operating constraints. If the fuel flow increases both firing temperature and shaft speed will increase until one of the limits is reached. If both the limits are reached simultaneously it is referred to as match temperature. The speed limit is first reached and thereby the limiting factor when the ambient temperatures are above the match temperature. When the ambient temperature is below the match temperature the firing temperature is the limiting factor [16].

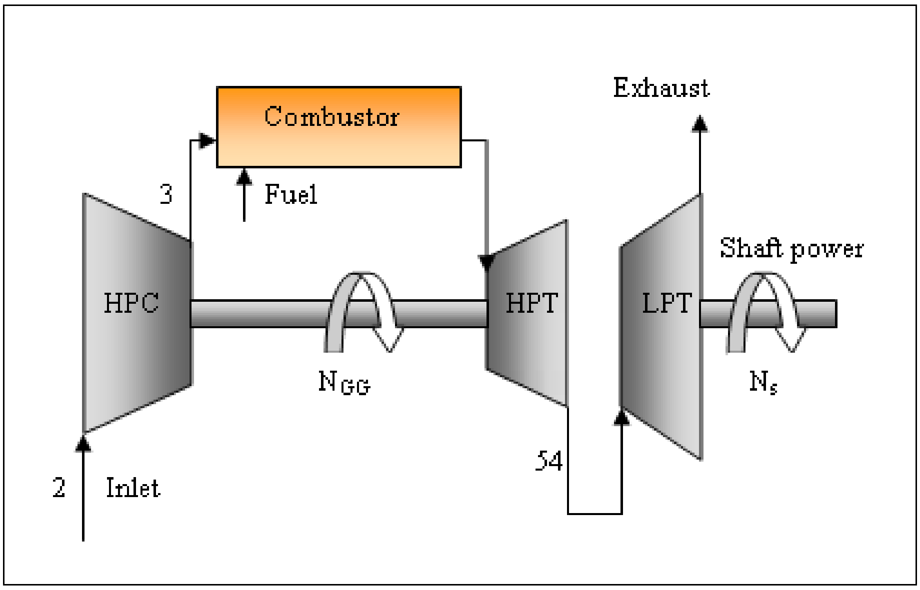


Figure 2-1: Two shafted gas turbine with sensor placement

2.2 Gas turbine sensors

All the LM 2500 gas turbines evaluated in this study are equipped with standard engine instrumentation supplied from the manufacturer. According to Krampf, [14], standard instrumentation is installed for equipment protection and more instrumentation is needed for performing detailed analysis. Several sensors are measuring the different parameters from different locations. The location of the main sensors used in this study is shown in figure 2-2 and are discussed in the following.

Pressure sensors
The sensor measuring ambient pressure, p_0, is not included in figure 2-2. This sensor is often located under the helideck. Usually ambient pressure is a commonly measured parameter for all gas turbines at an installation as variation in ambient pressure influences the gas turbine performance and is thereby an important parameter.

Compressor inlet pressure, p_2, and compressor outlet pressure, p_3, are measured locally for each gas turbine. p_2 is measured in the bellmouth before the first rotor stage and p_3 is measured after the last compressor stage before the combustion chamber. The gas turbines in the present study have separate sensors measuring p_2 and p_3.

The pressure between the high-pressure and the low-pressure turbine, p_{54}, is also measured by one sensor.

Temperature sensors
The sensor measuring ambient temperature, T_0, is not shown in the figure above, but is measured from the same location as ambient pressure. As for ambient pressure, ambient temperature is also normally a commonly measured parameter for all gas turbines at a plant. The gas turbines power output varies with varying ambient temperature making T_0 an important parameter.

The compressor inlet temperature, T_2, is measured by one sensor at the same location as p_2. The outlet compressor temperature, T_3, is also measured by one sensor and at the same location as $p_{3.}$ Both T_2 and T_3 are measured by one sensor for each of the gas turbine.

The temperature measure between the high pressure turbine and the low pressure turbine, T54, is measured by 8 sensors spread around the casing. This temperature is used to calculate the gas temperature exciting the exhaust chamber, EGT. If this temperature is too high it can easily lead to damage in the turbine nozzles and rotor blades, and is thereby a critical parameter.

Shaft speed sensors
Two different speed sensors are measuring the gas generator speed, N_{GG}, and the power turbine shaft speed, N_s. The location of these two sensors is shown in figure 2-2.

Other sensors
The location of the fuel flow sensor is shown in figure 2-2. Fuel flow measurements were not available for the gas turbines investigated in this study.

Sensors measuring relative humidity are available at some installations. However, this is not the case at field investigated in this study.

3 Gas turbine performance deterioration

All gas turbines deteriorate in performance over time. All the components in a gas turbine, in particular the aerodynamic components, will invariably degrade their performance. The deterioration rate depends on the atmospheric conditions and the manner in which the engine is run. Performance losses are divided into two categories; non-recoverable and recoverable losses. Non-recoverable losses, like erosion, corrosion and increased tip clearance, are results of mechanical problems and cause damage to the airfoils in the gas turbine. Recoverable losses are primarily the result of compressor fouling and can be regained by cleaning the engine.

This chapter focuses on the different deterioration losses, site specific conditions and common parameters used to detect performance deterioration.

3.1 Non recoverable deterioration

Non recoverable losses require replacement of parts or an engine overhaul in order to regain the performance [9]. Corrosion, erosion and increased tip clearance are examples of non recoverable deterioration losses that will cause damage to the air foils in the gas turbine.

Erosion is caused by hard particles, e.g. sand and dust. These particles hit the compressor blades and lead to mechanical material damage on these [18]. The damage includes changes in the airfoil shape, changes in the size of the flow area and increased blade and tip clearance. The result is increased losses and thereby decreased performance. Erosion is normally caused by articles greater than 10-20 μm, [2][15]. For industrial gas turbines particles this size are usually stopped by an inlet filtration systems which greatly decreases the erosion rate.

Corrosion is caused by chemical reactions between contaminants entering the gas turbines and flow path components. Moist salt particles, mineral acids and reactive gases e.g. chlorine and sulphur oxide can in combination with water lead to wet corrosion [9]. The blades roughness increases which leads to reduced gas turbine performance. Coating is commonly applied to the blades in order to decrease the corrosion rate and prevent deterioration [18]. Corrosion is particularly when exposed to salt water, and is therefore a problem for gas turbines operating offshore.

Increased tip clearance is also a typical non-recoverable loss. Increased tip clearance leads to increased leakage flows and thereby decreased stage efficiency and reduced head.

Damage can also be caused by large foreign objects entering the gas turbine through the inlet. For aircraft engines with no inlet filtration system it is possible for objects to enter through the gas turbine inlet. The objects are divided between foreign objects (FOD) and domestic objects (DOD). Foreign objects are any object entering the engine with the inlet air e.g. birds or lumps of ice. Domestic objects are components breaking off from the engine it self and then carried downstream.

3.2 Recoverable deterioration

Recoverable deterioration is degradation mechanisms that can be reversed. Losses due to fouling can normally be regained by water cleaning.

3.2.1 Fouling

Compressor fouling constitutes 70-85% of the performance loss caused by deterioration in gas turbines [9]. Fouling is caused by contaminants from the inlet airflow which adheres to the air foils and the internal surfaces. Fouling can occur in all compressors and the rate of fouling is dependent of several factors; compressor design, airfoil design and shape, level and amount of contaminants and ambient conditions. The majority of the contaminants causing fouling is smaller then 2μm [15].

The atmospheric air ingested into the gas turbine is not pure, especially not for gas turbines situated at offshore installations. The offshore environment consists of oil and water moist, salt particles and various hydrocarbons which all contribute to the build-up of material, causing increased surface roughness and changes in the airfoil shape. Prior investigations done by Brekke, Bakken and Syverud, [25], showed that sodium based salts are the dominant contaminant found in the compressor section.

A compressor exposed to fouling has deteriorated aerodynamic qualities. Fouling leads to reduced airflow through the engine and reduced compressor efficiency. Compressor fouling can also cause reduced surge margin which may result in compressor surge,[9][27][15]. Due to decreased airflow through the turbine the pressure ratio will decrease. Fouling leads to decreased gas turbine output and increased heat rate. Fouling also contributes to increased fuel consumption and increased environmental emissions.

3.3 The offshore environment

The deterioration rate is site specific, and the offshore conditions are more complex compared to the conditions at onshore installations. Ambient conditions offshore include high levels of humidity and salt particles, hydrocarbons, drilling dust and particles from maintenance activities.

The challenging conditions offshore are causing high deterioration rates. Inlet filter systems are installed at the inlet of the gas turbines in order to decrease the amount of contaminants entering the engine. A careful and well planned production facility can also help reducing the contaminants. Placing the air inlet far away from the exhaust channels and other outlets is one possible precaution.

Sleipner has one weather station measuring the ambient conditions, so local differences may occur. In this report the ambient conditions are assumed to be the same for all the engines considered.

3.3.1 Ambient temperature

Figure 3-1 shows ambient temperature for a one year period starting the 1st of January 2009. In this period the recorded temperature range was from -1.3 to 20.8 degrees Celsius with an average of 9.8 degrees.

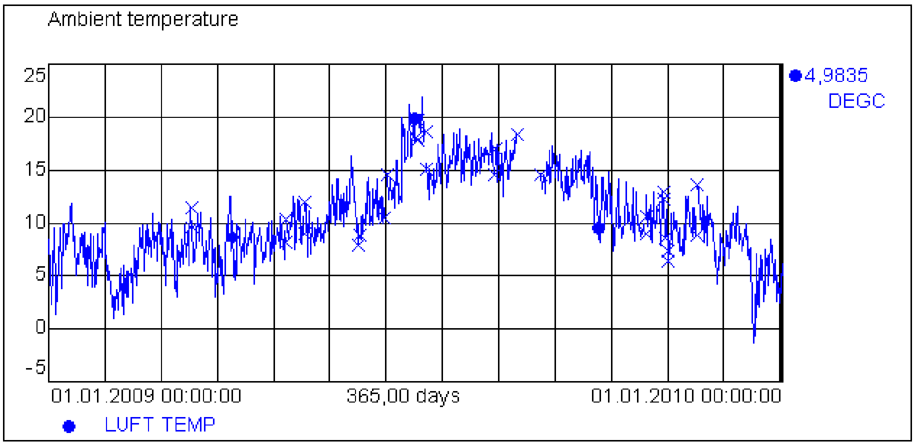


Figure 3-1: Ambient Temperature 01.01.2009-01.01.2010

3.3.2 Ambient pressure

Figure 3-2 shows the ambient pressure for a one year period, starting the 1st of January 2009. The average value is 970.9 mBara in a range from 961.6 to 1031.8 mBara.

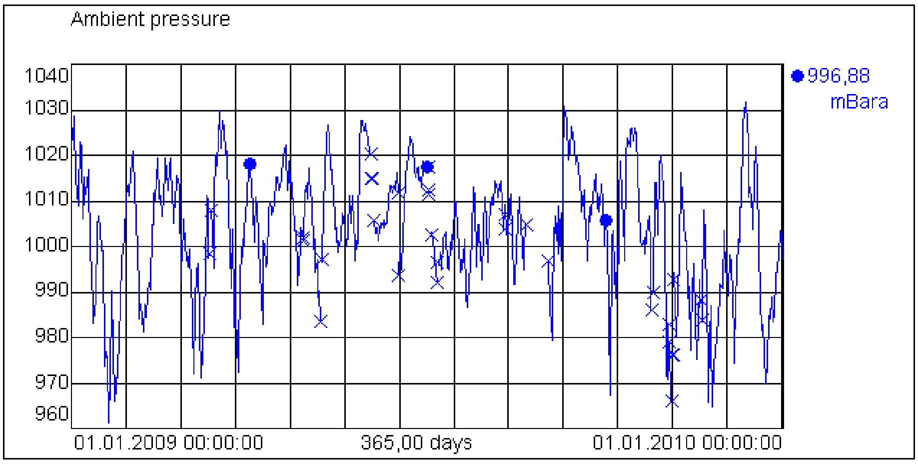


Figure 3-2: Ambient Pressure 01.01.2009-01.01.2010

3.3.3 Ambient humidity

Several platforms at the Norwegian shelf have experienced considerable losses in gas turbine performance when operating in foggy conditions. These losses were only recoverable by compressor cleaning. An investigation, [6], was performed for one of these incidents and it revealed that the foggy weather caused an increased pressure drop over the filter system followed by a reduction in the compressor efficiency. The suggested explanation was that the fog caused the filter elements to saturate which lead to increased pressure drop in addition to unloading of salts and other contaminants into the airflow downstream of the filter. The salts and contaminants entered the compressor section which caused reduced flow area and efficiency. This explanation is also supported by the fact that the performance only was recoverable by compressor cleaning, [6]. This shows that the level of humidity is an important factor to consider.

At the installation evaluated in this report the relative humidity is measured at ambient temperature and pressure. Due to losses over the inlet filter the ambient value does not fully represent the value at the compressor inlet. The ambient humidity is only measured an instant value, and trends are therefore difficult to trace.

3.4 Monitoring gas turbine performance

Maintenance of gas turbines offshore is usually either calendar based or condition based. The increasing demands for reduced emissions have lead to a higher focus on gas turbine performance in form of regularity and stable production. All these factors have increased focus on condition monitoring. This section will focus on different parameters which can be used to detect compressor fouling.

Fouling will reduce the overall performance of a gas turbine. However, variation in ambient conditions will also affect the engine performance and it is some times hard to separate which losses are due these changes and which losses are due to compressor fouling.

Different guide lines have been created in order to detect the level of fouling without having to shut down the engine and do a visual inspection. Engine manufacturers have developed different guide lines for detecting compressor fouling which all includes different parameters. Haq and Saravanamutto, [12], describe the ideal characteristics of a parameter when monitoring the condition of a compressors condition. The requirements for these characteristics are:

- It should not be complicated or time-consuming to collect data.
- External variables should not affect the characteristics
- Data interpretation should be easy and not depended on the operator's knowledge of turbo machinery.
- The parameters should give an accurate indicator of the compressor condition.

There is currently no existing parameter that takes all the above considerations in to account. However, different parameters are used to detect fouling. GE, [10], suggests several parameters for monitoring the condition of LM2500 engine compressor, and emphasises the importance of monitoring these parameters at gas generator speed of 9000 rpm. Different parameters used to detect compressor fouling, including the ones recommended by GE, are discussed in the following.

3.4.1 Parameters for detecting compressor fouling

3.4.1.1 Compressor Air Flow

The compressor air flow will decrease as a result of compressor fouling, [9][15][21]. Less airflow through the gas turbine leads to reduced power output. Compressor air flow can be used as a parameter to detect fouling when plotted against corrected gas generator speed, $N1_c$, or corrected exhaust gas temperature, EGT_c [2].

There are rarely on site measurements for compressor airflow. Compressor airflow is usually a calculated value in monitoring software. In order to calculate the airflow, measurements of total temperature (T_T), total pressure (p_T) and static pressure (p) at the bellmouth is needed. The airflow can then be found be using thermodynamic relations and the law for conservation of mass.

The lack of on site measurements and the fact that the calculated airflow is based on several parameters with their own uncertainties makes compressor airflow an uncertain parameter for monitoring compressor fouling.

3.4.1.2 Gas Turbine Power output

The reduced compressor airflow due to fouling leads to reduced gas turbine power output [21]. The power output could be measured directly at site by a torque meter. The gas turbines evaluated in this report don't have torque meters installed and the power output is only given as a calculated value. This calculation is based on several other parameters, making power output an unsuitable indicator of performance deterioration.

3.4.1.3 Pressure loss over at the Bellmouth

The reduction in compressor airflow due to fouling will lead to a pressure drop over the bellmouth assuming constant shaft speed. This pressure drop can be measured quite accurately by a simple manometer. Diakunchak, [9], recommends this parameter as one of three parameters used to detect compressor fouling. However, the engines investigated are not equipped with manometers at the bellmouth, and this parameter can therefore not be used to detect performance deterioration in this report.

3.4.1.4 Compressor Discharge Pressure (CDP)

Compressor fouling leads to decreased CDP due as a result of the reduction in mass flow and stage efficiency. CDP is a commonly measured parameter, and the advantage with using CDP is that only one measurement is needed to determine the value.

Haq and Saravanamutto, [12], found CDP to be most effective parameter for monitoring compressor condition for the engines evaluated in their study. CDP had a consistent downward trend when fouling progressed in Haq and Saravanamutto's study. Diakunchak, [9], also claims that CDP is a reliable parameter for detecting compressor fouling. However, Diakunchak, also acknowledge that one weakness with CDP as a monitoring parameters is the fact that CDP is depended on the losses over the inlet filtration systems.

3.4.1.5 Compressor Discharge Temperature (T3)

Compressor fouling leads to increased friction between the air and the compressor surfaces which causes higher discharge temperature from the compressor. However, the thermodynamic loss in temperature as result of higher delivery pressure works to neutralize this. The total change in T3 is therefore not very large.

Only one or two temperature sensors are measuring T3 making it challenging to get the exact measurement of this temperature. The gas turbines evaluated in this study only have one temperature sensor measuring T3. Saravanamutto and Haq, [12], concluded that T3 is an unsuitable monitoring parameter for monitoring performance deterioration.

3.4.1.6 Compressor Efficiency

The compressor efficiency is expected to decrease when fouling progresses, [9][12][15], and is for that reason expected to be a good indicator for monitoring compressor fouling. However, Saravanamuttoo and Haq, [12], claim that compressor efficiency not is a suitable parameter when monitoring the compressor conditions. They state that compressor fouling only leads to a small decrease in efficiency during an operating period and that trends therefore are difficult to obtain. A compressor exposed to fouling will normally have around 1 % decrease in efficiency throughout an operating period, which shows how marginal the efficiency changes are [29].

Efficiency is a calculated value based on several parameters, all with their own uncertainties. This indicates that the efficiency is an inaccurate parameter for detecting fouling. However, if compressor efficiency is plotted against corrected gas generator speed it can be interesting to see if any clear decreasing trend appears.

3.4.1.7 Exhaust gas temperature (EGT/T54)

Exhaust gas temperature is an arithmetic average of 8 measurements from 8 sensors spread around the casing. EGT is an important parameter to monitor, especially because a too high EGT can cause damage on the rotor and stator blades in the turbine part of the gas turbine. The condition of the combustor chamber and the turbine will affect the exhaust gas temperature making it an uncertain parameter.

EGT, if plotted against gas generator pressure ratio, is a recommended parameter by GE for monitoring the LM2500 engine compressor [10]. Saravanamutto and Haq, [12], discovered that EGT varied uniformly with ambient temperature when load and corrected gas generator speed was kept steady. However, they also discovered that when keeping load and GG speed steady the change in EGT was less than 1 % which is not enough for performance monitoring, especially not when measurements errors are taken into account.

3.4.1.8 Fuel Flow

It is expected that the fuel flow will decrease proportionally with the decreased airflow in a fouled engine. However, even with a fouled compressor the engine will still try to reach the expected power output. In order to compensate for the loss in performance due to fouling the engine will accelerate the shaft speed which causes increased fuel flow.

3.5 Summary

All gas turbines will loose performance over time. There are two different categories of deterioration losses; non recoverable and recoverable losses. Compressor fouling is the largest contributor to performance deterioration and is caused by contaminants entering the gas turbine and adhering to the surface area of the compressor. Compressor fouling is classified as a recoverable loss and can be regained by water washing.

The deterioration rate is site specific. The offshore environment is complex and includes high levels of humidity and salt particles, hydrocarbons, drilling dust and particles from other maintenance activities.

There are currently no standard rules or parameters that clearly predict deterioration rates. Several parameters for detecting performance deterioration were reviewed and briefly discussed in this chapter. Pressure loss over bellmouth has shown to be a good parameter and can be measured by a simple manometer. The gas turbines investigated in this study do not have manometers installed and pressure loss can not be used for performance monitoring. Gas turbine power output and compressor discharge temperature seemed like unfitted parameters and is not included in further analysis in this project. CDP was found a suitable parameter and will be used for further analysis in this report. Due to different opinions around several of the other parameters, EGT, compressor efficiency and airflow will be investigated further. None of the parameters can be used alone in order to detect performance trends. The parameters should be evaluated at similar conditions for example at constant gas generator speed or constant exhaust gas temperature.

4 Performance recovery

Due to increased focus on emission and operating costs, plant optimization has become important at offshore installations. The power demand is also increasing due to the fact that the reduced pressure in the oil and gas reservoirs over time is creating a need for increased head in the compressor stations. The increased power requirement forces the gas turbines to run at very high power levels. This will often cause the gas turbine to be the limiting factor for the production which makes regularity and stability a key element.

Effective filtration systems are installed at the gas turbine inlet to reduce the amount of contaminants entering the gas turbine in order to reduce the deterioration rate. However, some losses will always occur. Coating of the compressor blades will reduce the surface roughness and make the blades less susceptible for fouling. Coating is not very common at current gas turbines and water washing will be the focus area in this report.

The most common way of cleaning the compressor is by injecting fluid into the engine intake. There are currently two methods of water washing; offline and online washing.

4.1 Filter systems

The gas turbines operating in offshore conditions have great challenges with high rates of fouling, erosion and corrosion due to complex ambient conditions. All gas turbines operating as power generator units or compressor drivers in the Norwegian oil and gas industry are therefore equipped with inlet filter systems in order to reduce the deterioration rate. Current filtration systems usually prevent corrosion and erosion, but compressor fouling remains a challenge [8].

The complex environment offshore is causing challenges for the inlet filtration systems. The filtration systems at offshore installations are expected to operate in dry, humid and freezing conditions. Operating experience has shown that the deterioration rate is increasing in wet conditions and that that the pressure loss over the filter system is dependent on the level of ambient humidity [6].

A typical state-of-the-art filtration system is a static system with several stages [6]. When placing filtration systems at the inlet of a gas turbine, the air stream is exposed to more losses. The pressure loss over the filter system is causing reduced compressor suction pressure which again leads to reduced compressor discharge pressure.

Maintenance of gas turbines is always a focus area at offshore installations. However, maintenance procedures for filter sections do not always exist in the same manner. It is, however, great economic and environmental potential in assuring optimal performance of the filter systems offshore. There is also lack of an international standard to document and evaluate the performance of inlet filtration system. Most of the existing standards do not take salt removal and operations in wet and moist conditions in to account. Filter should not be changed according to only calendar and running-hour interval or pressure loss. Site specific air quality and gas turbine deterioration should be considered [6].

4.1.1 Filter systems at Sleipner

Sleipner has currently two different filter systems in operation on the gas turbines. Filter system A, illustrated in figure 4-1, have one set of vane separators located in front of the high efficiency filter. These filters are dependent on removing the majority of water and moist from the airflow before it enters the high efficiency filter. The high efficiency filter is specially designed to withstand moisture. Seals are installed on the bottom of each filter bag to make sure that the water reaching the filter elements are drained out upstream of the filter [8].

Figure 4-2 shows filter system B which have two sets of vane separators; before and after the high efficiency filter. The first stage of vane separators removes the majority of water and humidity from the airflow before it enters the high efficiency filters. The greater part of the pressure loss takes place over the high efficiency filters, and these filters determines the final filtration efficiency. The last vane separator is designed to remove the remaining water droplets from the airstream. This filter type also allow for the use of pre-filters, but pre-filters are normally not used on Norwegian oil platforms today [8].

The interval between filter changes at Sleipner is today 12 months, and the possibility of reducing the interval from 12 to 8 months has been discussed. However, it is a question about costs versus benefit.

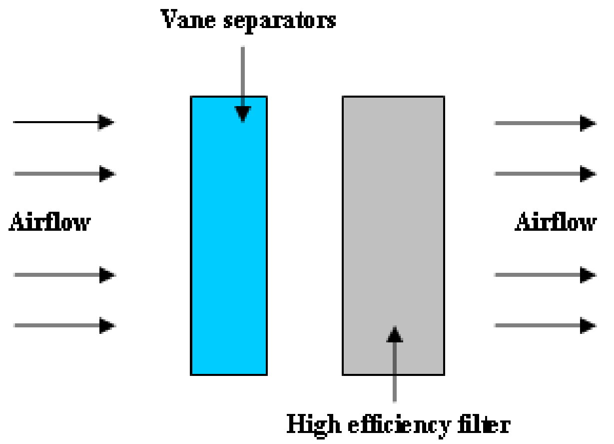


Figure 4-1: Filter system A

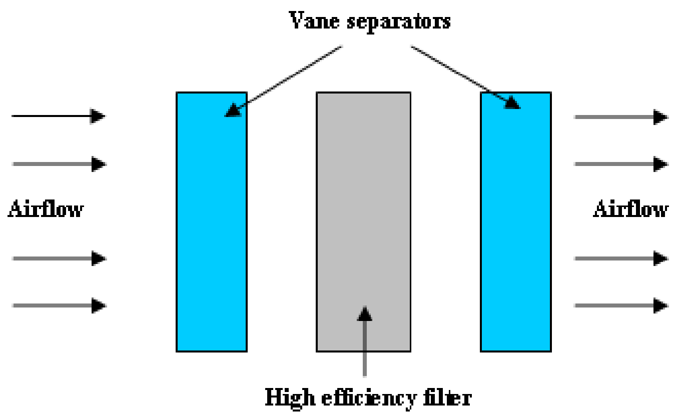


Figure 4-2: Filter systems B

4.2 Offline water wash

Offline washing, also called crank wash, is the most common method for compressor washing. In order to reduce thermal stresses in the engine, the engine needs to be cooled off before an offline washing sequence. A mixture of water and cleaning solution is then injected into the engine in order to dissolve the deposits adhered to the compressor blades. The water/cleaning solution is kept inside the engine for several minutes. Clean water is then injected in order to rinse the gas turbine. This process is repeated as many times as necessary. Offline washing is by far the most effective water wash, but a typical sequence last 3-4 hours and can cause 6-7 hours of lost production [26].

4.3 Online water wash

Online washing can be run at different loads. This report will focus on online washing at normal operating load in addition to idle washing. The advantage with online washing at normal operating load is that it does not affect the production. When performing online washing some of the water will evaporate inside the compressor due to the very high temperatures on the compressed air. Online washing at normal operating load is therefore most efficient at the first stages in the compressor section. Online washing is less effective than offline washing and can not fully replace offline washing. Normally online and offline washing are used in combination to keep the performance as high as possible and to increase the operating interval between each offline wash [26].

When performing idle washing the gas generator speed is run down to approximately 5000 rpm. If the engine works with redundancy the load can be transferred to a standby machine when running an idle wash sequence. Idle wash can not be favourable when compared to online at normal operating load if the engine does not work with redundancy.

In order to compare idle washing and online washing at normal operating load a complete operating interval need to be considered. Idle washing is more time demanding and is causing increased fuel consumption and lost production. These factors need to be taken into consideration when comparing the performance gain of idle washing with the gain of online washing at normal operating load.

4.3.1 Main parameters for online washing systems

There are different parameters to consider for an online washing system. Some of the most important issues are discussed in the following.

Water-to-air ratio

In order for an online washing sequence to have the desirable impact all the compressor blades need to be wetted by the injected water. This is influenced by the water-to-air ratio. According to Syverud, [22], water-to-air ratio is the most important parameter for performance recovery by online washing. The recommended water-to-air ratio is between 0.8% and 2%, and the lower flow rates can cause re-deposits of fouling in the last stages [22][23].

Droplet size

The droplet size is expected to be an important parameter in online washing. It is necessary that the droplets are able to wet all the compressor blades, including the last stages. If the droplets are too small it is likely that they will be deflected by the inlet guide vanes, and if the droplets are too big they might not be able to follow the airstream through the engine due to the influence from gravity and centrifugal forces from the rotor stages. Syverud and Bakken, [23], concluded that to small droplets increased the fouling in the after stages. According to Syverud, [22], the droplet size has a minor importance even though it is expected that smaller droplets would increase the fouling in the last stages.

Washing frequency and injection duration

Due to the fact that the deterioration rate is site specific and online washing is installed in attempt to reduce this rate, the wash frequency of an online wash sequence needs to be optimized for each site. Syverud, [22], claims that it is necessary with frequent washing intervals in order to clean off deposits. Stalder, [21], also claims that frequent washing intervals are necessary, especially when running online washing with water only.

It would be expected that longer injection duration would improve the recovery of lost performance. However, Syverud, [22], shows that increased injection duration will not compensate for low flow rates. This is also the findings done by Syverud and Bakken, [23].

4.4 Washing systems at Sleipner

Offline washing is run approximately every 3000 operating hours for the gas turbines evaluated in this report. In addition machine 1 and 3 have daily online washing and machine 4 run idle wash every 1000 hour between each offline wash. There are not many engines running with redundancy at Sleipner, and the goal with online washing is therefore to keep the performance as high as possible during the operating intervals. It is currently not a goal to increase the operating interval between each offline wash. This interval is already quite high compared to other Statoil platforms and other necessary maintenance is also performed during these shutdowns.

4.4.1 Online washing at normal operating load

Figure 4-3 shows the manually operated online washing system currently in use for machine 1 and 3. The washing unit enclosed with covers and doors is used for both offline and online washing. For water filling there is a deionization filter (DI-filter) and a 1 micron particle filter in front of the washing unit as shown in the figure below. The water is leaving the washing unit trough a HP filter, marked pressure outlet in figure 4-3. From the HP outlet the water continues to the HP nozzles at the bellmouth, as shown in figure 4-4.

DI- and particle filter is not necessary for running an offline washing sequence, so for the gas turbines without online washing a similar washing unit is used with a tube directly connected to the inlet of the unit.

There are two 50 litres containers inside the washing unit, one for the detergent/water mixture and one for clean water making sure that there is no trace of detergent in the clean water. When running an offline washing system a soap/water mixture is run through the compressor and left inside for about 5 minutes in order to dissolve all the surface deposits. Afterwards there will normally be two flushes of clean water, making sure there is no detergent left in the machine. One part soap is used for each 6 parts water. Detergent is not used for the online washing sequence; only clean water is run through the machine.

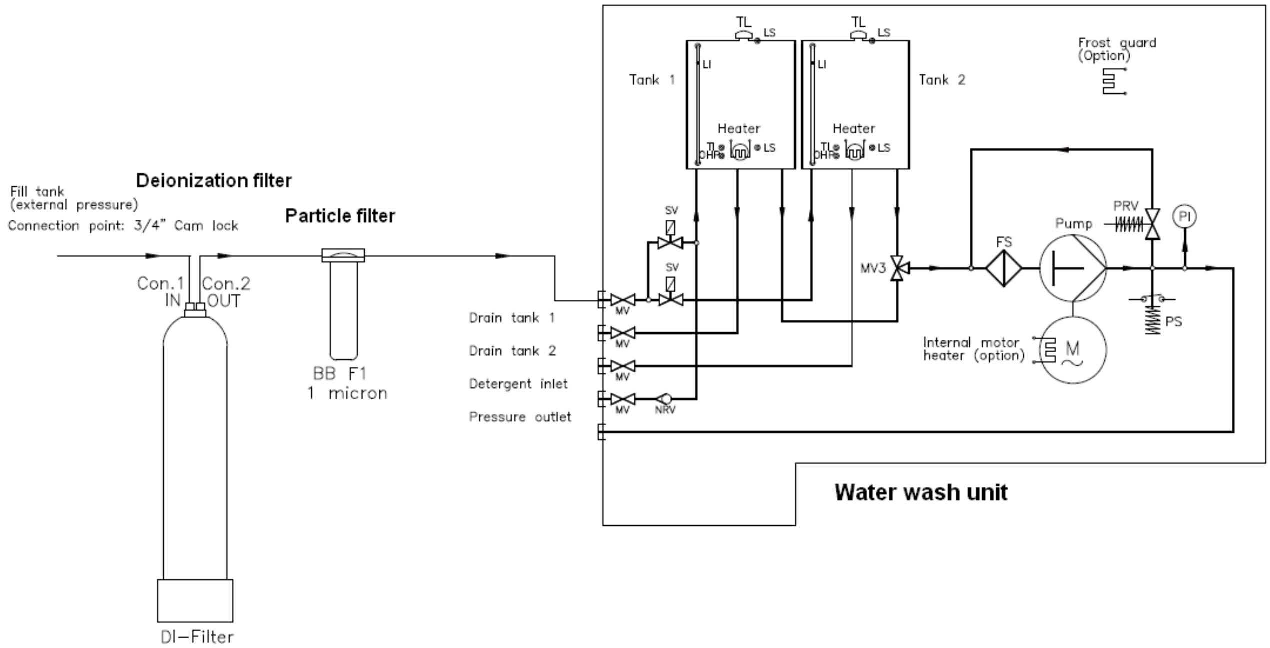


Figure 4-3: Online wash system

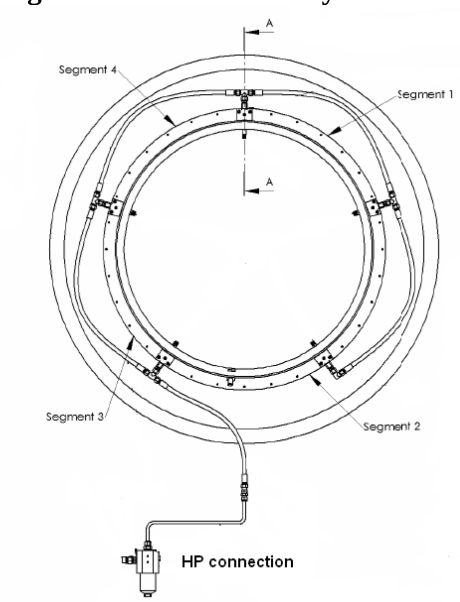


Figure 4-4: HP connection to bellmouth

4.4.2 Idle washing

Idle wash was started for machine 4 in order to test the performance of idle wash, and compare the effect of idle contra the effect of online at full load. Some problems arose the first time idle wash was run. The turbine tripped because the rotational speed went under a set safety value. This was fixed and the second idle wash was run with no further complications.

4.5 Summary

The deterioration rate is site specific and inlet filtration systems are installed at gas turbines offshore to protect the engine from contaminants from the surrounding air and to reduce the fouling rate. These filter systems are expected to operate in extremely varying conditions which can be quite challenging.

Compressor fouling can be regained by cleaning the engine, and there are currently two methods of water cleaning; online and offline washing. Offline washing is the most effective method, but it requires the engine to shut down which can result in lost production. Online washing does not affect the production, but is less effective. Normally offline and online washing are used in combination to increase the interval between each offline wash and to keep the performance as high as possible during the operating interval.

Offline washing is performed every 3000 operating hour on all the gas turbines evaluated in this study. In addition machine 1 and 3 have daily online washing and machine 4 run idle wash every 1000 hours between offline washing.

5 Data handling and processing

This chapter gives a brief description of how the operational data for the different gas turbines were extracted and processed. All data have been corrected to ISO standards in order to take the variation in ambient conditions into account. Data were extracted from TurboWatch and loaded into Excel files. MATLAB scripts were created to handle the large amount of data and visualize deterioration trends.

5.1 Data corrections

Great variation in ambient conditions offshore affect the engine performance, and correction factors are needed when processing and comparing data. The temperature and pressure correction factors (θ and δ) are defined as the ratio between actual conditions and a reference condition. The most common correction method is performed according to ISO standard with ISO reference conditions of 288.15K and 101.325kPa are used. All the corrected parameters originate from thermodynamics and Buckingham Pi Theorem.

The general equation is given by equation 5.1.

$$Parameter(corrected) = \frac{Parameter(actual)}{\theta^a \delta^b} \tag{5-1}$$

The values of the exponents *a* and *b* varies for different gas turbines and the traditional turbine parameters are given by Volponi, [28], and are shown in table 5-1. However, it is not unusual that manufactures adjust these exponents after collecting enough test data in order to take more considerations into account [14]. Table 5-2 shows corrected parameters adjusted to the LM 2500 series.

It is common practice to substitute the ambient parameters with the compressor inlet conditions in the correction formulas in order to take losses at the intake system into account. The parameters in this report is for this reason corrected against T_2 and p_{2}.

The temperature correction factor, θ, is given by equation 5.2:

$$\theta = \frac{T_{ambient}}{T_{reference}} = \frac{T_2}{T_{ref}} \tag{5-2}$$

The pressure correction factor, δ, is given by equation 5.3:

$$\delta = \frac{p_{ambient}}{p_{reference}} = \frac{p_2}{p_{ref}} \tag{5-3}$$

Table 5-1: General gas turbine parameters [28]

Parameter	a	b	Corrected parameter
Temperature (T)	1	0	$T_c = \frac{T}{\theta}$
Pressure (p)	0	1	$p_c = \frac{p}{\delta}$
Gas generator speed (N)	0.5	0	$N_c = \frac{N}{\sqrt{\theta}}$
Shaft power (P)	0.5	1	$P_c = \frac{P}{\delta\sqrt{\theta}}$
Compressor airflow (m_a)	-0.5	1	$\dot{m}_{ac} = \frac{\dot{m}_{ac}\sqrt{\theta}}{\delta}$
Fuel flow (m_f)	0.5	1	$\dot{m}_{fc} = \frac{\dot{m}_{fc}}{\delta\sqrt{\theta}}$

Table 5-2: Corrected parameters for the LM2500 series

Parameter	a	b	Corrected parameter
Inlet temperature PT, T_{54}	0.85	0	$T_{54c} = \frac{T_{54}}{\theta^{0,85}}$
Inlet pressure PT, p_{54}	0	1	$p_{54c} = \frac{0,985 \cdot p_{54}}{\delta}$
Compressor discharge pressure, CDP_c	0	1	$CDP_c = \frac{0,9897 \cdot p_3}{\delta}$
Gas generator speed (N)	0.5	1	$N_c = \frac{N}{\sqrt{\theta}}$
Compressor isentropic efficiency, η_c	0.0312	0	$\eta_{cc} = \frac{\eta_c}{\theta^{0,0312}}$

5.2 Data handling and processing

Data from TurboWatch were exported into Excel spreadsheets for further processing. Three MATLAB scripts, shown in appendix A, B and C, were created in order to better handle the great amount of data, and to visualize the trends. The function of these scripts was to import parameters from Excel to MATLAB, correct the parameters to inlet compressor conditions and than plot two and two parameters against each other. The values for corrected CDP are for example plotted against corrected EGT. The color of the points is changing as time pass by, making it easy to see how the trends are developing over time. A color bar is included next to the graph to show what date the different points belong to. All data were extracted from TurboWatch with a logging frequency of every second hour.

It is important to take filter change, installation of new filter system and change of gas generator into account because these issues will affect the performance. For the engines evaluated tables with dates for water washing, filter change and other maintenance are included in appendix D.

It is desirable to compare data at same operating conditions and eliminate as many variables as possible. GE, [10], suggests that data should be recorded at approximately the same gas generator speed. A MATLAB script was developed in order to only plot data within a certain interval (100 rpm for N1 or 5K for T54). This interval had to be large enough to ensure reliable trends, but at the same time eliminate data measured at great divergent conditions. It is also important to be aware of other conditions affecting the performance trends. If the engine has been run with anti-icing for parts of an operating period, this is one condition that could have had great impact on the performance trends.

5.3 Limitations

There are several limitations related to the correcting methods. When using the ISO correction method several conditions are not taken into account. First of all the results will depend on how far the pressure and temperature are from ISO conditions. The results will be more diverging as further away the operating conditions are from ISO conditions. There may also be mechanical limits that exist at low ambient conditions but not at high ambient conditions. Relative humidity is not taken into account in the ISO correction method and at high ambient temperature conditions the relative humidity will affect the performance [14]. According to Krampf, [14], the ISO correction method is only valid relatively near ISO conditions.

5.4 Summary

Due to great variations in ambient conditions at offshore installations correction methods are needed for processing and handling data. The most common method, also used in this report, is the ISO correction method. Even though this method is commonly used, it still has several limitations. The ISO method is most accurate when used for temperature and pressure close to ISO standards, and not as reliable for conditions far away from ISO conditions. Relative humidity is not taken into consideration, and this could also have great affect on the performance during high ambient conditions.

The analysed data was extracted from the monitoring software, TurboWatch, and loaded into Excel spreadsheets. MATLAB scripts were created in order to easier handle the great amount of data available. Series of two parameters were plotted against each other, and the operating points are changing colour with time in order to better visualize performance trends over time.

6 Validation of TurboWatch

TurboWatch is a set of software tools used to monitor the performance of industrial processes. TurboWatch is used by several platforms including Sleipner. The program collects data from various sources and computes performance indicators based on thermodynamically relations. The measured and computed data is stored in historical tables for long term trending.

The present study only focuses on TurboWatch. A comparison with other software tools used for monitoring as e.g. TIKS also used by Statoil, could be interesting. It was, however, decided not to do this because the calculations beyond the software TIKS are protected by the manufacturer.

The chapter below concentrates on the validation of TurboWatch routines for monitoring compressor degradation and water wash routines. Calculations and equations applied in the program are evaluated and discussed, and a sensitivity analysis is performed.

6.1 TurboWatch Console

TurboWatch uses a console panel in order to organize the data from the specific plant. The console panel is a collection of individually monitored parameters and work as a graphical browser for monitoring the process. The main panel gives an overall view of all the different engines and a green or red light symbolize whether the engine is running or not. This gives the user a quick overview of the current situation at the plant and allows choosing the area of interest. Figure 6-1 shows the console panel for machine 1.

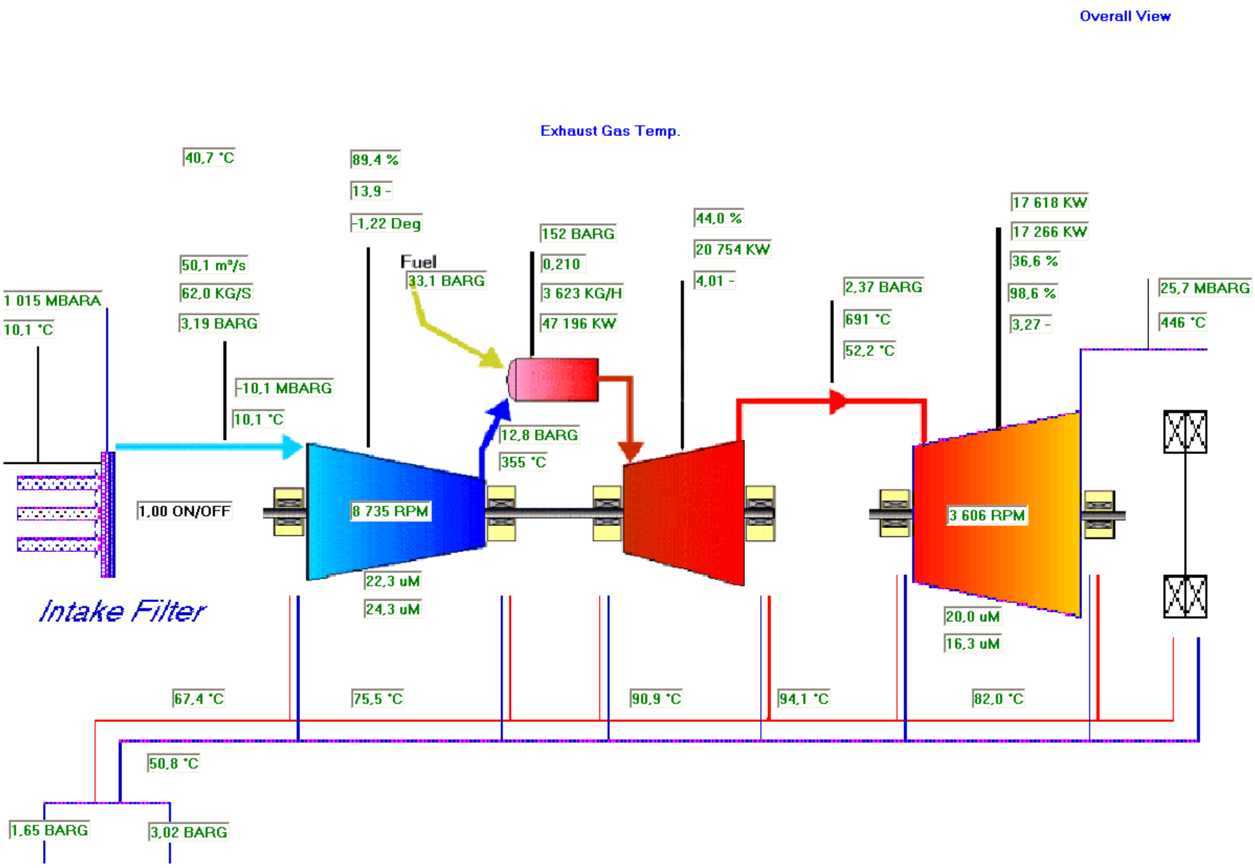
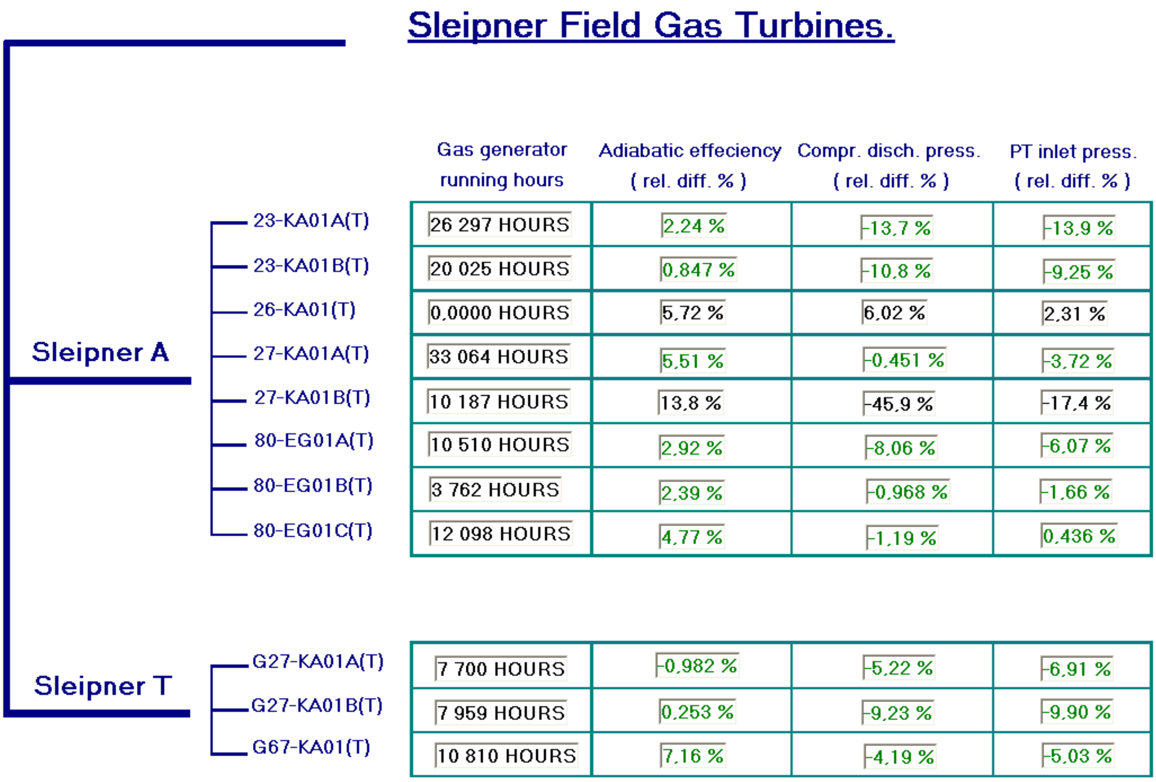


Figure 6-1: Console panel for machine 1

6.1.1 Water wash panel

The Sleipner field has an integrated water wash panel monitoring several parameters. The panel is intended for monitoring different parameters and suggest when maintenance is necessary. This panel is currently not being used at Sleipner due to lack of confidence in the panel and which calculations it is based on. This section will therefore discuss the user value and possible improvements.

The water wash panel is shown in figure 6-2. In the left column all the gas turbines at Sleipner are represented with TAG numbers. It is common that all equipments at a process plant are equipped with TAG numbers valid for that specific engine or part. Operating hours, adiabatic efficiency, compressor discharge pressure and power turbine inlet pressure for all the gas turbine are shown in the next columns. The values are given in relative percentage differences.


Sleipner Field Gas Turbines.

		Gas generator running hours	Adiabatic effeciency (rel. diff. %)	Compr. disch. press. (rel. diff. %)	PT inlet press. (rel. diff. %)
Sleipner A	23-KA01A(T)	26 297 HOURS	2,24 %	-13,7 %	-13,9 %
	23-KA01B(T)	20 025 HOURS	0,847 %	-10,8 %	-9,25 %
	26-KA01(T)	0,0000 HOURS	5,72 %	6,02 %	2,31 %
	27-KA01A(T)	33 064 HOURS	5,51 %	-0,451 %	-3,72 %
	27-KA01B(T)	10 187 HOURS	13,8 %	-45,9 %	-17,4 %
	80-EG01A(T)	10 510 HOURS	2,92 %	-8,06 %	-6,07 %
	80-EG01B(T)	3 762 HOURS	2,39 %	-0,968 %	-1,66 %
	80-EG01C(T)	12 098 HOURS	4,77 %	-1,19 %	0,436 %
Sleipner T	G27-KA01A(T)	7 700 HOURS	-0,982 %	-5,22 %	-6,91 %
	G27-KA01B(T)	7 959 HOURS	0,253 %	-9,23 %	-9,90 %
	G67-KA01(T)	10 810 HOURS	7,16 %	-4,19 %	-5,03 %

Figure 6-2: Water wash panel Sleipner – TurboWatch

The relative difference (in %) for a parameter is

$$X_r = \frac{X_c - X_{\exp}}{X_{\exp}} \qquad (6\text{-}1)$$

Where
X_r = relative difference
X_c = Corrected value (actual value)
X_{exp} = expected value (obtained from performance map)

For each of the three parameters shown in the water wash panel the expected value is calculated using associated performance maps. These maps compare current operating conditions with expected behavior.

Operating hours

The operating hours, from when the last gas generator was installed, are shown in the first column of the panel. Communication and logging problems is causing down periods in TurboWatch. Due to these challenges the number of operating hours showing in the panel does not correspond to the real number of operating hours. The corrected number of operating hours is available in other software and tables and it is therefore recommended to remove this column from the water wash panel.

Efficiency and pressure

The values for efficiency, CDP and p54 are given as relative differences in the panel. These calculations are based performance maps where the expected value in form of a general baseline is implemented. These baselines are generated for a general engine. Values for the relative differences are only useful if the baseline for the specific machine is known and corrected to the current gas generator. It is also of great importance that the acceptable limits for the different values are known for the specific machine so it is defined when to take action.

Other parameters

For the purpose of detecting fouling and performance deterioration there are also other parameters that could have been included in the water wash panel. Reduction in compressor airflow is one parameter used in order to detect fouling and could be implemented in the panel. Airflow is, however, not a monitored quantity for the gas turbines at Sleipner. The airflow is simply a calculated value in TurboWatch, allowing uncertainties in the parameters included in the calculation to affect the result. Another parameter that could be implemented in the panel is exhaust gas temperature. This parameter is suggested by GE, [10], to help detect performance deterioration if evaluated for constant gas generator pressure ratio.

6.2 Evaluations of equations and calculations

Different monitored parameters can be extracted from TurboWatch. In addition the software also calculates values for efficiency, power output and other quantities. The greater part of these calculations is, however, protected by the manufacturer, making the validation of TurboWatch challenging.

6.2.1 Performance maps

The calculations for the relative differences are based on performance maps. For compressor discharge pressure and power turbine inlet pressure these maps are shown in appendix E for a chosen operating period (10^{th} of March to 10^{th} of July 2010).

The performance maps for compressor efficiency are shown in figure 6-3 and 6-4. The grey curve symbolizes the expected value (the baseline) while the blue dots are the measured operating points. For machine 3, in figure 6-4, all the measured operating points appear far above the baseline. This is a surprisingly since the compressor normally will degrade over time causing the operating points to appear below the expected value. This implies that the baseline is not adjusted to the current gas generator. However, this deviation could also be caused by other factors e.g. errors in measurements and calculations.

All the operating points have the same color, and it is therefore not possible to observe possible deterioration trend for the different engines. For machine 1, in figure 6-3, some of the operating points are below the baseline. However, it is not clear if these points belong to the start or the end of the operating period which making this graph useless.

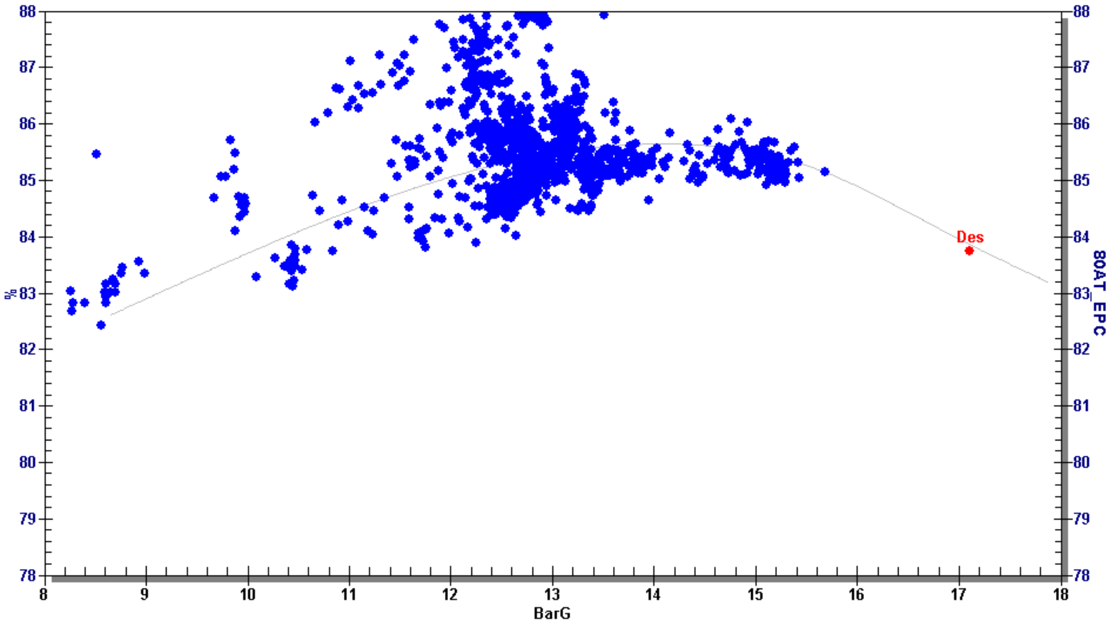


Figure 6-3: Performance map compressor efficiency Machine 1

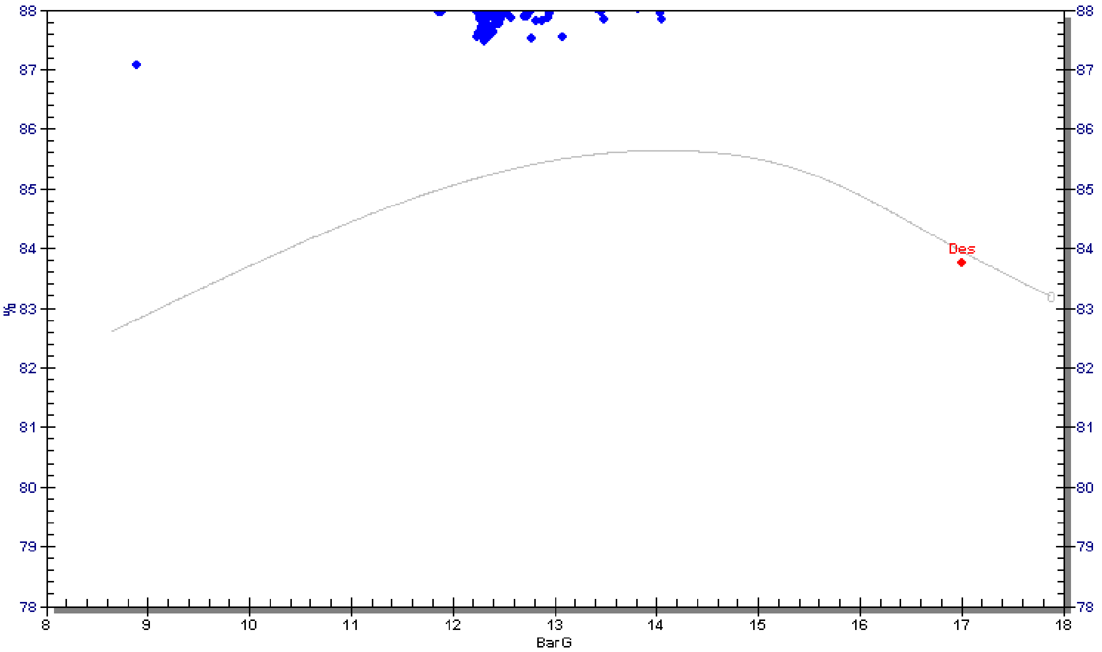


Figure 6-4: Performance map compressor efficiency Machine 3

6.2.2 Efficiency

When considering machine 1, 2 and 3, TurboWatch claims to calculate the polytropic compressor efficiency for machine 3 and the isentropic compressor efficiency for machine 1 and 2. To test this a comparative study was performed by calculating compressor efficiencies in HYSYS and comparing these values with those extracted from TurboWatch..

Figure 6-5 shows the compressor simulation model. This model calculates the compressor efficiencies by implementing monitored values for pressure and temperature. An adjust function calculates the polytropic compressor efficiency by specifying the compressor outlet temperature. The calculated values for polytropic and isentropic efficiencies are compared to calculations done in TurboWatch. The calculations are shown in Appendix F and were performed for four randomly selected operating points.

The results indicate that TurboWatch calculates the isentropic compressor efficiency for all the three machines, implying that the notation for compressor efficiency for machine 3 is incorrect.

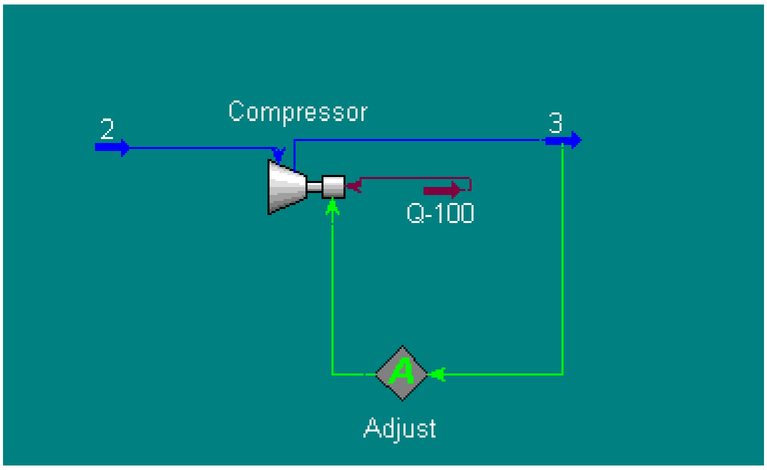


Figure 6-5: HYSYS compressor model

Since the difference between the polytropic and isentropic efficiency appears to be quite large it is important to assure that the right notation is being used. However, when working with and analysing trends the most interesting aspect is whether the choice of efficiency actually affect the deterioration trends. The efficiency for the four random operating points are plotted in figure 6-6 and 6-7. The four operating points are selected randomly and do not reflect the real deterioration trend, only the shape of the trend lines.

The two graphs clearly illustrates that the choice of efficiency does not affect the shape of the deterioration trend despite the great difference between polytropic and isentropic efficiency. This implies that the choice of efficiency is not critical when trying to reveal deterioration trends. The critical point seems to be that the choice of efficiency needs to be consistent throughout an analysis.

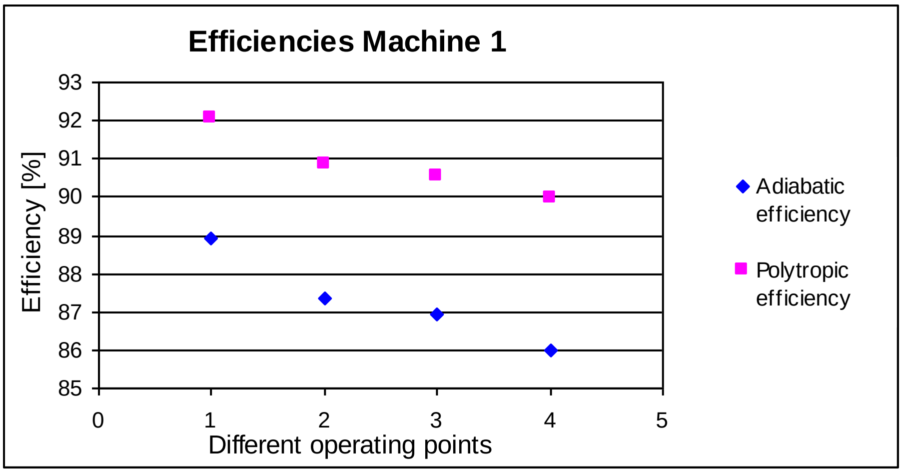


Figure 6-6: Comparison of efficiencies Machine1

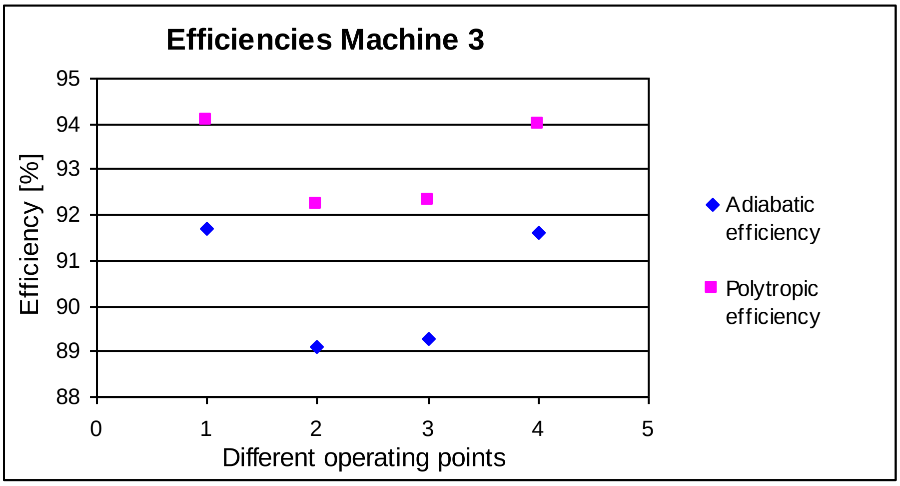


Figure 6-7: Comparison of efficiencies Machine 3

6.2.3 Data corrections

Due to the varying ambient conditions offshore correction factors are needed when comparing and analysing data. Since the calculations behind TurboWatch are protected by the manufacturer, it is challenging to reveal which correcting formulas the software has based its calculations on. The only known formulas are pressure and temperature correction factors (δ and θ). These formulas are equivalent with equations 5.2 and 5.3, where the pressure and temperature is corrected against p_2 and T_2.

The section below discusses possible corrections formulas implemented in TurboWatch. For a chosen period (10.03.10-10.07.10) corrected values calculated in TurboWatch are compared to corrected values calculated with the correction formulas in table 5-1 and 5-2.

Comparing general and specific equations

There is only one equation available for corrected gas generator speed (N1c). Values for corrected GG speed were extracted from TurboWatch and plotted against manually calculated values. As shown in figure 12-5 in Appendix G, the graphs fully overlap implying that TurboWatch have used the general equation for corrected GG speed.

The equation for corrected compressor discharge temperature (T3c) is not given by GE and only one formula is available in this project. The calculated values for T3c are plotted against the extracted values from TurboWatch (figure 12-6 in appendix G). The fact that the graphs fully overlap clearly demonstrates that TurboWatch is based on the general formula.

One general and one specific equation were available for corrected CDP. Calculations done with the two different equations for machine 1 and 3 are shown in figure 6-8 and 6-9. These values, when applying the specified and general equations, are plotted respectively against the values extracted from TurboWatch. It is interesting to note that these graphs imply that there are small deviations when using both the equations. It is therefore impossible to conclude which formula for corrected CDP is used in the software TurboWatch. The deviation is, however, quite small and it is therefore concluded that for further calculations in the present study the specified equation will be applied.

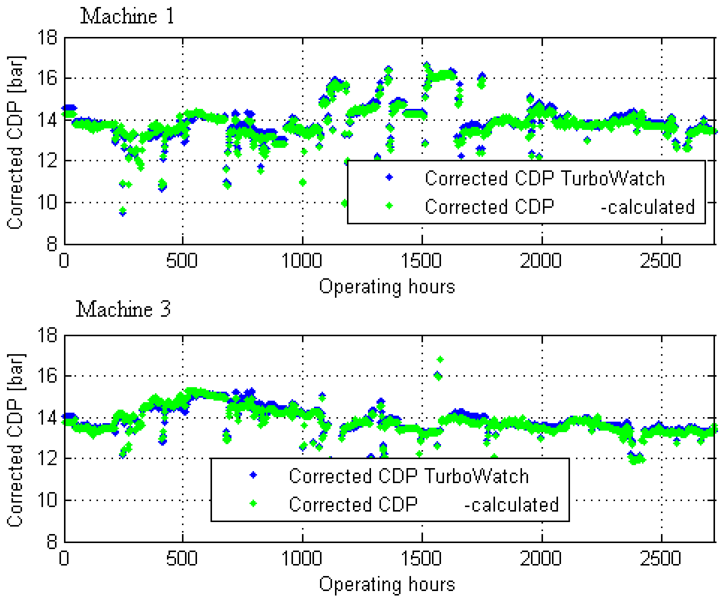


Figure 6-8: Comparison of corrected CDP when using the specified formula

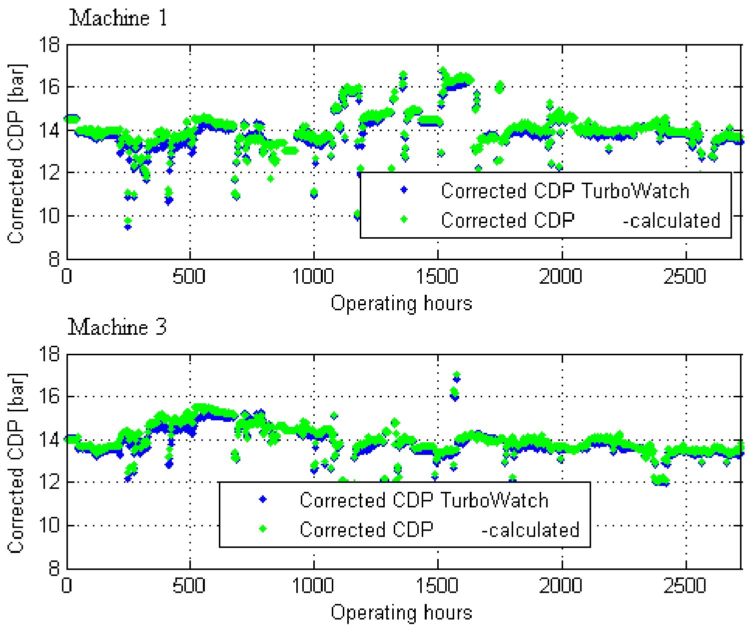


Figure 6-9: Comparison of corrected CDP when using the general formula

Corrected EGT has a specified formula for the LM2500 series in addition to the general gas turbine formula. The extracted values from TurboWatch were compared to the values calculated by the two different equations, shown in figure 12-7 and 12-8 in appendix G. The curves fully overlap when the general equation is applied.

In order to identify the effect of using various correction equations, corrected EGT is calculated with the two different equations over reasonable range of inlet temperature for offshore conditions (-5 to 20 degrees Celsius). The deviation between the two equations was evaluated demonstrating that the deviation at cold ambient conditions was largest, but all over quite small. However, if small deviations (from 0.26 % to about 1 %) occur for several parameters due to use of different correction equations, it can actually affect the performance trends. The effect will be greatest when the gas turbines are operating in cold conditions which are also furthest away from ISO conditions (15 degree Celsius).

Table 6-1: Deviation when using different equations for EGT

T2 (oC)	T2 (K)	Theta	EGTc (general)	EGTc (specific)	Deviation [%]
-5	268,15	0,931	1046,81	1035,57	1,07
0	273,15	0,948	1027,65	1019,44	0,80
5	278,15	0,965	1009,17	1003,84	0,53
10	283,15	0,983	991,35	988,75	0,26
15	288,15	1,000	974,15	974,15	0,00
20	293,15	1,017	957,53	960,01	0,26

6.3 Sensitivity

When extracting the monitored values from TurboWatch it is important to be aware of the fact that some uncertainties and inaccurate measurements may occur. A sensitivity analysis was performed in order to investigate how small changes in T3, T2 and CDP would affect the polytropic head and polytropic compressor efficiency. Inaccurate instrumentation is one factor that could affect the monitored parameters and lead to imprecise calculated values.

The change in polytropic efficiency when varying T3, T2 and CDP with 1 and 2 % is shown in figure 6-10. The graph clearly illustrates that polytropic compressor efficiency is mostly influenced by changes in T3. Changing T3 with ± 2% results in a 3 % range in efficiency. Changes in both CDP and T3 lead to great variations in polytropic head. Changes in T2 had small affects on both the polytropic head and the efficiency.

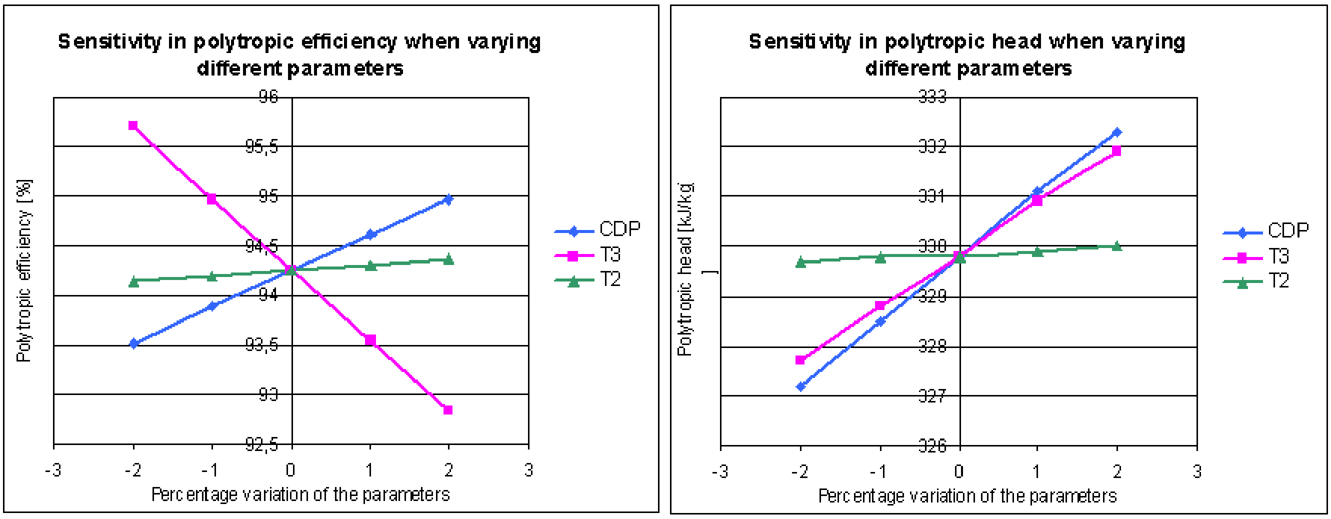


Figure 6-10: Sensitivity in polytropic head and efficiency when varying certain parameters

According to Kramp, [14], a higher deviation is expected as further the conditions are from ISO standards. Accuracy in instrumentation is also an important factor. If the allowed deviation in instrumentation is more than 2 % it will have en even higher impact on the polytropic efficiency and head. Attempts have been made to uncover the accuracy used for instrumentation at Sleipner without succeeding. This needs to be known for further work.

6.4 Discussion

TurboWatch was validated with main focus on routines for performance monitoring and water washing. The calculations, on which the software TurboWatch is based, are to a certain degree protected by the manufacturer making the validation of the software challenging

The water wash panel implemented in TurboWatch was first evaluated. This panel is currently not in use, and it is desirable to make it more user-friendly to help monitoring and controlling performance deterioration. It is not of current interest to start condition monitoring at Sleipner due to already quite high operating intervals and the fact that other important maintenance is performed during each shutdown. However, the water wash panel could be used as an extra control for the engines and to show the performance effect with and without online water wash.

The evaluation of the water wash panel demonstrated that the amount of operating hours in the panel was not correct. The deviation in operating hours is most likely due to communication and logging problems which periodically lead to shut downs of the software, TurboWatch. The other parameters included in the panel were shown in relative differences based on baselines for general engines. All engines have different behaviour and the baseline should be adjusted to the current gas generator in order to give precise feedback on the performance deterioration.

One important finding was that the isentropic compressor efficiency for machine 3 turned out to be far above the baseline implemented in TurboWatch. As it is not likely that the engine's performance has improved over time, this strongly indicates that the general baseline deviates greatly from where it should be for the current gas generator. This has serious implications for monitoring performance deterioration since the relative differences are only useful when the current baseline is known and corrected to the current gas generator.

TurboWatch claims to calculate the polytropic efficiency for machine 3 and the isentropic efficiency for machines 1 ands 2. To test this, a comparative study was performed by calculating compressors efficiencies in HYSYS and comparing these values to those extracted from TurboWatch. The result of this analysis showed clearly that TurboWatch actually is calculating the isentropic efficiency for all three engines, and not the polytropic efficiency for machine 3. If the efficiencies are used uncritical and the notation is wrong, this can lead to serious consequences when evaluating performance. It seemed like the choice of efficiency did not affect the deterioration rate itself. However, the choice of efficiency and the notation need to be consistent. So when comparing two engines, polytropic efficiency can not be used for one engine while isentropic efficiency is used for the other one.

The correction equations were validated by plotting corrected parameters extracted from TurboWatch and manually calculated parameters. The result of this validation demonstrated that TurboWatch uses the general correction formulas for most parameters. For corrected compressor discharge pressure (CDP) none of the two correction formulas matched the values extracted from TurboWatch. CDP is a measured value and uncertainties in measurements may occur. It could also be possible that TurboWatch is using a different correction formula for CDP.

For corrected exhaust gas temperature (EGT) the difference between the general and the specified equation was evaluated over a normal range of inlet temperature for offshore installations (-5 to 20 degrees Celsius). This showed that the deviation could constitute for up to 1%, and was highest during cold conditions, furthest away from ISO conditions.

A sensitivity analysis was performed in order to investigate how small changes in certain parameters could affect the polytropic head and efficiency. This study showed that changes in CDP and T3 had substantial influence on polytropic head and polytropic efficiency. This demonstrates that accurate instrumentation and monitoring has great influence on the results and thereby also affect the deterioration trends.

6.5 Conclusion

When evaluating the water wash panel the amount of operating hours turned out to be incorrect and should therefore be removed from the panel as this can cause misunderstandings. This information is also available in other software programs and should be withdrawn from there. The water wash panel also includes relative differences for several parameters. The calculations for relative differences are based on baselines for general engines. However, all engines behave differently, so when analysing performance deterioration, relative differences are only useful when the current baseline is known and corrected to the current gas generator.

The calculations behind TurboWatch are to a certain degree protected by the manufacturer which made the validation of the software challenging. However, it was discovered that TurboWatch is using the general correction equation and not the specified equations for the LM 2500 series. This can lead to additional deviation, especially far away from ISO conditions.

TurboWatch is not using the correct notation for compressor efficiency for machine 3, and this should be corrected. The software claims to calculate the polytropic compressor efficiency, but actually calculates the isentropic compressor efficiency for this engine.

All cases discussed above clearly demonstrate the importance of having in-depth knowledge on calculations and equations on which a monitoring program is based. Only when having all the background information (current baselines, equations, calculations etc) confident decisions based on this software can be made.

7 Deterioration rates at Sleipner

The evaluated gas turbines at Sleipner are discussed in this chapter. The choice of gas turbines is first described and then deterioration rates are analysed.

Sleipner has a turnaround every other year in order to perform maintenance and other tasks which cannot be performed during operation. The last turnaround was from the 18th of August the 7th of September 2010, and these weeks are therefore neglected in the analysis of performance trends.

7.1 Gas turbines selected

The gas turbines analysed and evaluated in this project was selected in collaboration with my supervisor and after requests from Statoil. All the engines evaluated are located at Sleipner and are in the LM2500 series similar to the gas turbine in figure 7-1.

Different aspects were considered when selecting the engines, the most important being the attempt to reveal the effect of online water wash. Based on this criteria four machines were chosen. Machine 1, 2 and 3 were chosen as they run with load distribution and thereby represent comparable engines. The engines run offline washing every 3000 operating hour. In addition machine 1 and 3 have a daily online washing routine, started on the 10^{th} of July 2010 and 1^{st} of May 2009 respectively. Machine 3 is equipped with filter system B, while machine 1 and 2 have newer filter systems (filter system A). Machine 1 and 2 are situated on the same level while machine 3 is situated at a higher level at the platform.

Machines 1 and 2 were selected as they make a very good basis of comparison due to their quite similar load, equal filter systems and their location at the same level. Machine 3 was selected due to the fact that it has been running with online washing continuously since May 2009.

The last engine selected (machine 4) runs with idle washing and was for that perspective an interesting test base. Idle washing started in September 2010 and is run every 1000 operating hours between each offline washing. Machine 4 run with redundancy and the load can therefore be transferred to a standby machine during idle washing. In this study the period before idle washing started was compared to the first period running with idle wash. During November 2010 the engine unfortunately had to shut down due to high vibrations. An extra offline washing sequence was performed influencing the analysis, in many ways a typical situation occurring when a project is undertaken during operational conditions.

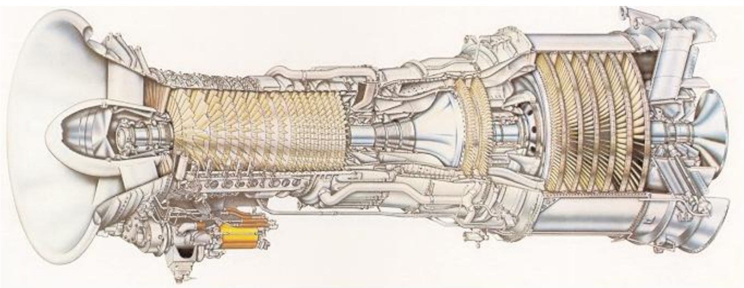


Figure 7-1: LM2500

7.2 Borescope inspection

Although analysis of borescope inspections was not a part of this work, boresecope pictures taken of the compressor section in machine 3 after online washing started gave valuable additional information. Several borescope inspections have been performed after the installation of the online system on machine 3, all indicating cleaner compressor blades in machine 3 compared to machines 1 and 2 after 3000 running hours.

Online washing was started for machine 1 in July 2010, and a borescope inspection was performed after the first 3000 hours interval with daily online washing. Some of the pictures of the compressor sections from this borescope inspection are shown in Appendix H, indicating that the compressor blades appear cleaner than after the previous 3000 hours interval. Despite low picture quality it was possible to observe that the compressor blades were considerable cleaner in machine 1 then in machine 2 after the 3000 hour operating interval.

7.3 Deterioration rates for different engines

Deterioration rates for four selected gas turbines are discussed in the following section. Different parameters are plotted against each other in attempt to reveal stable and reliable deterioration trends.

The literature study performed initially (see chapter 3) documented that compressor discharge pressure, when plotted against corrected gas generator speed or exhaust gas temperature, is a stable parameter. The present study therefore includes plots of CDPc plotted against corrected GG speed. CDPc was also plotted against EGTc. However, no clear trends could be detected from these plots, and these graphs are therefore included in appendix I.

The literature study revealed that a reduction in compressor efficiency is an expected result of compressor fouling. Compressor efficiency is a calculated value based on several parameters each with their own uncertainty. A discrepancy regarding the accuracy of compressor efficiency for detecting performance deterioration therefore exists in the published literature. In the following analysis corrected isentropic compressor efficiency is plotted against corrected GG speed.

The last relation analysed is corrected exhaust gas temperature (EGT) plotted against gas generator pressure ratio, a relation recommended by GE, and is for that reason considered in this report. However, GE [10], is also basing its procedure on a baseline for a general engine. This general curve is currently not available and is therefore not evaluated in this study.

Compressor airflow is plotted against corrected gas generator speed and these graphs are included in appendix I. Machines 2 and 3 have quite similar airflows during the same operating periods, but the airflow for machine 1 varied greatly from the two other engines. This was not expected since the three engines run with load distribution. It could be caused by errors in measurements or calibration. Due to the unknown reason for this great deviation between airflows, it was chosen to not focus on these graphs during the following evaluation.

It was also made an attempt to analyse the deterioration rates over a limited area of rotational speed and exhaust gas temperature. These graphs are shown in appendix J. The analysis revealed high proportion of the data scatter even though the area of rotational speed and exhaust gas temperature was limited. It was therefore challenging to find a limited but at the same time representative area, so in the following analysis all the data points are therefore included in the graphs.

7.3.1 Machine Three (3)

Machine 3 has been running with a daily online washing routine since May 2009. Bore scope picture has shown good visual results after online washing started, but documentation of the performance gain still remains. Since machine 3 had been running continuously with online washing for a long period, it was decided to start evaluating this engine using machine 1 as reference. Performance trends are shown in the graphs in the following section; the plot for machine 1 is always shown to the left of machine 3.

The tables in appendix D shows the planned maintenance and water washing for machine 1 and 3. Different periods were considered, and the last operating considered was the period with the most stable operation. It was decided to focus on this period, from the 10th of March to the 10th of July 2010 when comparing deterioration rates for machine 3.

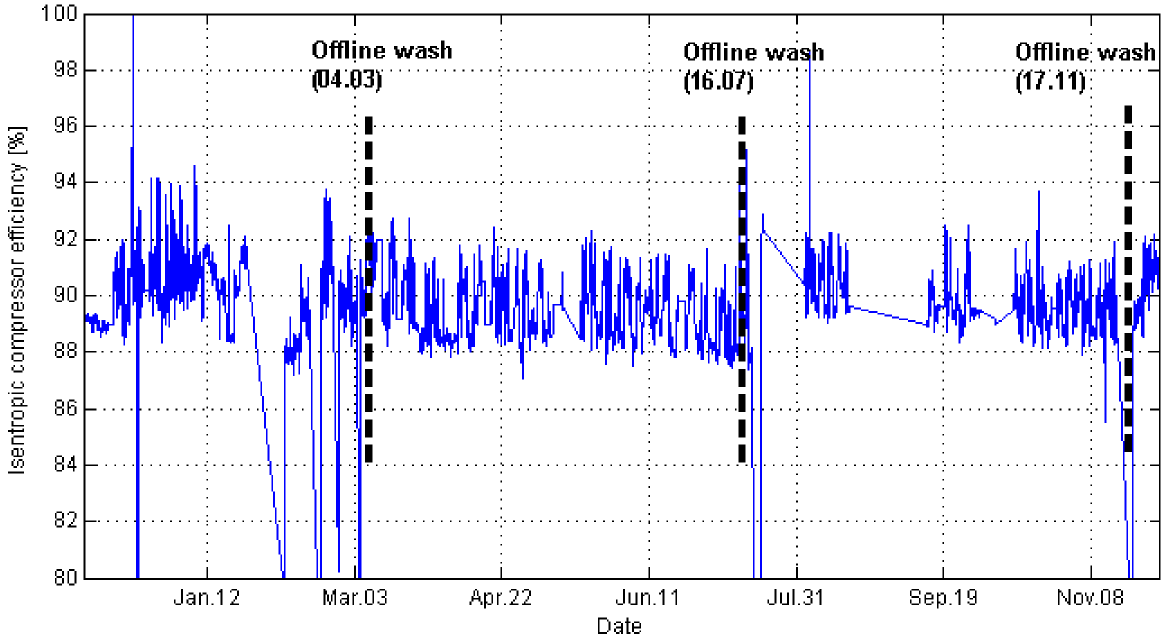


Figure 7-2: Compressor efficiency for Machine 3 – 01.12.09-01.12.10

Figure 7-2 shows the isentropic compressor efficiency for machine 3 over a one year period. The performed offline washes are indicated in the graph. The plot clearly shows that the efficiency is quite stable during the period. Even though the efficiency varies between 88 and 92% it does not have a decreasing trend towards the end of the operating period. The corrected GG speed for this period is shown in appendix K and it is clear that it is quite constant during the period. It mostly lay between 8700 and 8900 rpm.

Compressor discharge pressure

Compressor discharge pressure versus corrected gas generator speed

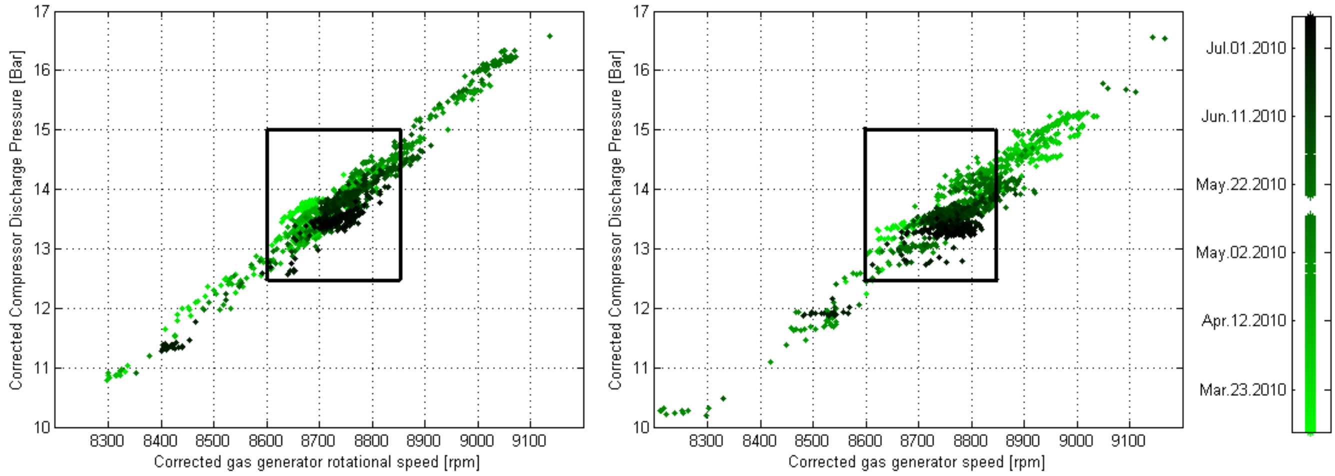


Figure 7-3: CDPc vs corrected GG speed - Machine 1 and 3

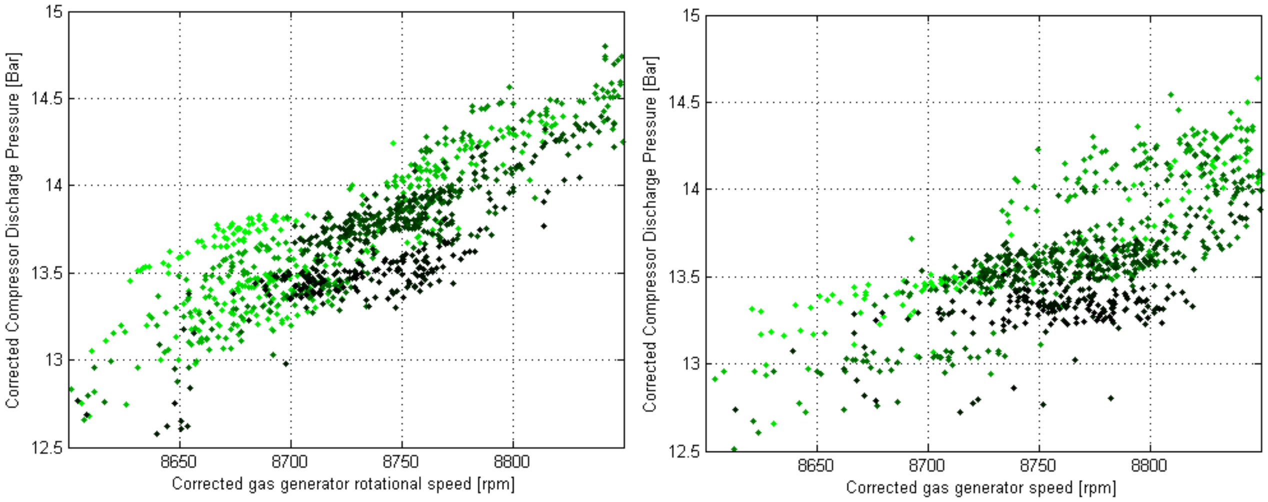


Figure 7-4: A selected area of CDPc vs corrected GG speed –Machines 1 and 3

Figure 7-3 shows corrected compressor discharge pressure plotted against corrected gas generator speed for both engines. Corrected CDP is quite concentrated for constant GG speed and the marked area in figure 7-3 is enlarged in figure 7-4 to better visualise possible trends.

CDPc for machine 3 is expected to be higher than CDPc for machine 1 throughout the period due to the online washing at machine 3. The plot for machine 1 reveals a decreasing trend in CDPc when evaluated for constant GG speed. The plot for machine 3 has more diverging points making it more challenging to observe a specific trend. It seems like there is a slightly decreasing trend in CDPc for machine 3 as well, but it is not as obvious as for machine 1.

Compressor efficiency

Corrected compressor efficiency versus corrected GG speed

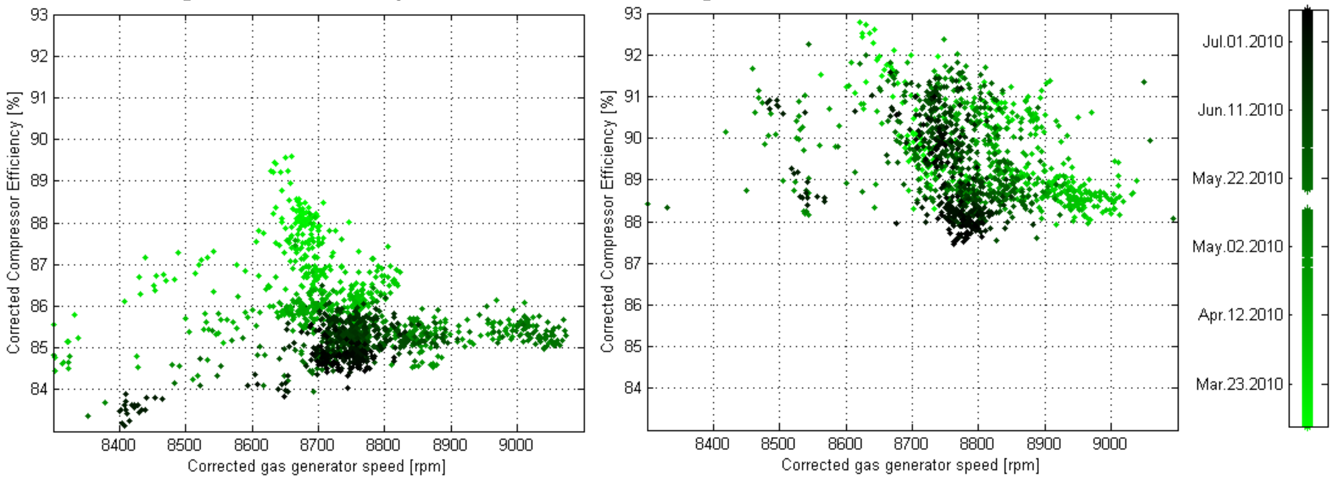


Figure 7-5: Corrected compressor efficiency vs corrected GG speed - Machine 1 and 3

Figure 7-5 shows corrected compressor efficiency plotted against corrected gas generator speed. The data scatter is quite high for both engines. The compressor efficiency for machine 1 clearly has a decreasing tendency when evaluated for constant gas generator speed. This is not the case for machine 3 where the efficiency is more equal when evaluated at constant GG speed.

From the graph above it is clear that the efficiency is much higher for machine 3 than machine 1 during the operating period. Machine 1 is older than machine 3 and this could be the explanation for the difference in compressor efficiency. However, the deviation could also be caused by errors in the instrumentation.

Exhaust gas temperature

Corrected exhaust gas temperature versus gas generator pressure ratio

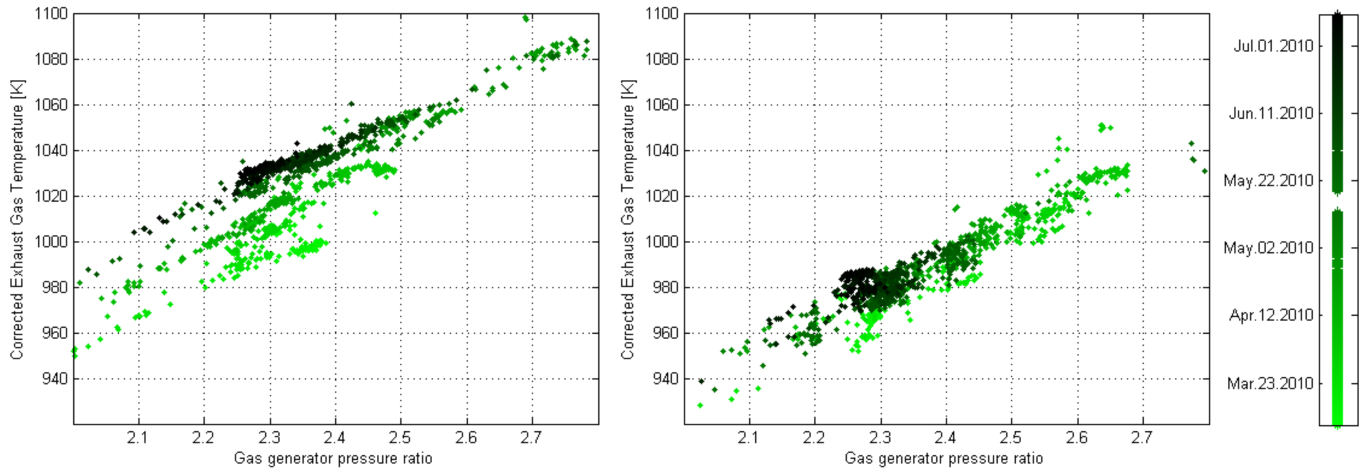


Figure 7-6: Corrected EGT vs GG pressure ratio - Machine 1 and 3

In figure 7-6 corrected exhaust gas temperature is plotted against gas generator pressure ratio. This is a recommended relation from GE, [10]. The figure above shows a great difference in exhaust gas temperature between machine 1 and 3. Machine 1 has a clear increasing trend for EGTc when evaluated for constant GG pressure ratio. Machine 3 has a slightly increasing trend for EGTc as well, but has a smaller range in EGTc than machine 1.

7.3.2 Machine One (1)

This section evaluates the performance effect of online washing at machine 1 using machine 2 as reference. Online washing was started on machine 1 the 15th of July 2010.

Machine 1 is evaluated for two operating intervals; the interval before online washing started and the first interval running with online washing. The second interval included a turnaround; from the 18th of August to the 7th of September. These three weeks are left out of this analysis. The last maintenance shutdown considered in this report was the 15th to the 17th of November. Dates for offline washing sequences and other maintenance are included in Appendix D.

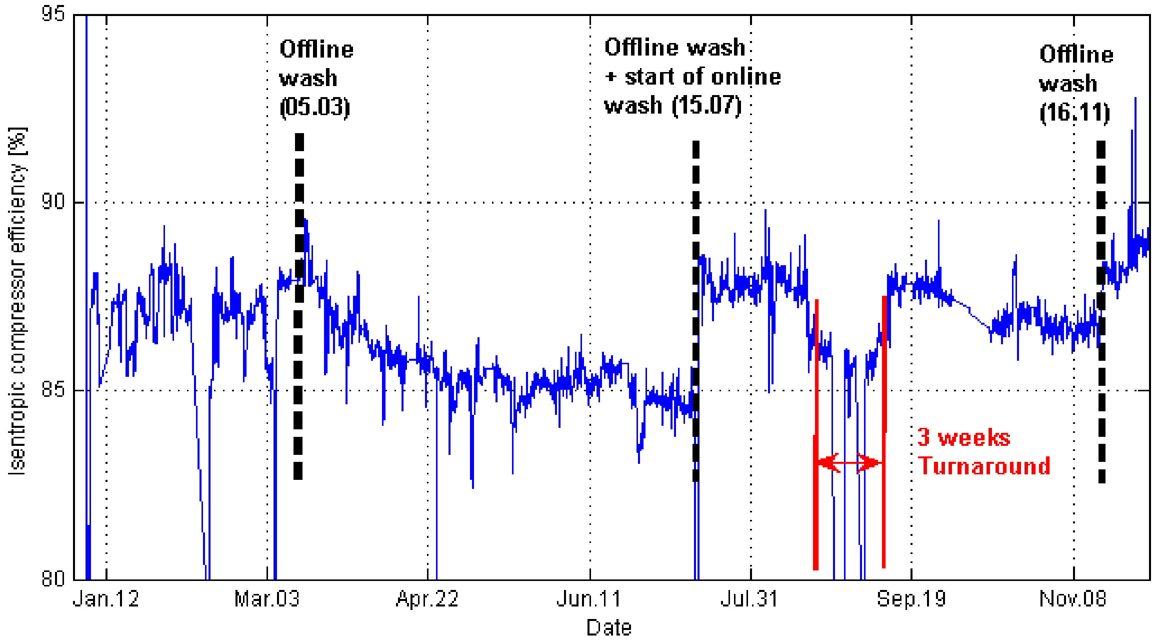


Figure 7-7: Compressor efficiency from 01.01.09-01.12.10

Figure 7-7 shows the compressor efficiency for machine 1 from the 1st of January 2009 to the 1st of December 2010. The offline washing sequences and the three weeks turnaround are indicated in the graph. The compressor efficiency during the first period clearly has a decreasing trend. The regain in performance after the offline washing sequence between the two periods evaluated is also clearly illustrated. For the second period the engine experiences a drop in compressor efficiency towards the middle of the period. This is explained by the three weeks revision shut down August/September where the engine was running at very low load. It seems like the engine has an all over higher compressor efficiency during the second period when the three weeks turnaround is left out of the evaluation.

7.3.2.1 Before installing online washing at Machine 1 - 10.03.10-10.07.10

Neither machines 1 nor 2 run online washing during this operating period, so similar deterioration trends are expected. The different performance trends for machine 1 and 2 are visualised in the same figure; machine 1 is always to the left of machine 2.

Compressor discharge pressure

Compressor discharge pressure versus corrected gas generator speed

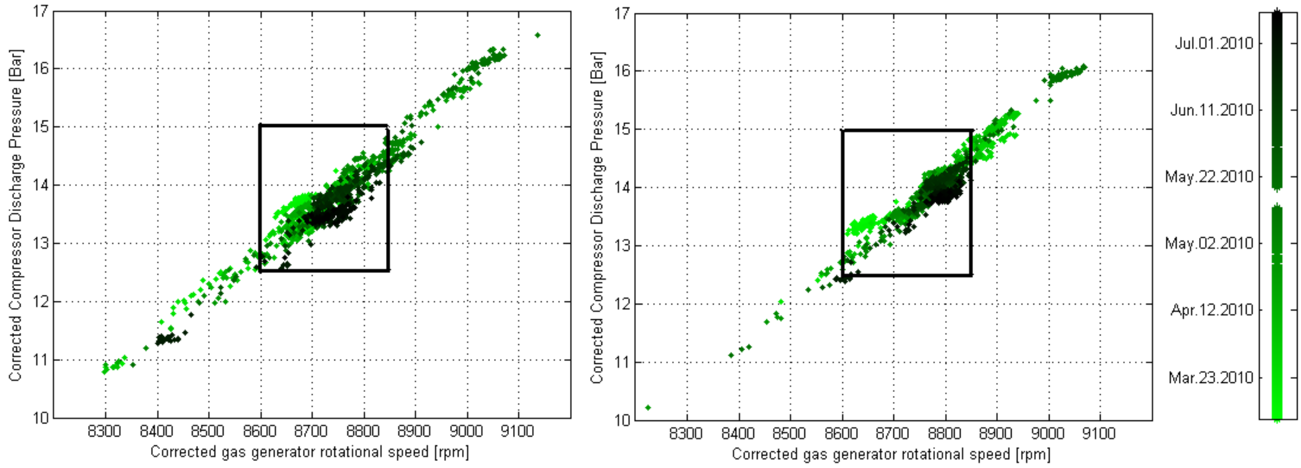


Figure 7-8: Corrected CDP vs corrected GG speed - Machine 1 and 2

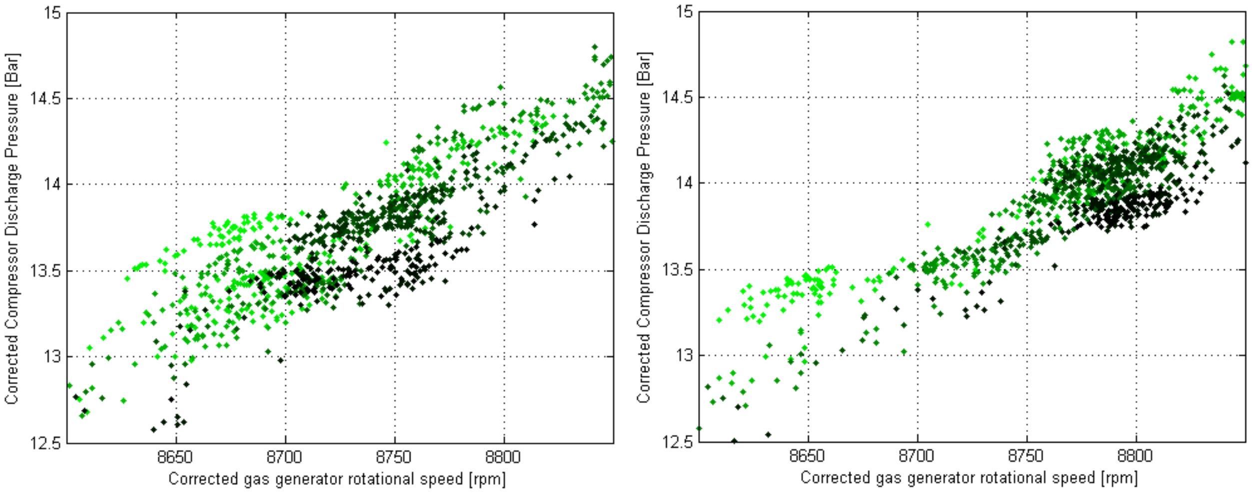


Figure 7-9: A selected area for CDPc vs corrected GG speed – Machine 1 and 2

Figure 7-8 shows corrected compressor discharge pressure plotted against corrected gas generator speed. The variation in corrected CDP is small over the interval. The marked areas in figure 7-8 are enlarged in figure 7-9 in order to better visualize possible trends. When enlarging the two areas a slightly decreasing trend in CDPc is observed for both engines when evaluated for constant GG speed.

Compressor efficiency

Corrected compressor efficiency versus corrected gas generator speed

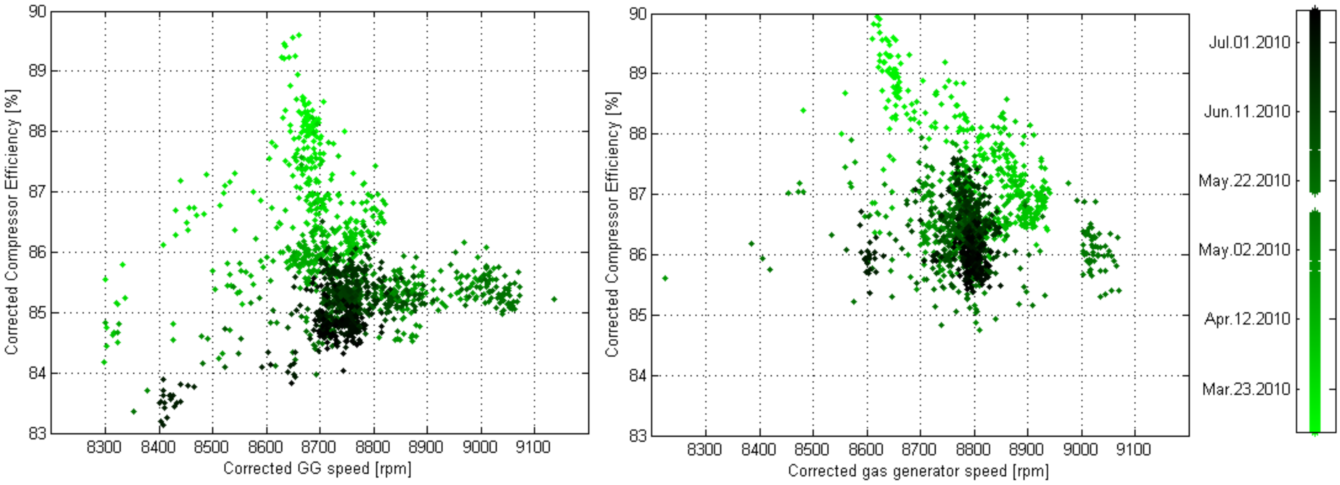


Figure 7-10: Compressor efficiency vs Corrected GG Speed- Machine 1 and 2

In figure 7-10 corrected isentropic compressor efficiency is plotted against corrected gas generator speed. Due to compressor fouling and equal operating conditions a similar reduction in compressor efficiency is expected for both engines. The data scatter is quite high for both engines, but a decreasing efficiency trend is still observed for both engines when evaluated at constant GG speed.

Exhaust gas temperature

Corrected exhaust gas temperature versus gas generator pressure ratio

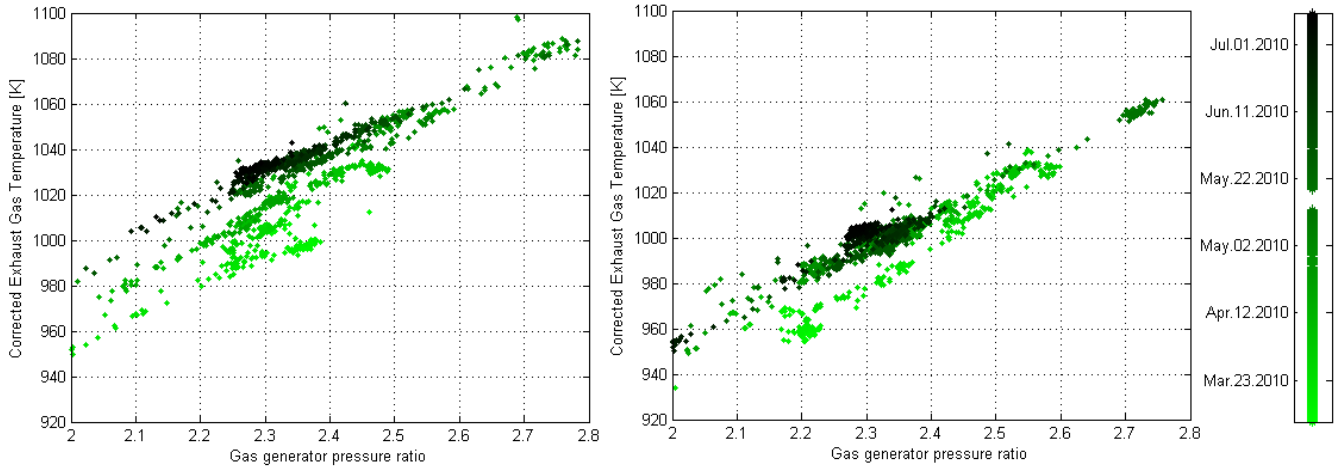


Figure 7-11: Corrected EGT vs GG pressure ratio - Machine 1 and 2

In figure 7-11 corrected exhaust gas temperature is plotted against corrected gas generator pressure ratio. Corrected EGT is higher for machine 1 than machine 2. However, both plots show distinct increasing trends in EGTc when evaluated for constant gas generator pressure ratio.

7.3.2.2 *After installing online water wash at Machine 1 - 15.07.10-12.11.10*

Due to the 3 weeks turnaround the total amount of operating hours in this period is less than for the interval before online washing was started.

Compressor discharge pressure

Compressor discharge pressure versus corrected gas generator speed

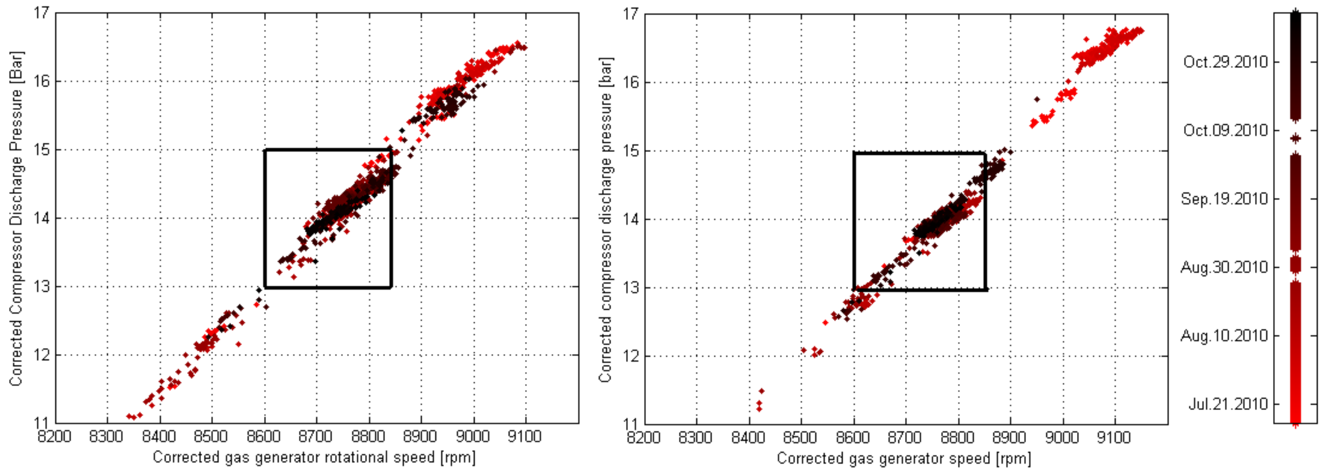


Figure 7-12: Corrected CDP vs corrected GG speed - Machine 1 and 2

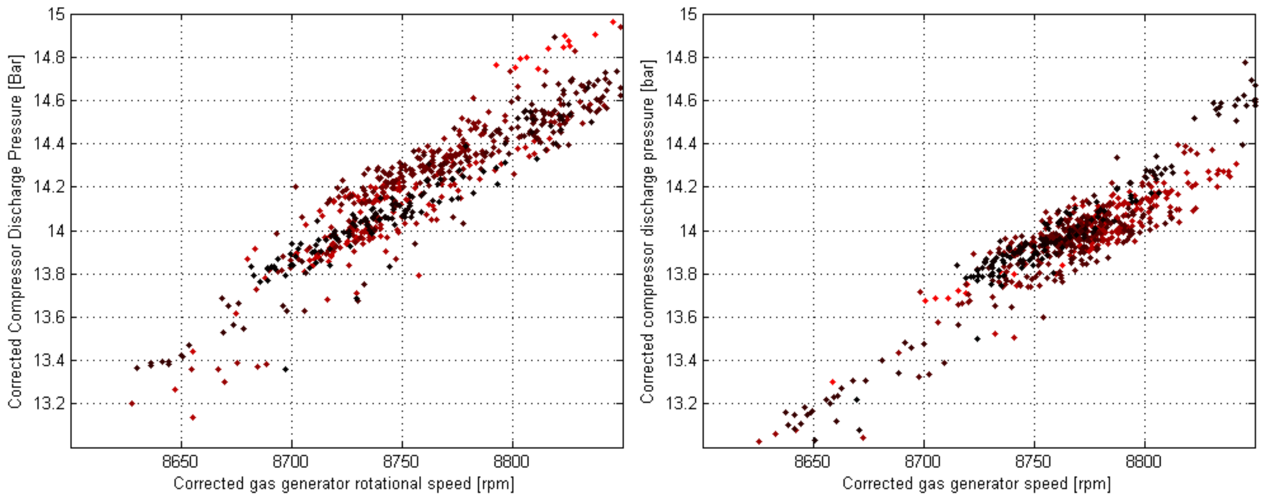


Figure 7-13: Corrected CDP vs corrected GG speed for a selected area – Machine 1 and 2

Figure 7-12 shows corrected compressor discharge pressure plotted against corrected gas generator speed. Corrected CDP varies little for the two engines and the marked areas in figure 7-12 are therefore enlarged in figure 7-13. It was expected that corrected CDP would be more stable for machine 1 than machine 2 due to online washing of machine 1. Even when enlarging the two areas no clear deterioration trends are seen.

Compressor efficiency

Corrected compressor efficiency versus corrected gas generator speed

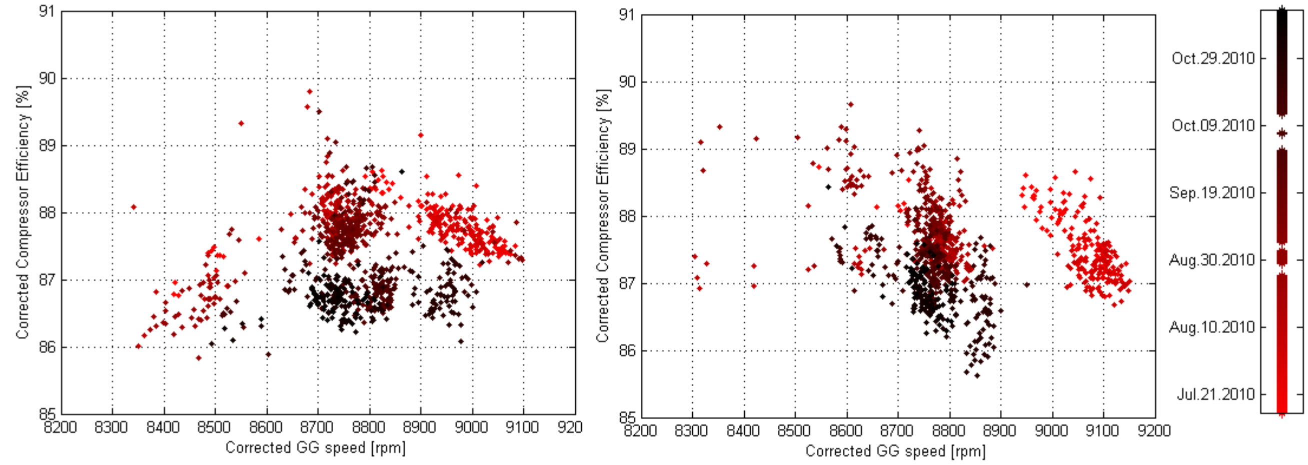


Figure 7-14: Corrected compressor efficiency vs GG speed - Machine 1 and 2

Figure 7-14 shows corrected compressor efficiency plotted versus corrected gas generator speed. From the plots above it seems like the compressor efficiency for both engines have a decreasing trend for constant GG speed. Both the efficiencies also seem to be in the same range, while in the previous period the compressor efficiency for machine 2 was higher than machine 1.

The compressor efficiency for machine 1 is higher during this operating period than during the previous period (Figure 7-10). For the previous period the compressor efficiency for machine 1 was in the range 84-88% while in this period it lays between 86 and 88%.

Exhaust gas temperature

Corrected exhaust gas temperature versus gas generator pressure ratio

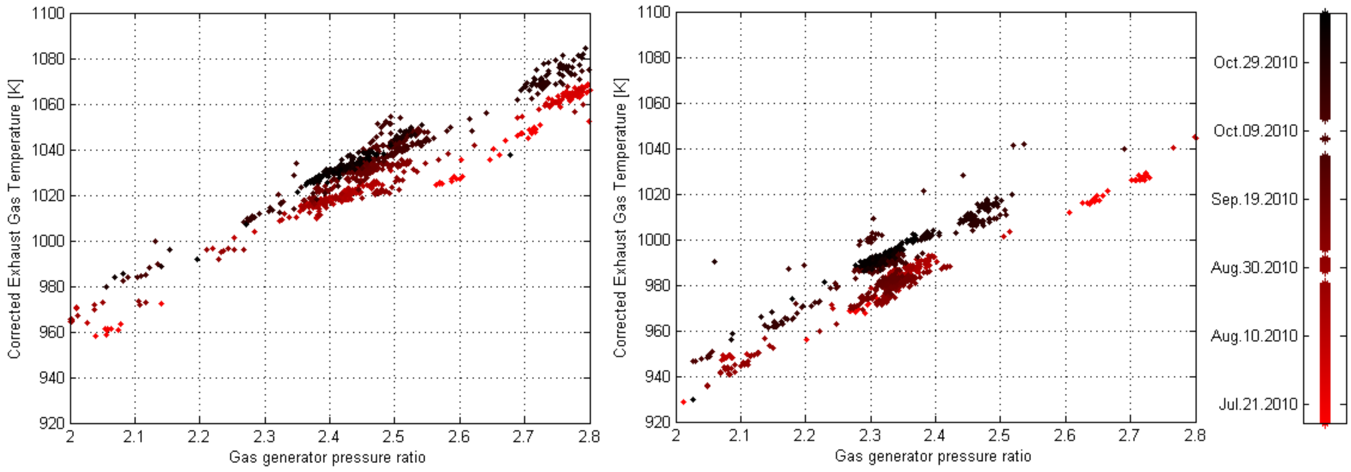


Figure 7-15: Corrected EGT vs GG pressure ratio - Machine 1 and 2

Figure 7-15 shows corrected exhaust gas temperature plotted against gas generator pressure ratio. As for the previous period the exhaust gas temperature is higher for machine 1 than machine 2. Both engines also have an increasing trend when evaluated at constant GG pressure ratio.

7.3.3 Machine Four (4)

Machine 4 is stopped every 3000 operating hours for offline washing and other maintenance. Idle wash was started in September 2010 and is run every 1000 hours between each offline wash. Machine 4 runs with redundancy making it a good machine to test idle wash on.

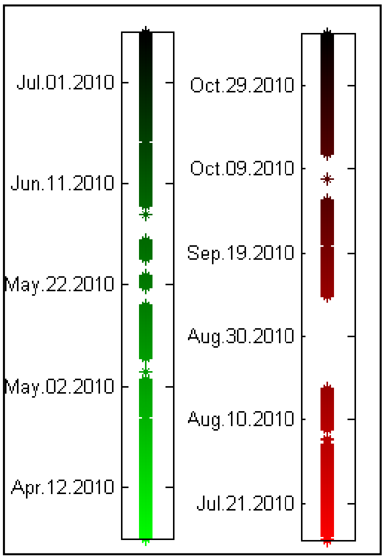
In the following section the operating period before idle wash was started is compared to the first period running with idle washing. The 12th of November the engine was stopped due to vibrations in the system. The load has to be transferred to the standby machine and machine 4 was out of operation for a period of time. In addition an extra offline wash was performed and for that reason the last period considered is less then 3000 operating hours.

Table 7-1: Offline wash and other maintenance for machine 4

Machine 4	
12.10.2009	Offline wash
28.02.2010	Offline wash, boroscope
12.07.2010	Offline wash, filter change, boroscope
21.09.2010	IDLE wash (first)
28.10.2010	IDLE wash (second)
15.11.2010	Offline wash
11.12.2010	Offline wash

Table 7-2: Periods considered for Machine 4

Period	Machine 4
Period 1	01.04.10-10.07.10 (10.03.10-31.03.10: Project shutdown revamp)
Period 2	12.07.10-10.11.10 (Turnaround (18.08-08.09.10) is left out)



Corrected isentropic compressor efficiency – 01.04.2010-10.11.2010

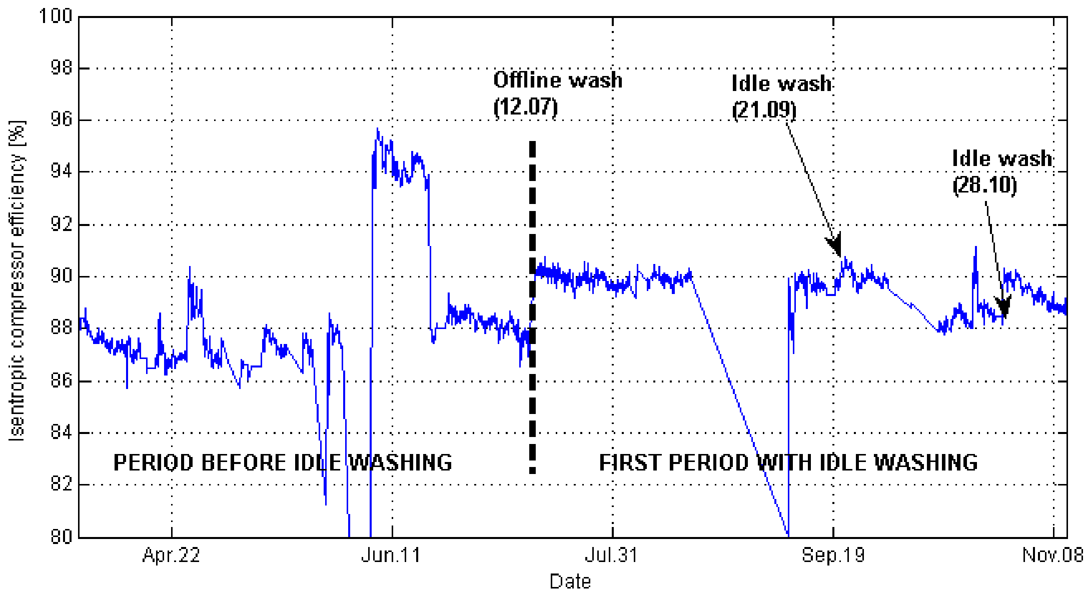


Figure 7-16: Corrected isentropic compressor efficiency versus Date- Machine 4

Figure 7-16 shows corrected isentropic efficiency for machine 4 plotted in the period from 01.04.10-10.11.10. The loss in compressor efficiency due to compressor fouling is expected to be regained after an offline wash and partly regained after an idle wash. The offline washing between the two operating periods is indicated in the figure above. The first period contains some strongly diverging points which will be left out of the evaluation. The regain in performance after the offline washing sequence between the two periods is quite well demonstrated. However, the gain from the two idle washes is not so obvious.

Compressor discharge pressure

Compressor discharge pressure versus corrected gas generator speed

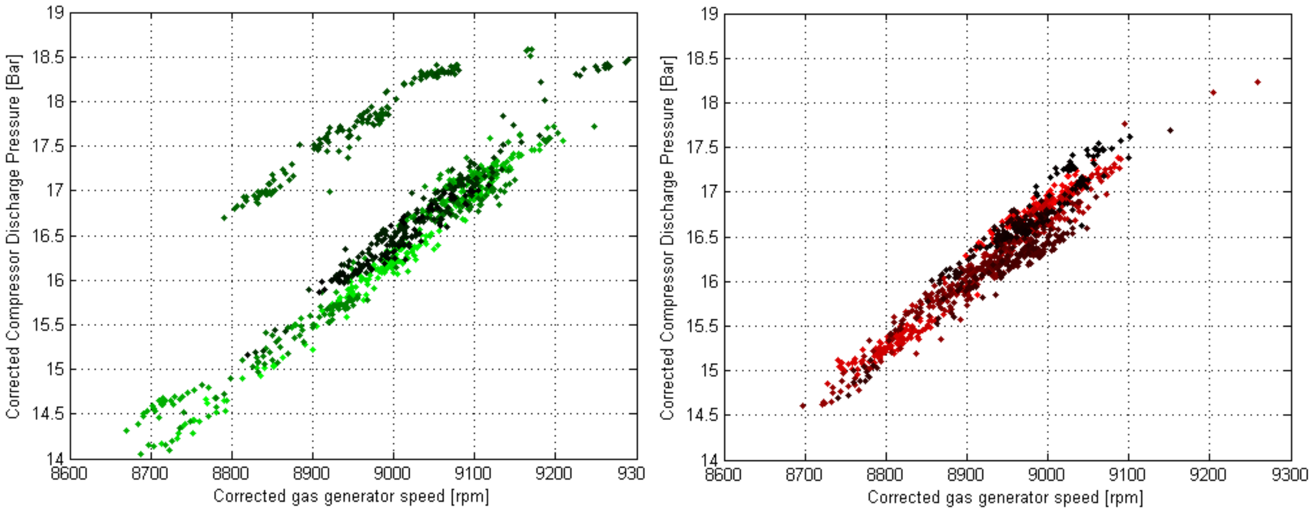


Figure 7-17: Corrected CDP vs corrected GG speed- Machine 4

Figure 7-17 shows corrected compressor discharge pressure plotted against corrected gas generator speed. When looking at the period before idle washing was started a considerable jump in CDP_c is observed. This could be related to the jump in efficiency observed in figure 7-16 and these points will not be considered when evaluating the performance trends. When leaving these points out, it actually seems like CDPc has a slightly increasing trend. This is very surprisingly since CDPc is expected to decrease due to performance deterioration. For the second period CDPc seems quite stable throughout the period.

Compressor efficiency

Corrected compressor efficiency versus corrected gas generator speed

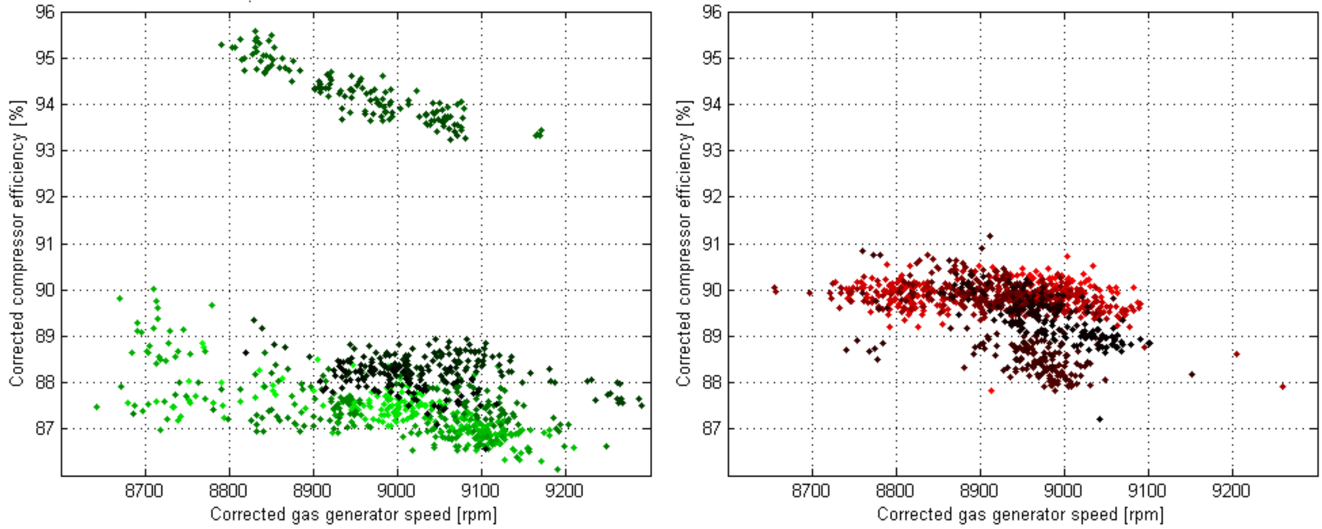


Figure 7-18: Corrected isentropic compressor efficiency vs GG speed - Machine 4

Corrected isentropic compressor efficiency is plotted against corrected gas generator speed in figure 7-18. The compressor efficiency is expected to decrease during an operating period due to fouling. However, for the first period it seems like the compressor efficiency is increasing during the period when evaluated at constant GG speed. It is mostly unlikely that a compressors performance improve during an operating period. For the second period, where idle washing is run every 1000 hour, the efficiency seems to be more stable and overall higher than during the previous period.

Exhaust gas temperature

Corrected exhaust gas temperature versus gas generator pressure ratio

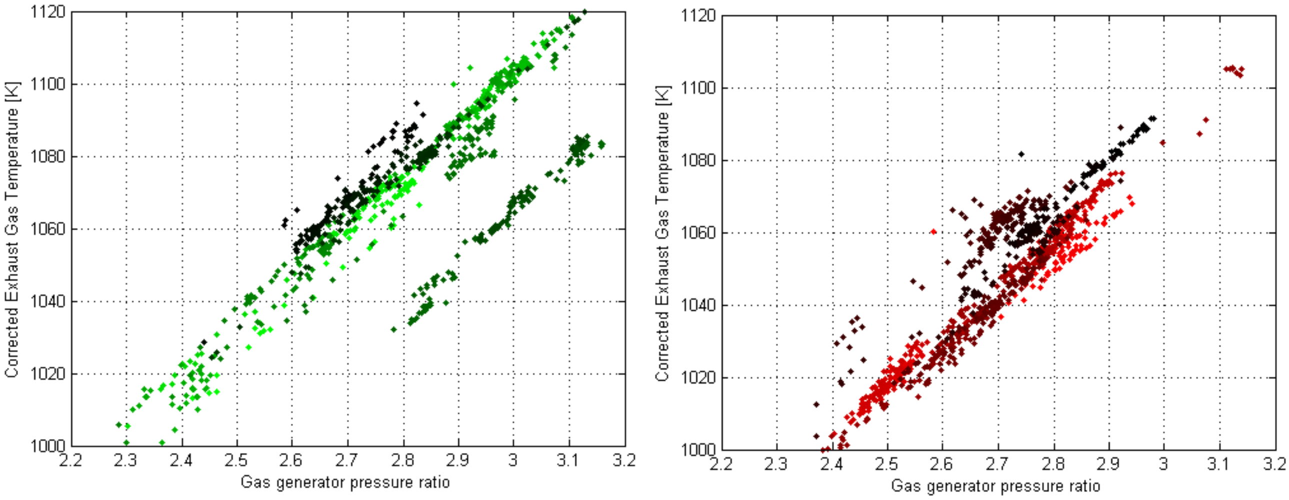


Figure 7-19: Corrected EGT vs GG pressure ratio - Machine 4

Figure 7-19 shows corrected exhaust temperature plotted against gas generator pressure ratio. In the first period EGTc has an increasing trend when evaluated for constant gas generator pressure ratio. In the second period no clear trends are observed.

7.4 Results and discussion

Machine 3 was evaluated using machine 1 as reference. The corrected compressor efficiency was evaluated at constant gas generator speed. The compressor efficiency for machine 3 seemed stable through the operating interval compared to the decreasing efficiency trend for machine 1. Machine 1 also showed a clear decreasing tendency in corrected CDP when evaluated at constant GG speed. No clear trend in CDPc was seen for machine 3. Increasing trends were observed for both engines when evaluating EGTc for constant GG pressure ratio, but the trend was more distinct for machine 1. These results show that machine 1 has a higher fouling rate then machine 3. It seems like an all over trend that the performance for machine 3 is higher then for machine 1 during the operating interval.

Machine 1 was analysed for two operating intervals using machine 2 as reference. When plotting the compressor efficiency for both periods a distinct decreasing trend was seen for the first period, while the trend seemed more stable during the second period. Further analysis showed quite equal deterioration rates for the two engines during the first operating interval. This was expected since both engines operated with equal washing routines and intervals. Both engines had decreasing trends in compressor efficiency and CDPc when evaluated for constant GG speed. Increasing trends in corrected EGT when evaluated at constant GG pressure ratio was also observed for both engines during the first operating interval.

During the second operating period the deterioration rates where not as clear as initially expected. The efficiency was evaluated for constant GG speed, and decreasing trends were observed for both engines. Even though the efficiency for machine 1 had a decreasing trend in the operating period with online washing, the compressor efficiency was all over higher during this period than the previous one. Besides efficiency trends no clear differences were seen between the deterioration rates for the two engines during this period. The second period included a three weeks turnaround and is therefore shorter than 3000 operating hours. The operation was less stable due to the shutdown during these three weeks, and it is possible that the performance trends would have been clearer if evaluated over longer period of time.

Deterioration rates for machine 4 were evaluated by comparing the operating period before idle was started with the first period running with idle washing. The compressor efficiency was plotted for both operating periods and demonstrated that the compressor efficiency was highest during the second period. However, analyses of several other parameters did not reveal any clear deterioration trends. Machine 4 was stopped during the second interval and an additional offline wash was performed. The second operating period is therefore shorter than the normal 3000 hours interval, and this could have influenced the trend. Fouling occurs gradually over time, so at the end of a shorter operating interval it is only natural that the performance is somewhat higher then at the end of a longer operating interval. Due to the time frame in this project, and the fact that only one operating interval with idle washing was available during this study, further analysis was not possible to perform. Further investigation over longer time is clearly recommended.

There are several uncertainties related to the use of performance trends. Inaccurate instrumentation and readings, uncertainties attached to the accuracy of the software (TurboWatch) etc. might influence the result. All the engines investigated in this study were only equipped with standard instrumentation which initially is meant for equipment protection and not for performing detailed analysis.

7.5 Conclusion

Deterioration rates were documented for four selected engines. Series of two parameters were plotted and the graphs were analysed. Machine 1, 2 and 3 were selected in order to document the effect of online washing. Machine 4 was selected in order to evaluate the effect of idle washing.

Machine 3 had been running continuously with online washing for a longer period. The engine was evaluated using machine 1 as reference, and when plotting the different parameters most of the performance trends indicated that online washing was contributing to improved performance.

Machine 1 had recently started online washing and was evaluated with machine 2 as reference. Machine 2 was selected as the two engines have equal filtration systems and are situated at the same level, facing the same direction. It thereby made a good reference. During the first operating period the performance trends and deterioration rates seemed quite equal for the two engines. This was expected since the engines were running with equal washing routines during this interval.

For the second period both engines also had decreasing trends in compressor efficiency even though online washing was running for machine 1. However, the compressor efficiency for machine 1 was higher for the operating interval with online washing then during the previous operating period. This could be explained by online washing, but it could also be explained by other factors like the fact that the gas turbines not were installed at the same time or errors in instrumentation.

Machine 4 was evaluated by comparing the period before idle washing started to the first period running with idle washing. The compressor efficiency for the two intervals indicated that the compressor efficiency was higher during the second period. However, most of the analysed graphs did not give any clear performance trends and more research needs to be undertaken before the effect of idle wash is clearly understood. Due to the recent start-up of idle washing and the time frame for this project only two idle wash sequences analysed. It is clearly recommended to undertake more analysis with emphasis on analysis several operating intervals before and after idle wash was started before the effect of idle washing is clearly understood.

There are several uncertainties related to performance trends. There are currently no standard parameters used in order to detect performance deterioration, and different recommendations exist. CDP was recommended as a monitoring parameter from several sources during the initial literature study. CDP also showed to give clear performance trends when evaluated for constant GG speed in this analysis. Even though the existing literature disagrees on the accuracy of compressor efficiency as a monitoring parameter, the compressor efficiency gave quite clear deterioration trends in this report.

Inaccurate instrumentation and readings and uncertainties attached to the accuracy of the software (TurboWatch) etc. might also influence the result. All engines investigated in this study were only equipped with standard instrumentation which initially is meant for equipment protection and not for performing detailed analysis.

8 The effect of performance deterioration

The function of a gas turbine is a result of the cooperation of many components which all will invariably degrade their performance. A reduction in component efficiency will have large effect on the overall performance of the gas turbine.

Compressor fouling is the major contributor to performance deterioration for gas turbines and has been the main focus of this report. Degradation in component efficiency can be caused by compressor fouling, but also by increased tip-clearance, erosion and corrosion. These effects will cause reduced maximum power available, increase the turbine operating temperature and increase the fuel consumption. It is interesting to evaluate how compressor fouling and performance deterioration of other components will affect the total gas turbine output

It is possible to establish a superior gas turbine model for detecting performance deterioration, but in order to compare performance trends and the relations between them, a very detailed gas turbine model is necessary. Due to the timeframe of this project, the main focus has been to collect and evaluate existing literature in order to investigate the relation between compressor fouling and the gas turbine performance.

8.1 Literature

During the initial literature study (see chapter 3) it was documented that compressor fouling leads to a reduction in compressor discharge pressure (CDP), reduced airflow in addition to reduced compressor efficiency when evaluated for constant gas generator speed or exhaust gas temperature (EGT). These issues are causing decreased gas turbine power output.

During the initial literature study it was also documented that CDP was considered one of the most reliable monitoring parameters for detecting compressor fouling. CDP is decided by the engine firing temperature which impacts the engine power output.

The relation between decreased CDP and gas turbine power output and efficiency was investigated. However, very little open literature exists on this field. It has not been possible to obtain much information on the relation between CDP, shaft power, total efficiency and the performance deterioration of two-shafted gas turbines. General Electric's paper on heavy-Duty gas turbines and maintenance considerations, [4], shows the effect of compressor fouling on the gas turbine performance. Figure 8-1 shows that a 5 % reduction in airflow due to fouling will cause 13 % reduction in output and 6 % increased heat rate. This indicates the importance of reducing the fouling rate in order to keep the gas turbine performance at its maximum. The graph below is valid for single-shafted gas turbines, but it still demonstrates the basic theory.

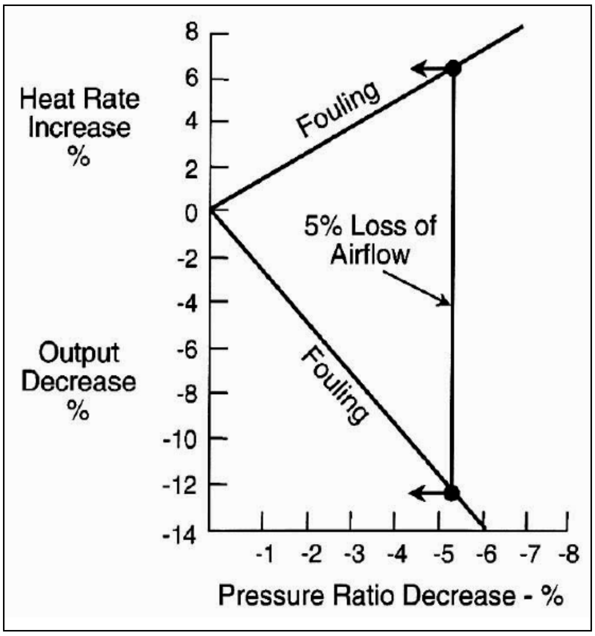


Figure 8-1: Deterioration of gas turbine performance due to compressor blade fouling [4]

Attempts were made to locate similar graphs as the one in figure 8-1 for LM2500 engines. In the General Electric's paper on GE Aero derivative gas turbines, [1], a figure for deterioration trends for LM2500 was found. This graph is shown in figure 8-2, but only illustrates long-term non-recoverable deterioration. The graph does not include losses regained by water washing e.g. compressor fouling. Operating hours is shown on the x-axis and reduction in power and increase in heat rate is shown on the y-axis. At 25000 operating hours the power deterioration is approximately 4% while the increase in heat rate is 1%. Usually HPT components are replaced after 25 000 operating intervals, which is also the case at Sleipner, and over 80% recovery can be achieved. At 50 000 hours several component replacements are done which can result in nearly 100 % restoration of the performance.

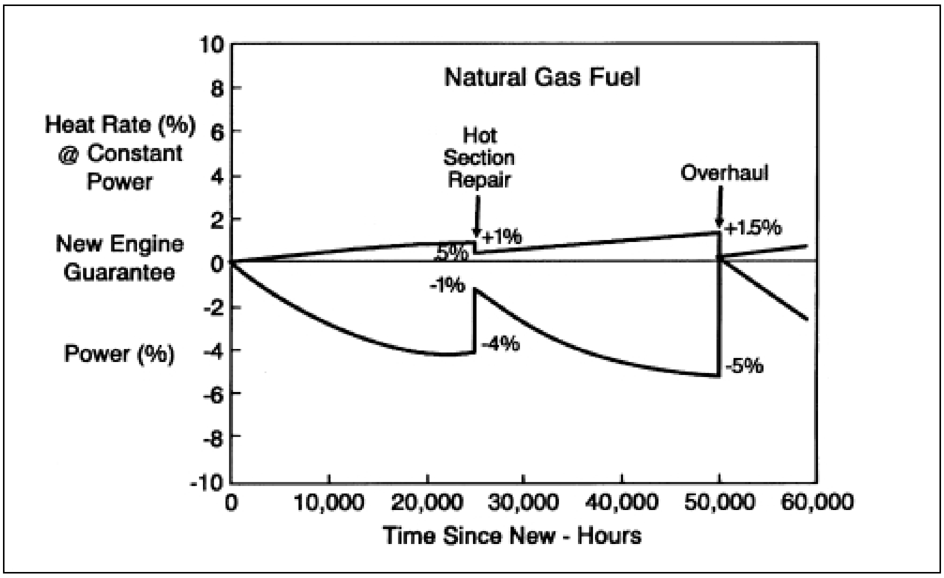


Figure 8-2: LM2500 field trends – power and heat rate deterioration [1]

8.2 General approach

The general equation for net power output

$$P_{net} \approx \dot{m}(\Delta h_t - \Delta h_{comp}) = \dot{m}((h_4 - h_{54}) - (h_3 - h_2)) \quad (8\text{-}1)$$

Enthalpy:

$$h = C_p \Delta T \quad (8\text{-}2)$$

Thermal efficiency:

$$\eta_{th} = \frac{P_{net}}{P_{combustor}} \approx \frac{\Delta h_t - \Delta h_{comp}}{\Delta h_{comb}} = \frac{c_P(T_4 - T_{54}) - c_P(T_3 - T_2)}{c_P(T_4 - T_3)} \quad (8\text{-}3)$$

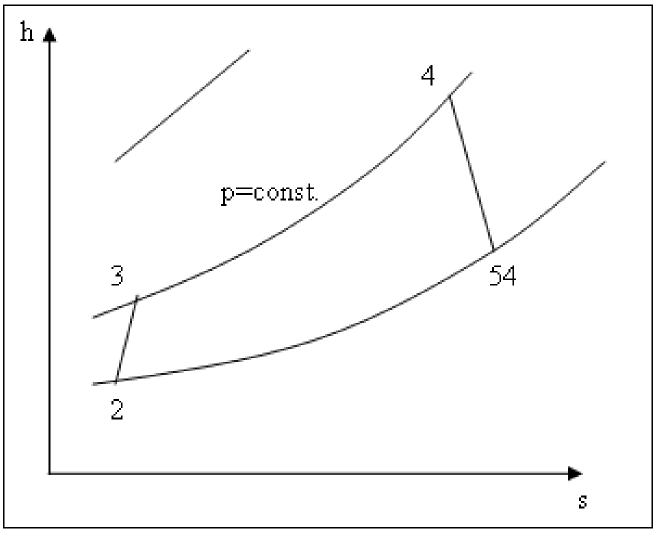


Figure 8-3: Simple h-s diagram for a gas turbine

Several parameters affect the gas turbine performance. Fouling causes decreased CDP, decreased airflow through the compressor and decreased compressor efficiency. All these parameters will lead to reduced power output. A fouled compressor will therefore cause reduced gas turbine power output and reduced thermal efficiency.

From the equations above it is quite obvious that if the compressor airflow reduces as a result of compressor fouling it will lead to decreased power output assuming that the other parameters are constant. It the power is reduced that will also lead to decreased thermal efficiency.

The h-s diagram in figure 8-3 shows the compression process through the compressor section and the expansion through the high pressure turbine (HPT). From this diagram it can be seen that if EGT (T54) increases it causes increased power produced by the high pressure turbine and thereby reduced total work and reduced thermal efficiency.

Filter systems

The inlet filtration system is also an important factor in limiting the deterioration rate. Filter systems protect the engine from airborne contaminants and thereby helps decreasing the fouling rate. However, the pressure loss over the filter system will increase over time causing reduced power and efficiency. Figure 8-2 shows the effect of increased pressure loss over the filter system for a two-shafted turbine. Form the graph below it is obvious that as the pressure drop over the filer increases this will increase the relative gas turbine power in addition to contribute to increase heat rate.

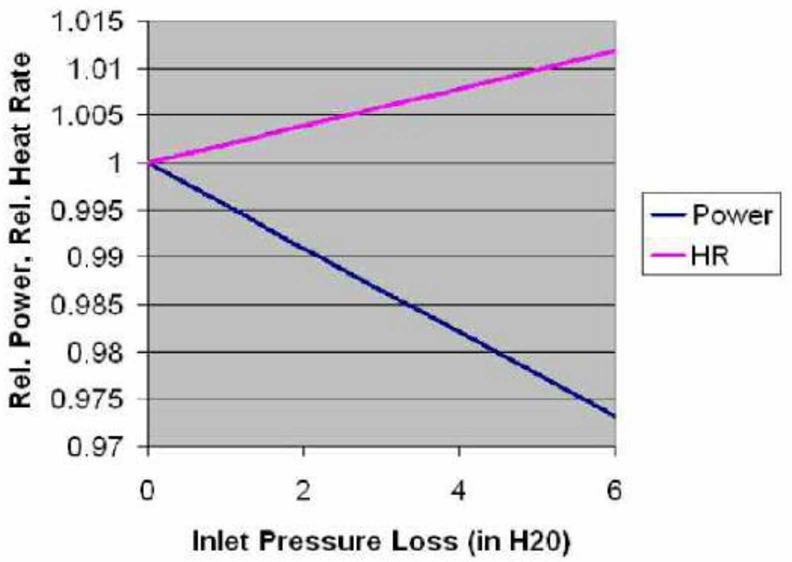


Figure 8-4 Impact of Inlet System Pressure losses on Power and Heat Rate for a Typical two shafted Gast turbine [16]

8.3 Conclusion

Compressor fouling causes reduced compressor efficiency, reduced airflow and reduced CDP which will affect the gas turbine performance. This chapter aimed to reveal the relation between compressor fouling and gas turbine performance. Due to the timeframe of this project a superior gas turbine model was not established in order to investigate this relation. However, existing literature were collected and a general approach was described. No calculations were performed, but the tendency of reduced power output and reduced thermal efficiency as a result of compressor fouling was shown.

Very little open literature exists on the relation between CDP, power and thermal efficiency. The one graph found showed that reduced airflow as a result of compressor fouling contributed to a considerable reduction in gas turbine power output and increased heat rate. This graph was only valid for single-shaft turbines, but still gives an implication of the large consequences of compressor fouling. Only one graph was found for two-shafted gas turbines, and only for non-recoverable deterioration. The graph showed that non-recoverable deterioration also contributes to reduced power output and a small increase in heat rate. However, the effects of non-recoverable deterioration are smaller than for compressor fouling and occur over a longer time.

Gas turbines operating at offshore installation have inlet filtration systems. The filter systems reduce the fouling rate by limiting the amount of contaminants entering the gas turbine. However, as other components filter systems experience degradation over time. Filter deterioration causes increased pressure drop over the filter which causes an additional reduction in power and an increase in heat rate.

When looking at the performance deterioration graph for single-shafted gas turbines in addition to the general approach, it is quite obvious that compressor fouling greatly affects the overall gas turbine performance. This implies that the main focus regarding performance deterioration should be to reduce of the fouling rate, and that the evaluation of online washing is valuable in order to do so.

9 Conclusion

The aim of this project was to map and document deterioration rates for selected gas turbines in order to investigate the potential for improving efficiency rates through online and idle washing by monitoring several parameters.

A validation of the monitoring software, TurboWatch, was performed. The validation clearly demonstrated the importance of having in-depth knowledge of calculations and equations on which a monitoring program is based. Only when having all the background information (current baselines, equations, calculations etc.) confident decisions based on this software can be made. The validation of TurboWatch clearly demonstrated that the wrong notation was being used for the compressor efficiency for machine 3 investigated in the present study. This has implications due to the considerable difference between polytropic and isentropic efficiency. Inconsistent use of efficiencies could cause significantly effects when calculating the performance. The choice of efficiency does not affect the deterioration trend it self, but the utilisation needs to be consistent.

The validation of TurboWatch included an evaluation of the existing water wash panel. The number of operating hours in this panel turned out to be wrong and should be removed from the panel. This information also exists in other software programs and should be withdrawn from there. The water wash panel is also using relative differences which are based on baselines for general engines. However, all engines behave differently, so when analysing performance deterioration, relative differences are only useful when the current baseline is known and corrected to the current gas generator.

Gas turbine deterioration rates were evaluated for four selected gas turbines. Machine 3 had been running with daily online washing since 2009, and by analysing several parameters recommended for monitoring performance deterioration, online washing seemed to have good effect. The performance was more stable during the operating interval compared to the reference engine which was only run with regular offline washing routines.

Machine 1 had recently started online washing and was evaluated using machine 2 as reference. Machine 2 was selected as the two engines have equal filtration systems and are situated at the same level facing the same direction. It thereby made a very good reference. During the first operating period the performance appeared as expected; both engines showed equal signs of performance deterioration.

During the second operating period machine 1 was running with daily online washing, and improved performance was expected for this machine. However, no clear trends were detected in corrected compressor discharge pressure (CDP) for any of the engines when evaluated at constant gas generator speed. The compressor efficiency had decreasing trends for both engines, but the compressor efficiency for engine 1 was higher during this period than the previous. Borescope pictures taken after the operating period with daily online washing on machine 1 showed promising results; the compressor blades appeared much cleaner in machine 1 than machine 2 after the 3000 hour operating interval. Considering this fact, in relation to increase efficiency for machine 1, these are promising indications as to the effect of online washing. In order to get better estimates and validations, further studies over longer time and for several operating intervals are recommended.

Machine 4 was evaluated by comparing the period before idle wash started to the first period running with idle washing. The compressor efficiency was plotted and indicated that the compressor efficiency was higher during the second period where idle washing was running every 1000 hours. However, most of the analysed graphs plotted for this study did not reveal any clear deterioration trends. Due to the recent start-up of idle washing at Sleipner and the time frame for this project, only two idle wash sequences was analysed. Further investigations over a longer time span is clearly recommended. Since no clear deterioration trend is demonstrated it is not at present possible to conclude whether idle washing or online washing at normal operating load is most effective. However, it is important to note that idle washing causes small production losses as well as increased fuel consumption due to start-up of the standby machine. This implies that the performance gain from idle washing needs to be higher than for online washing at normal operating load in order for idle washing to be the most profitable method.

There are several uncertainties related to the use of performance trends. Inaccurate instrumentation and readings, uncertainties attached to the accuracy of the software (TurboWatch) etc. might influence the result. All the engines investigated in this study were only equipped with standard instrumentation. To fully reveal the performance gain of online washing and to achieve more precise performance trends, extra test instrumentation should been installed.

A main conclusion to be drawn form this study is that no negative effects has been documented by initiating online washing. It causes some extra work to operate the system manually, but the sequence is not very time-consuming for the operators to run. As most analysis points towards a performance gain, it is strongly recommended to carry on further testing to consider installation of online washing on additional turbines. The consequences of compressor fouling have shown to affect the gas turbine performance greatly which again shows the importance of focusing on decreasing the fouling rate.

10 Recommendations for future work

The literature study performed initially during this project revealed a general understanding that the pressure loss over the bellmouth is a good and stable parameter for detecting performance deterioration. In order to measure this pressure loss a simple manometer is needed. It could therefore be considered to install a manometer on one of the engines at Sleipner for further analysis of the effect of online washing. Since there are different uncertainties related to performance trends and the accuracy of the instrumentation, it is recommended to perform a calibration of instrumentation before further testing and analyses.

It was also recommended to carry on further investigations during a longer time-span for both machines 1 and 4. Online washing for machine 1 should be investigated over several operating periods in order to better document the effect. Since the reference engine (machine 2) used in this study now has been set for online washing both should be investigated for further work.

If the performance gain of online washing is clearly proven through further analysis, online washing at engines running at peak load would be interesting to test in order to evaluate cost savings. Engines running at part load only save fuel by installing online washing. However, compressor trains where the turbine runs at temperature control (peak load) would have fuel saving and increased production. It is, however, challenging to measure the amount of fuel gas.

Depending on the result from future evaluations by using more online washing, a test instrumented gas turbine in operation would certainly give more exact parameters and performance trends. Such an installation is costly and would require extra work and logistics, but would be possible to install for example during turbine replacements. It would all depend on a cost/benefit analysis in order to evaluate potential savings.

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[29] Langtidstest av høytrykks on-line vannvasksystem på LM2500PE, generator turbin B, på Sleipner A.

12 Appendix

A: Script for importing parameters from MATLAB to Excel

```
% Importing values from Excel
[driftsdata]=xlsread('Machine 3-5-10.03.10-10.07.10.xls');

%Date: Needs to be syncronized with MatLab
Date=driftsdata(:,2)+ datenum('30-Dec-1899');

%Operating hours
Hours=driftsdata(:,1);

%EPC:compressor efficiency
EPC=driftsdata(:,4);

%Massflow air [kg/s]
ma=driftsdata(:,15);

%Corrected massflow air from TW [kg/s]
macTW=driftsdata(:,16);

%Fuelflow [kg/h]->[kg/s]
mf=driftsdata(:,17)/3600;

%corrected fuelflow [kg/h]->[kg/s]
mfcTW=driftsdata(:,19)/3600;

%GG speed
N1=driftsdata(:,21);

%Corrected GGspeed from TurboWatch
N1CTW=driftsdata(:,22);

%P0=ambient pressure (bar)
P0=driftsdata(:,27)/1000;

%P2=Compressor inlet pressure 80 (mbarg->bara)
P2=P0+driftsdata(:,28)/1000;

%P3(CDP)=Compressor discharge pressure(bar)
P3=driftsdata(:,29)+P0;

%Corrected P3
P3TW=P0+driftsdata(:,30);

%p54
p54=driftsdata(:,32);
%Corrected p54
p54TW=driftsdata(:,33);

%T0: Ambient temperature
T0=driftsdata(:,53);

%T2: compressor inlet temperature (K)
T2=driftsdata(:,54)+273.15;

%T3(CDT): Compressor discharge temperature
```

```
T3=driftsdata(:,55)+273.15;

%T3A corrected
T3TW=driftsdata(:,56)+273.15;

%T54 Exhaust gas temperature
T54=driftsdata(:,69)+273.15;

%T54A corrected
T54TW=driftsdata(:,70)+273.15;

%Shaft power PT [kW]
SPPT=driftsdata(:,49);

%Kappa for luft
k=1.4;

%Compressor efficiency calculated (ln benevnes log i matlab)
CCE=((k-1)/k)*(log(P3./P2)./log(T3./T2));

%Temperature correction
Theta=T2/(15+273.15);

%Pressure correction
Delta=(P2)/(1.01325);

%Corrections
T54c=T54./(Theta).^0.85;
T54c2=T54./(Theta);
T54c3=((T54./(T2./288))-288);
N1c=N1./sqrt(Theta);
P3c=0.985*P3./Delta;
EPCc=EPC./(Theta).^0.0312;
T2c=(T2)./(Theta);
T3c=(T3)./(Theta);
mac=((ma).*sqrt(Theta))./Delta;
SPPTc=SPPT./(Delta.*sqrt(Theta));

%Pressure ratio
Pr=P3./P2;
%GG Pressure ratio
GGPr=p54./P2;
```

B: Plotting corrected CDP versus corrected EGT

```
start=2; % Start row in Excel
slutt=1362; %End row in Excel

a=(slutt-start);
int=1/a;
color=zeros(3,3); %Creating a matrix
for i= 1:a  %generates green -> black colour
    color(start+i,2)=(1-int.*i);
end

figure

for i=start:slutt
    h=plot(T54c(i),P3c(i),'.'); %Plots two parameters against eachother
    set(h,'color',color(i,:))
    xlabel('Corrected Exhaust Gas Temperature [K]')
    ylabel('Corrected Compressor Discharge Pressure [Bar]')
    axis([850 1100 9 17])
    grid on
    hold on
end
```

C: Script for chosen intervals of EGT or N1c

```
start=2;
slutt=1362;
Date2=1;
cdp=1;
j=1;
for i=start:slutt
    if T54c(i)>=983.15  %Chooses the lower limit for a interval for T54
        if T54c(i)<=988.15 Chooses the higher limit for a interval for T54
         cdp(j)=P3c(i);
         Date2(j)=Date(i);
            j=j+1;
        end
    end
end

figure
plot(Date2,cdp,'m-*') % Plot date vs CDP
datetick('x','dd.mm.yy','keepticks')
axis tight
grid on
ylim([12 15])
p=polyfit(Date2,cdp,1);

hold on

yfit=polyval(p,Date2);
plot(Date2,yfit,'black','linewidth',2)

hold off

ylabel('CDPc [bar]')
xlabel('Date')
title('EGTc between 983.15 and 988.15 K')
```

D: Water wash and other maintenance

Table 12-1: Water wash and other maintenance machines 1 and 3

Offline wash and other maintenance			
Machine 1		**Machine 3**	
30.12.2008	Offline wash	13.02.2008	Offline wash
08.04.2009	GG was changed	02.04.2008	Extra offline wash
29.04.2009	Offline wash, filter change	14.06.2008	Offline wash
24.06.2009	Offline wash	20.08.2008	Offline wash
10.11.2009	Offline wash	24.10.2008	GG and filter change
05.03.2010	Offline wash, filter change	01.01.2009	Offline wash
14.07.2010	Offline wash, filter, borescope	25.03.2009	Offline wash
16.11.2010	Offline wash, borescope	29.04.2009	01.05:Online wash started
		24.07.2009	Offline wash
		13.11.2009	Offline wash, filter change
		04.03.2010	Offline wash, filter change
		16.07.2010	Offline Wash, borescope
		17.11.2010	Offline wash, borescope, filter

Table 12-2: Periods considered for machines 1 and 3

Period	Machine 1	Machine 3
Period 1	01.01.09 - 07.04.09	02.01.09 - 24.03.09
Period 2	09.04.09 - 23.06.09	26.03.09 - 22.07.09
Period 3	01.07.09 - 08.11.09	26.07.09 - 09.11.09
Period 4	14.11.09 - 03.03.10	14.11.09 - 03.03.10
Period 5	10.03.10 - 10.07.10	10.03.10 - 10.07.10

Table 12-3 Maintenance for machine 1 and 2

Offline wash and other maintenance			
Machine 1		**Machine 2**	
30.12.2008	Offline wash	05.01.2009	Offline wash
08.04.2009	GG was changed	26.03.2009	Offline wash
29.04.2009	Offline wash, filter change	25.07.2009	Offline wash
24.06.2009	Offline wash	08.11.2009	Offline wash, filter change
10.11.2009	Offline wash	03.03.2010	GG turbine was changed
05.03.2010	Offline wash, filter change	14.07.2010	Offline wash
14.07.2010	Offline wash, filter, borescope	15.11.2010	Offline wash, filter, borescope
16.11.2010	Offline wash, borescope		

E: Performance maps TURBOWATCH

Machine 3: CDP VS T54

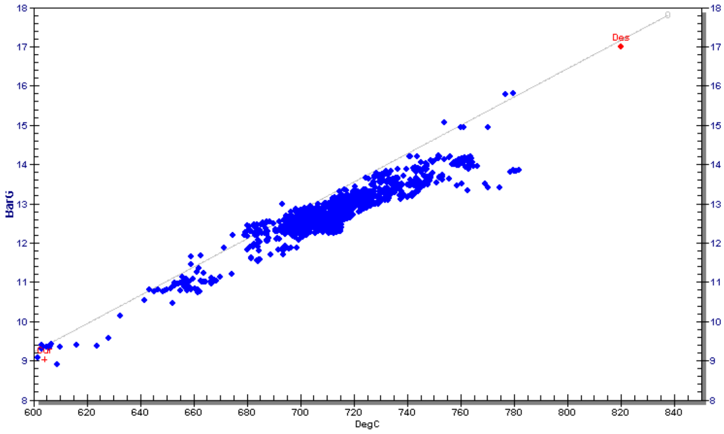


Figure 12-1: Performance maps – CDP vs T54 for Machine 3

Machine 3: p54 vs T54

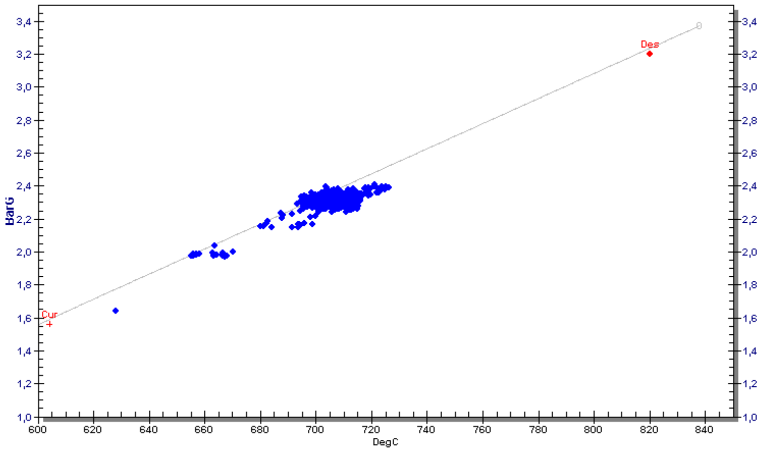


Figure 12-2: Performance maps – PT inlet pressure vs T54 for Machine 3

Machine 1: CDP vs T54

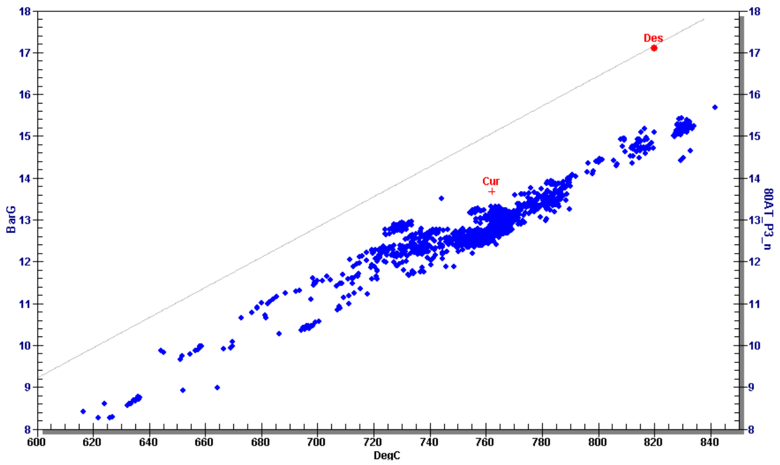


Figure 12-3: Performance maps – CDP vs T54 for Machine 1

Machine 1: p54 vs T54

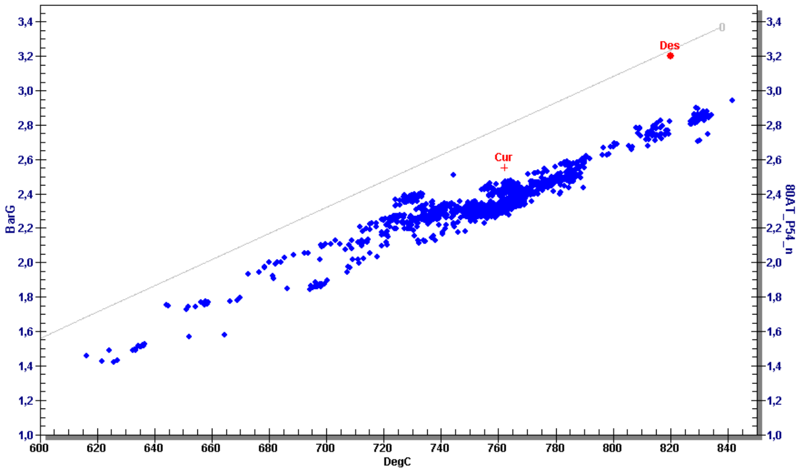


Figure 12-4: Performance maps – PT inlet pressure vs T54 for Machine 1

F: Efficiency calculations – TurboWatch and Hysys

Machine 3 – Period 5 (10.03.-10.07.2010)

1. Point (row 2 Excel file)
P0=1023,5 mBARA
P2= -10,92188 mBARG = 1,0125 bar
P3= 12,99219 BARG = 14,01569 bar
T2= 12,75°C
T3=355,25°C

Parameter	Value	Deviation from TurboWatch
Polytropic compressor efficiency (TurboWatch)	91,23	-
Polytropic compressor efficiency (Hysys)	94,07	3,11%
Adiabatic compressor efficiency (Hysys)	91,72	0,54%
Polytropic Head [m] (Hysys)	33660	

2. Point (row 390 Excel File)
P0=1028,563mBARA = 1,028563 BARA
P2= -10,375 mBARG = 1,018188 bar
P3= 14,05469 BARG = 15,083253 bar
T2= 6,09375°C
T3=364,875°C

Parameter	Value	Deviation from TurboWatch
Polytropic compressor efficiency (TurboWatch)	88,18	-
Polytropic compressor efficiency (Hysys)	92,22	4,58%
Adiabatic compressor efficiency (Hysys)	89,10	1,04%
Polytropic Head [m] (Hysys)	33270	

3. Point (row 610 Excel File)
P0=1004,75mBARA = 1,00475 BARA
P2= -10,07031 mBARG = 0,994679 bar
P3= 12,53906 BARG = 13,54381bar
T2= 8,53125 °C
T3=352,8438°C

Parameter	Value	Deviation from TurboWatch
Polytropic compressor efficiency (TurboWatch)	88,69	-
Polytropic compressor efficiency (Hysys)	92,30	4,07%
Adiabatic compressor efficiency (Hysys)	89,29	0,67%
Polytropic Head [m] (Hysys)	33140	

4. Point (row 860 Excel File)
P0=1002,313mBARA = 1,002313 BARA
P2= -10,34375mBARG = 0,991969 bar
P3= 12,96875 BARG = 13,97106 bar

T2= 14,03125 ^{o}C
T3=362,9063 ^{o}C

Parameter	Value	Deviation from TurboWatch
Polytropic compressor efficiency (TurboWatch)	90,72	-
Polytropic compressor efficiency (Hysys)	94,00	3,61%
Adiabatic compressor efficiency (Hysys)	91,61	0,98%
Polytropic Head [m] (Hysys)	33270	

Machine 1 –Period 5 (10.03.-10.07.2010)

1. Point (row 25 Excel File)

P0=1010,75mBARA = 1,01075 BARA
P2= -8,3125mBARG = 1,0024375 bar
P3= 12,83594 BARG = 13,84669 bar
T2= 5,4373 ^{o}C
T3=350,75 ^{o}C

Parameter	Value	Deviation from TurboWatch
Adiabatic compressor efficiency (TurboWatch)	88,19	0,84%
Adiabatic compressor efficiency (Hysys)	88,93	
Polytropic compressor efficiency (Hysys)	92,05	
Polytropic Head [m] (Hysys)	33090	

2. Point (row 200 Excel File)

P0=986,1875mBARA = 0,9861875 BARA
P2= -6,664063mBARG = 0,97952bar
P3= 12,23438 BARG = 13,220568 bar
T2= 5,46875 ^{o}C
T3=353 ^{o}C

Parameter	Value	Deviation from TurboWatch
Adiabatic compressor efficiency (TurboWatch)	86,53	0,94%
Adiabatic compressor efficiency (Hysys)	87,34	
Polytropic compressor efficiency (Hysys)	90,87	
Polytropic Head [m] (Hysys)	32850	

3. Point (row 410 Excel File)

P0=1022,563 mBARA = 1,022563 BARA
P2= -6,992188mBARG = 1,0155708bar
P3= 12,33594 BARG = 13,358503 bar
T2= 7,5 ^{o}C
T3=354,5 ^{o}C

Parameter	Value	Deviation from TurboWatch
Adiabatic compressor efficiency (TurboWatch)	86,05	1%
Adiabatic compressor efficiency (Hysys)	86,91	
Polytropic compressor efficiency (Hysys)	90,54	
Polytropic Head [m] (Hysys)	32670	

4. Point (row 700 Excel File)

P0=1014,5mBARA = 1,0145 BARA

P2= -8,10563mBARG = 1,006398bar

P3= 13,86719 BARG = 14,88169 bar

T2= 6,6873 ^{o}C

T3=377,75 ^{o}C

Parameter	Value	Deviation from TurboWatch
Adiabatic compressor efficiency (TurboWatch)	85,25	0,84 %
Adiabatic compressor efficiency (Hysys)	85,97	
Polytropic compressor efficiency (Hysys)	89,97	
Polytropic Head [m] (Hysys)	33300	

G: Calculations of Corrected parameters HYSYS

Corrected gas generator speed (N1c=N1/√ θ)

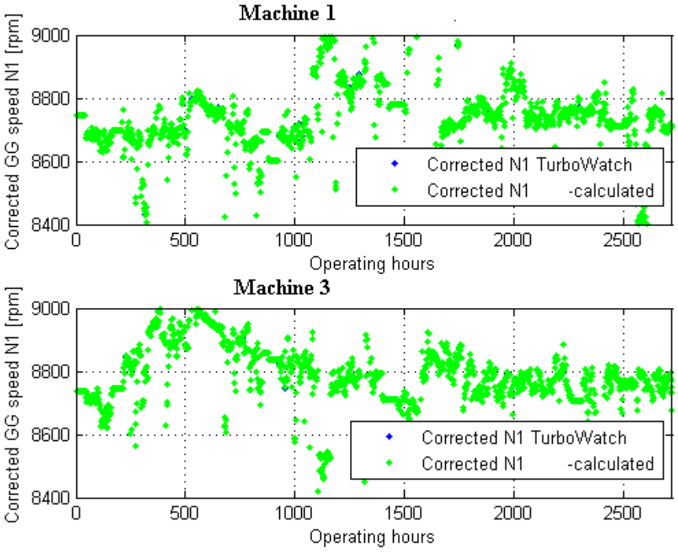


Figure 12-5: Corrected GG speed – general equation - Machine 1 and 3

Corrected compressor discharge temperature (CDTc=CDT/θ)

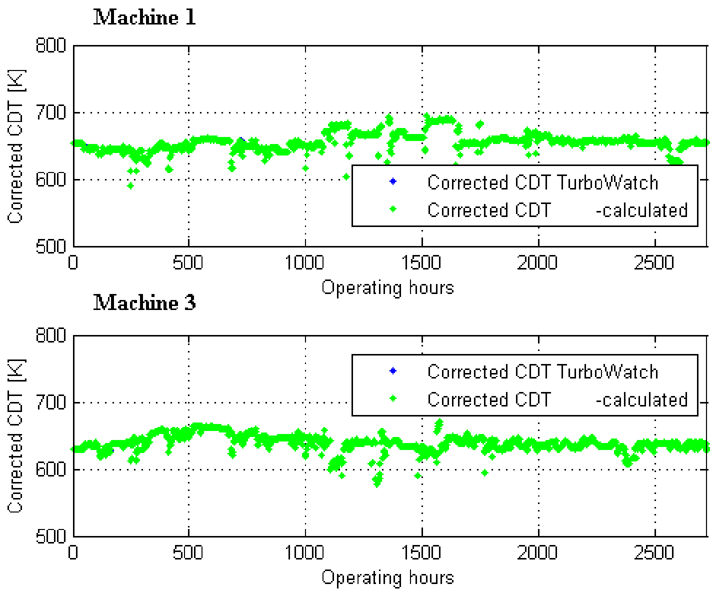


Figure 12-6: Corrected CDT – general equation - Machine 1 and 3

Corrected exhaust gas temperature (EGT/T54) when using the specific equation (T54c=T54/θ^0.85)

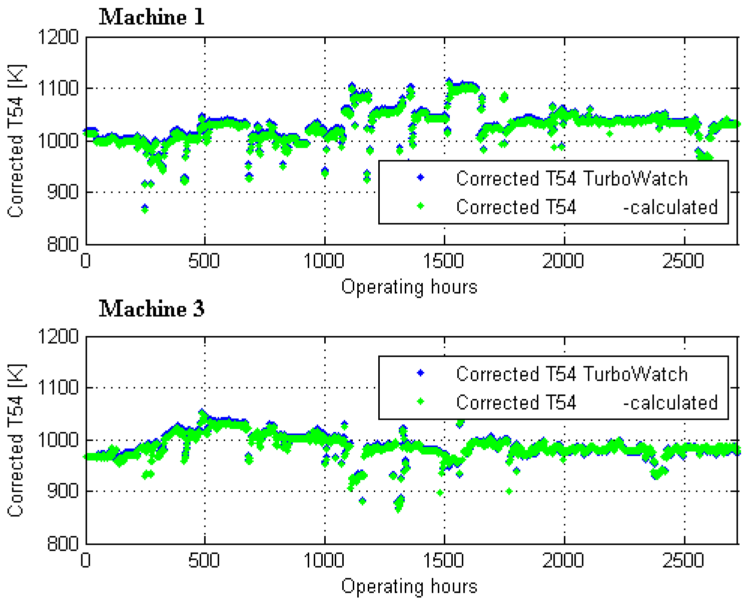


Figure 12-7: Corrected T54 – specific equation – Machine 1 and 3

Corrected exhaust gas temperature (EGT/T54) when using the general equation- T54c=T54/θ

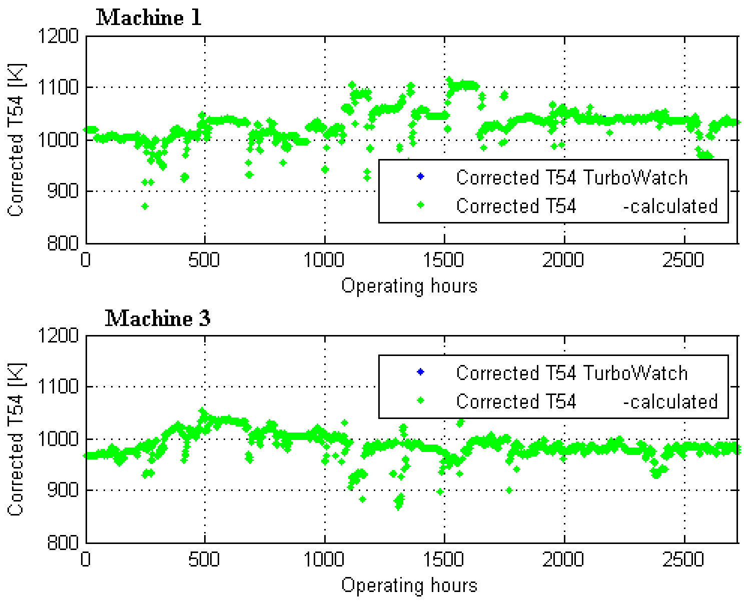
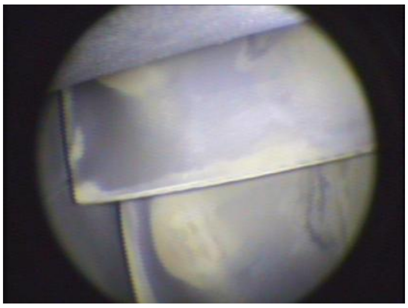
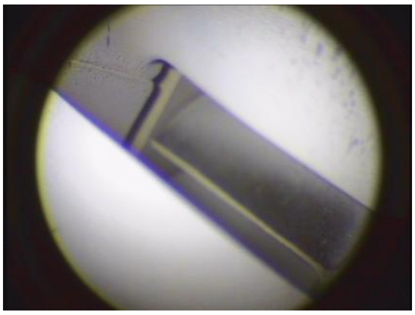
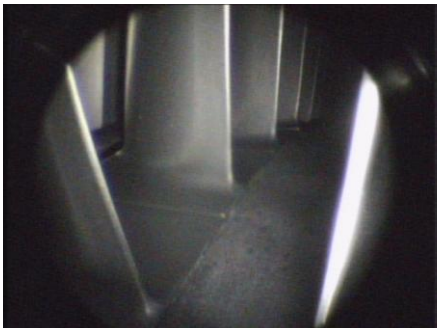


Figure 12-8: Corrected T54 - general equation – Machine 1 and 3

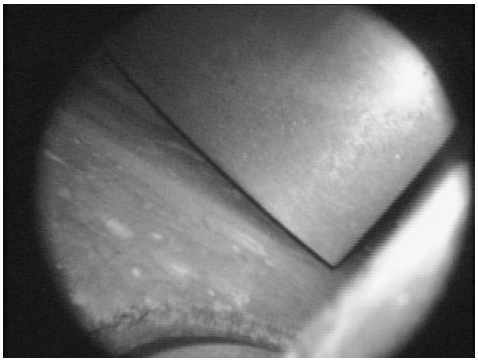
H: Borescope inspection (the compressor section) 14.-16.November 2010

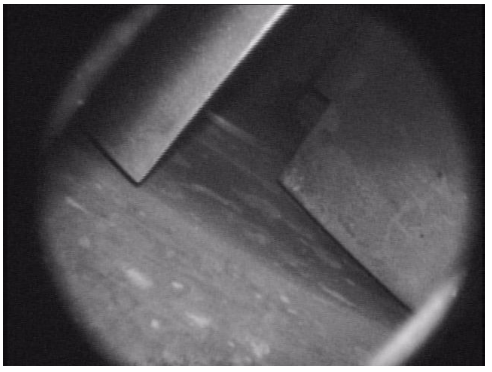
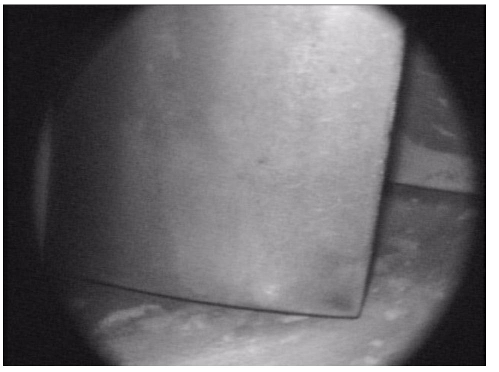
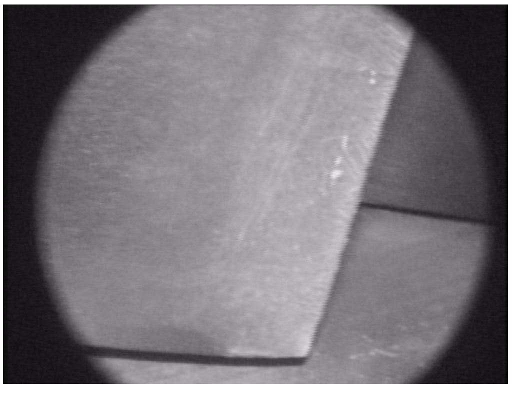
Machine 1

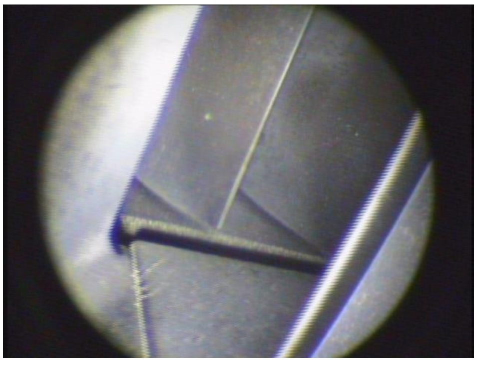
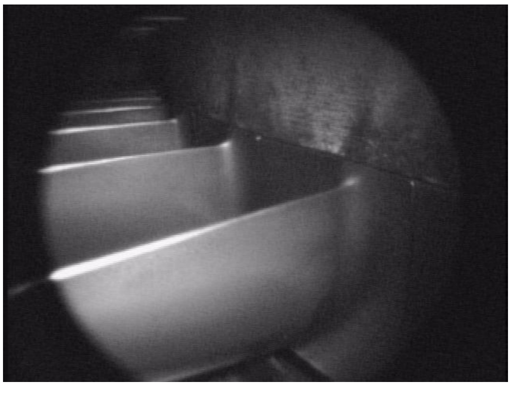
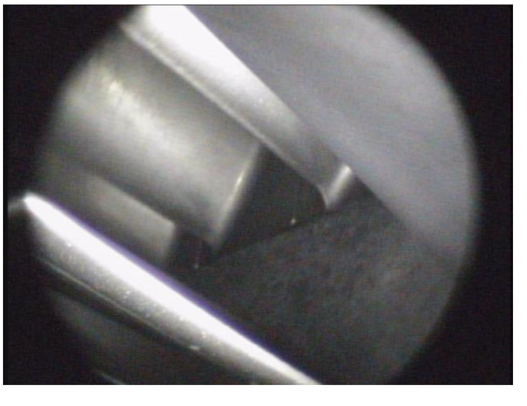


Machine 2

Machine 3

I: Deterioration rates

COMPRESSOR AIR FLOW

Machine3 versus Machine1 (10.03-10.07.10) - Corrected air flow versus corrected GG speed

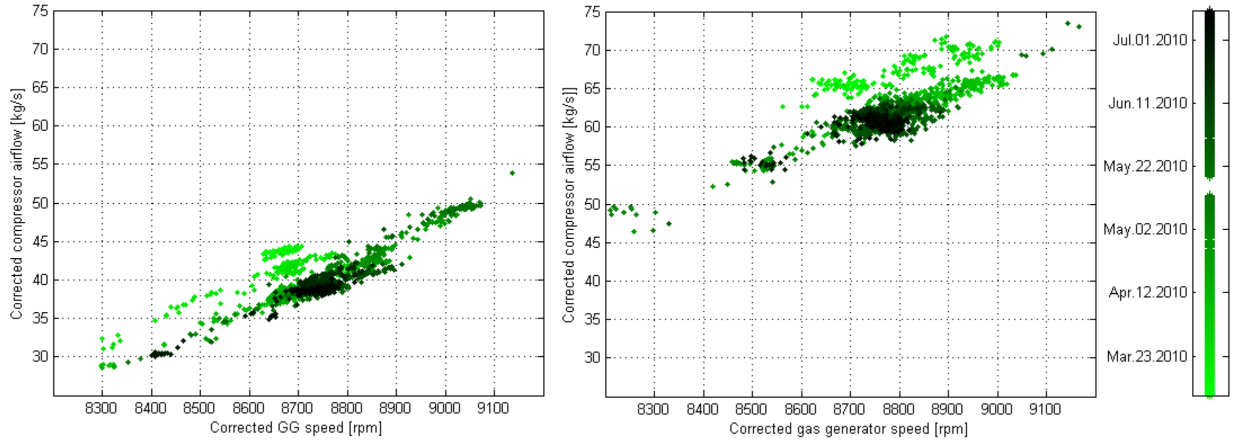


Figure 12-9: Corrected mass flow vs corrected GG speed- Machines 1 and 3 – period 5

Machine 1 versus Machine 2 (10.03-10.07.10) - Corrected compressor airflow versus corrected GG speed

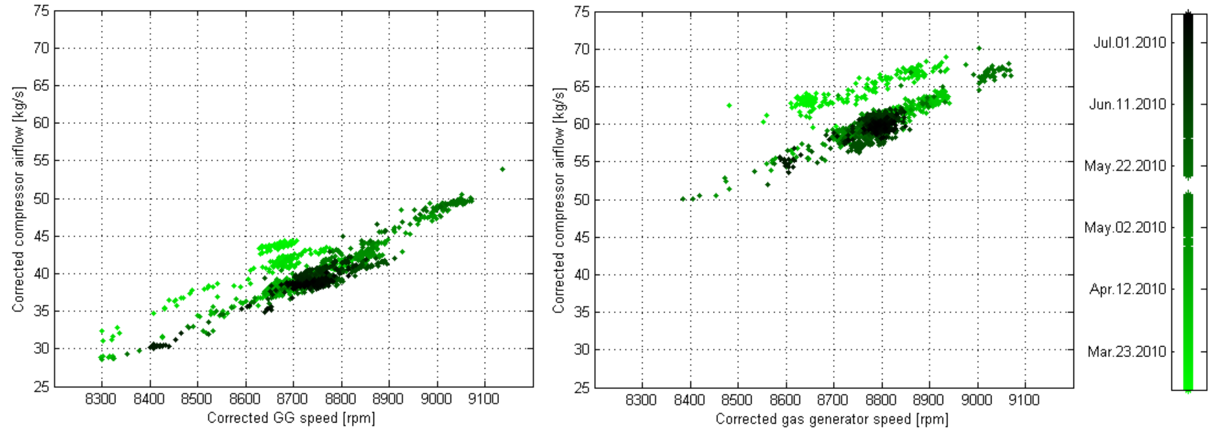


Figure 12-10: Corrected compressor air flow vs corrected GG speed - Machines 1 and 2

Machine 1 versus Machine 2 (15.07-12.11.10) - Corrected compressor airflow versus corrected gas generator speed

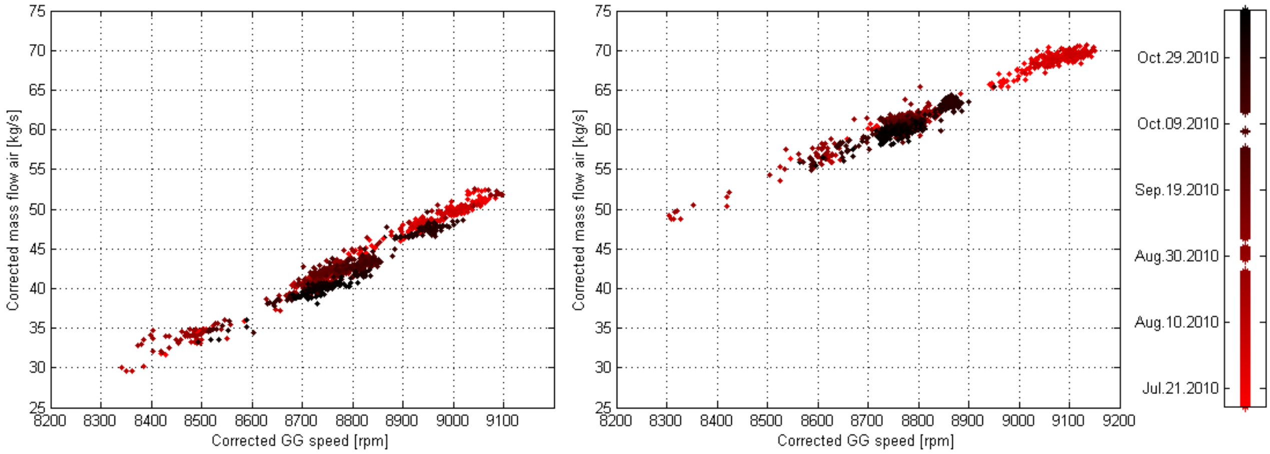


Figure 12-11: Corrected compressor airflow vs corrected GG speed - Machines 1 and 2

Machine 4 – Corrected compressor airflow versus corrected gas generator speed

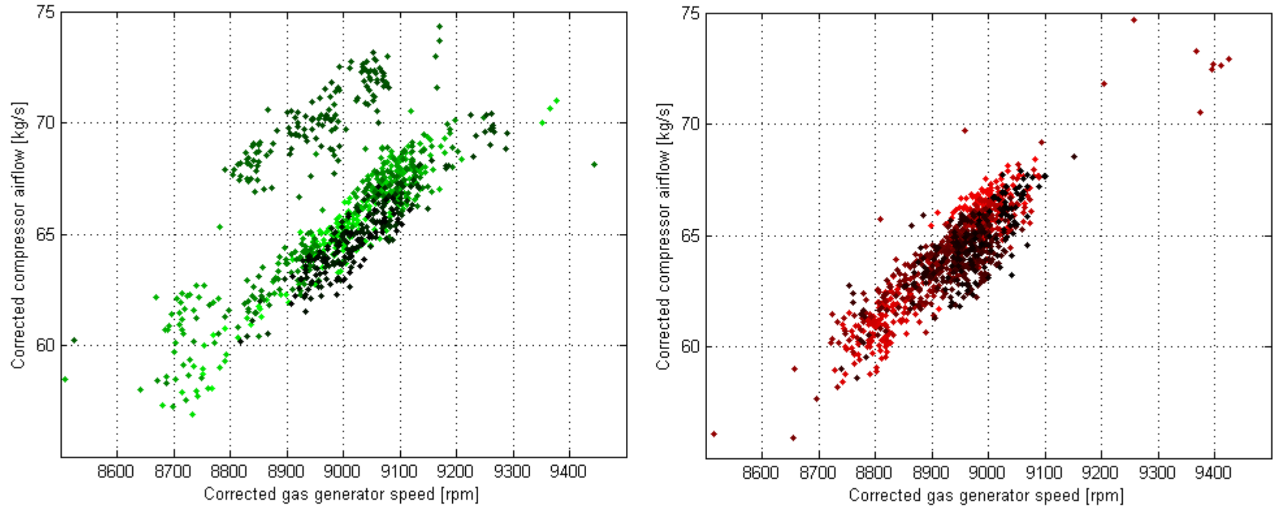


Figure 12-12: Corrected compressor airflow vs corrected GG speed - Machine 4

EXHAUST GAS TEMPERATURE

Machine 3 versus Machine 1 - Corrected exhaust gas temperature versus corrected gas generator speed

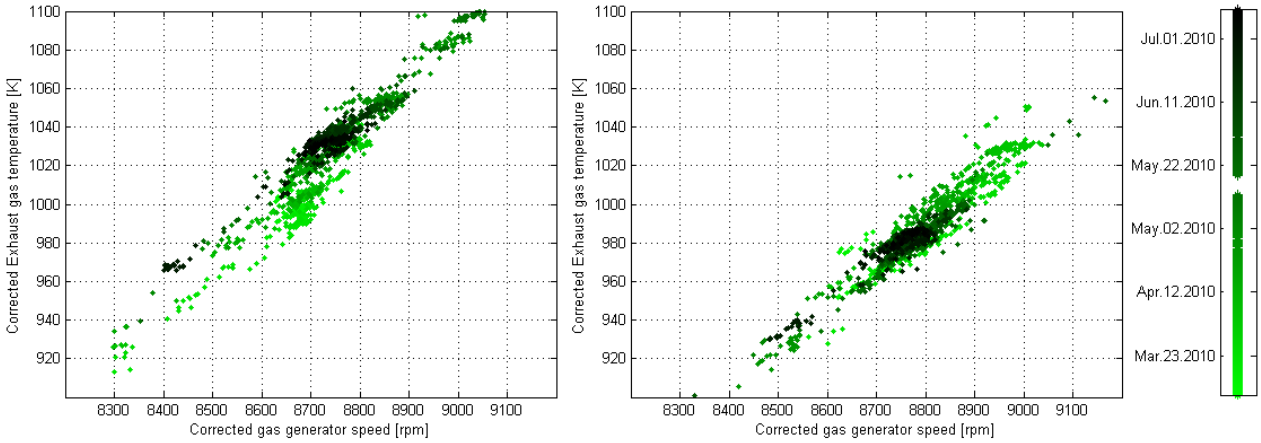


Figure 12-13: Corrected EGT vs corrected GG speed - Machines 1 and 3

Machine 1 versus Machine 2 (10.03-10.07.10) - Corrected exhaust gas temperature versus corrected gas generator speed

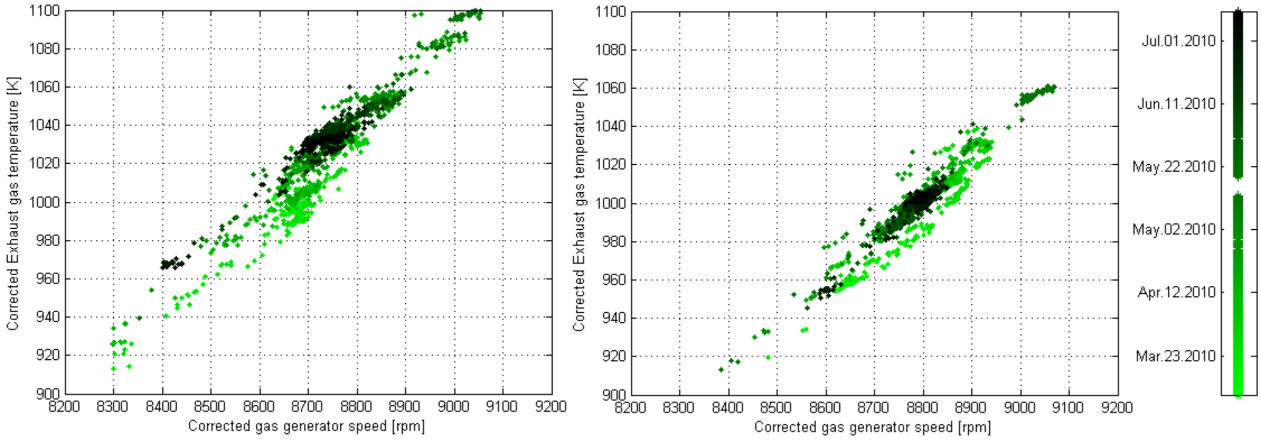


Figure 12-14: Corrected EGT vs corrected GG speed - Machines 1 and 2

Machine 1 versus Machine 2 (15.07-12.11.10) - Corrected exhaust gas temperature versus corrected gas generator speed

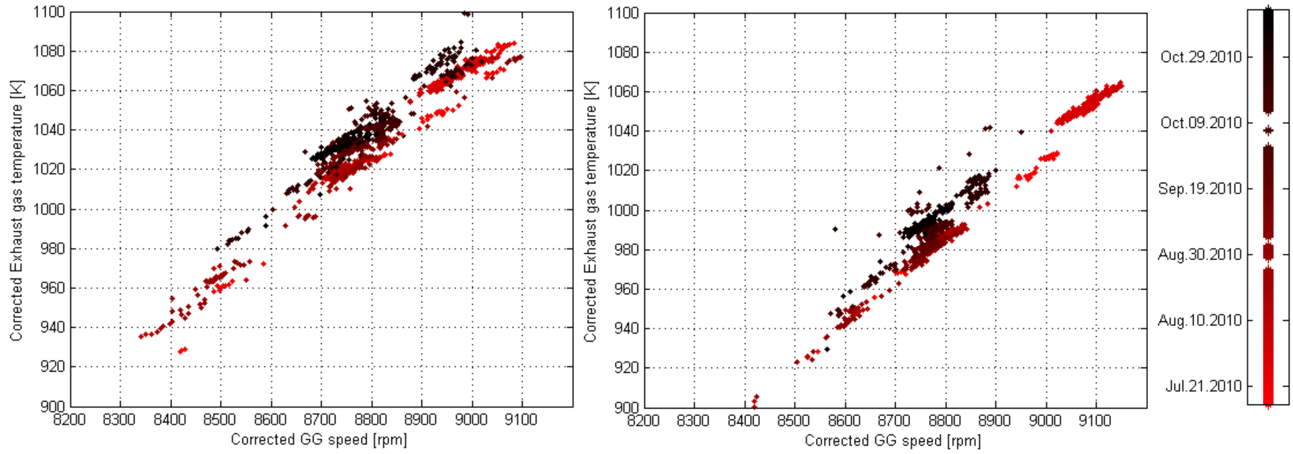


Figure 12-15: Corrected EGT vs Corrected GG speed - Machines 1 and 2

Machine 4 - Corrected exhaust gas temperature versus corrected gas generator speed

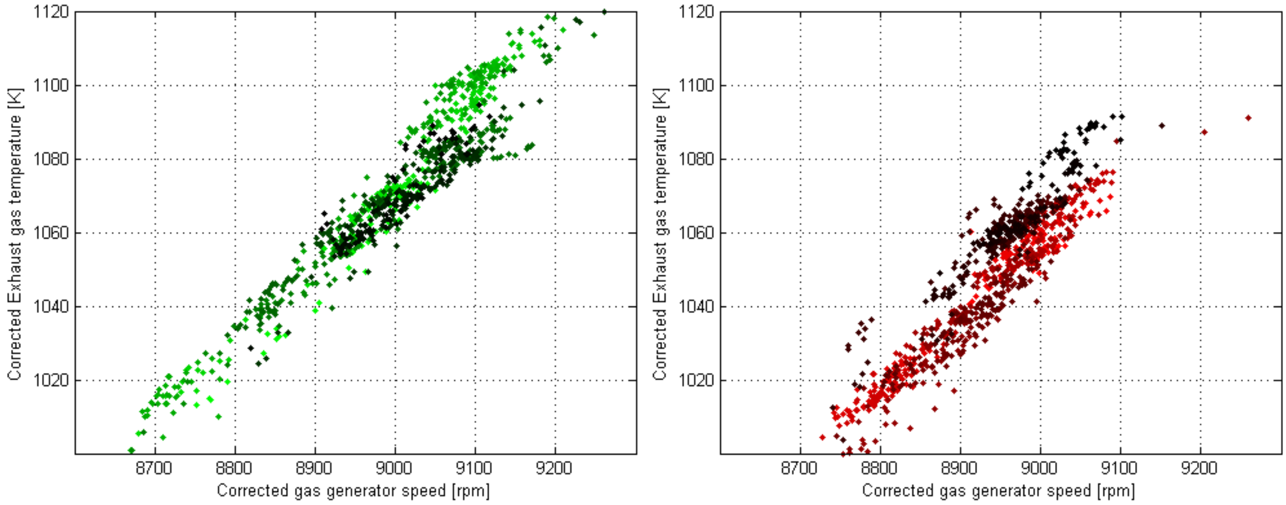


Figure 12-16: Corrected EGT versus corrected GG speed - Machine 4

COMPRESSOR DISCHARGE TEMPERATURE VS EXHAUST GAS TEMPERATURE

Machine 1 vs Machine 3 - Corrected compressor discharge pressure versus exhaust gas temperature

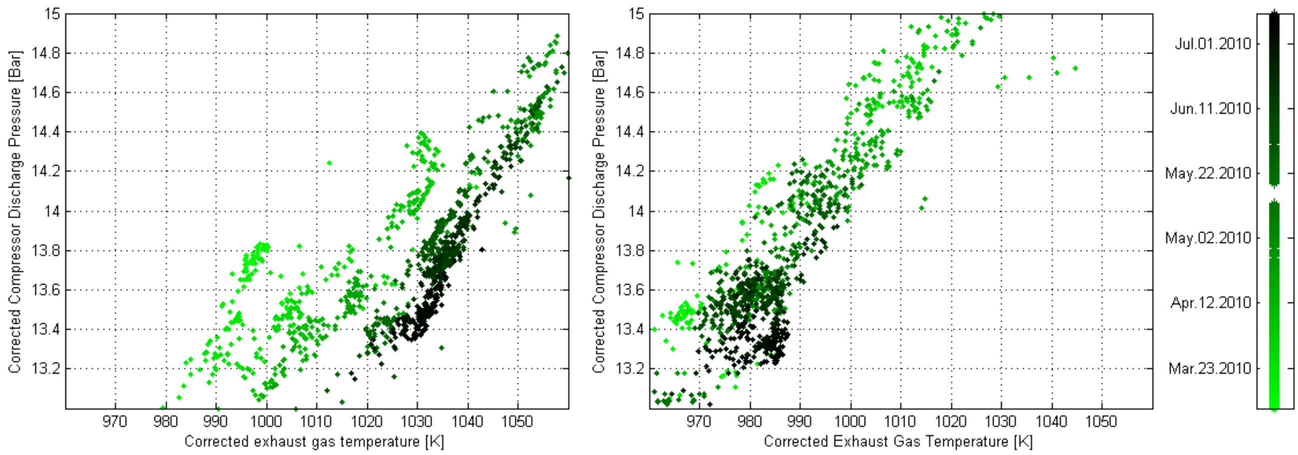


Figure 12-17: EGTc vs CDPc - Machines 1 and 3

Machine 1 versus Machine 2 (10.03-10.07.10) -Corrected compressor discharge pressure versus exhaust gas temperature

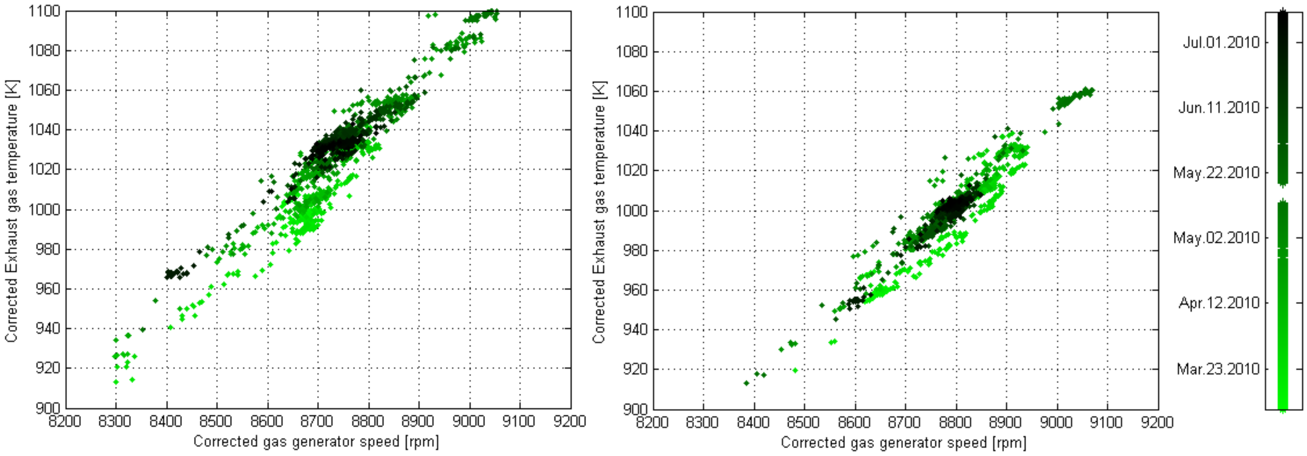


Figure 12-18: CDPc vs EGTc - Machines 1 and 2

Machine 1 versus Machine 2 (15.07 - 12.11.10) Corrected compressor discharge pressure versus exhaust gas temperature

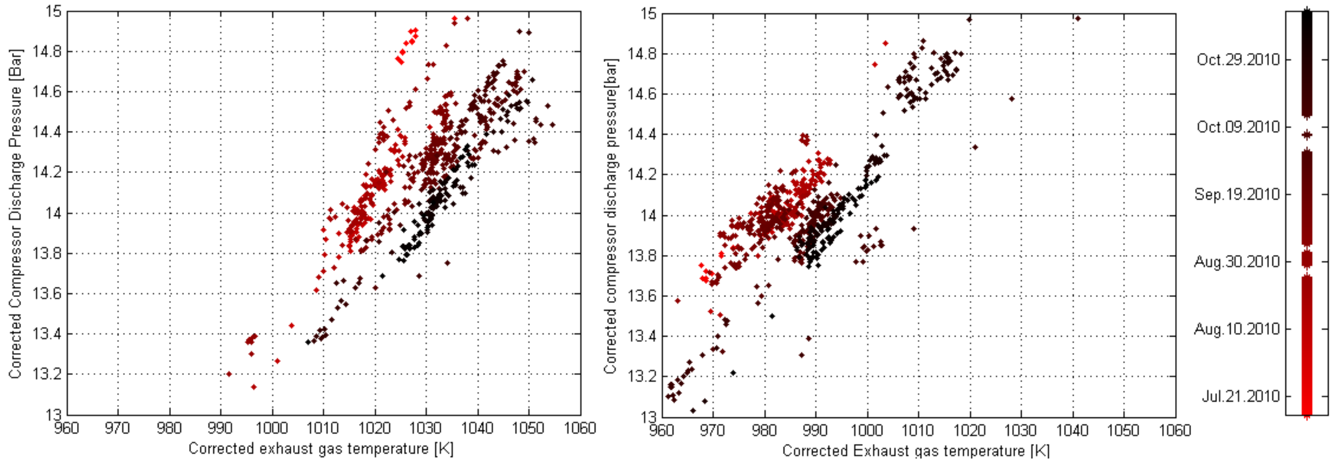


Figure 12-19: Corrected CDP vs corrected EGT for Machines 1 and 2

Machine 4 - Corrected compressor discharge pressure versus exhaust gas temperature

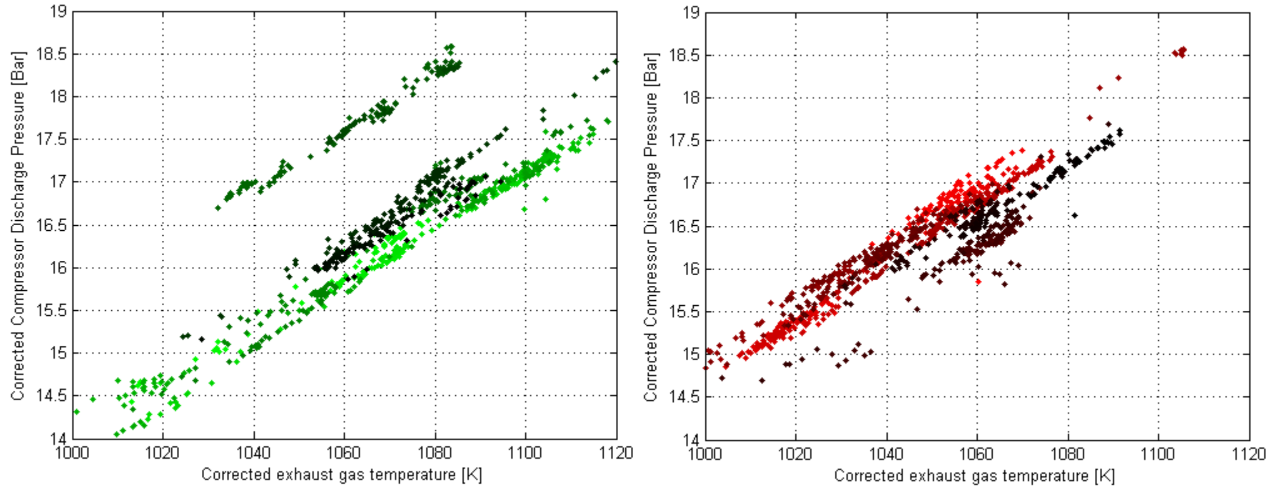


Figure 12-20: Corrected CDP vs corrected EGT - Machine 4

J: Deterioration rates for limited areas of EGTc and N1c

Machines 1 and 3: 10.03.10-10.07.10 (Blue graph: Machine 1, Pink graph: Machine 3)

Corrected CDP versus Date for a 50 rpm interval

Machine 1: N1c – 8710–8760 rpm, Machine 3: N1c – 8750-8800 rpm

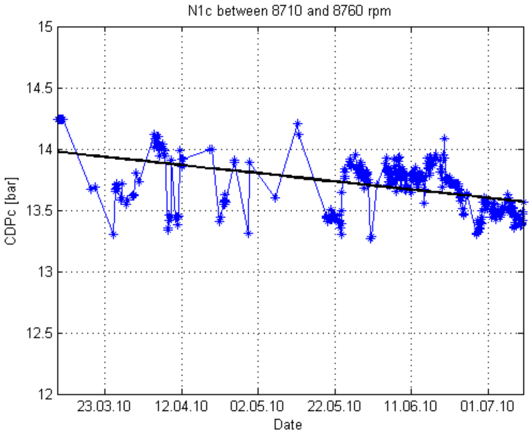


Figure 12-21: CDPc vs Date -N1c between 8710 and 8760 rpm - Machine 1

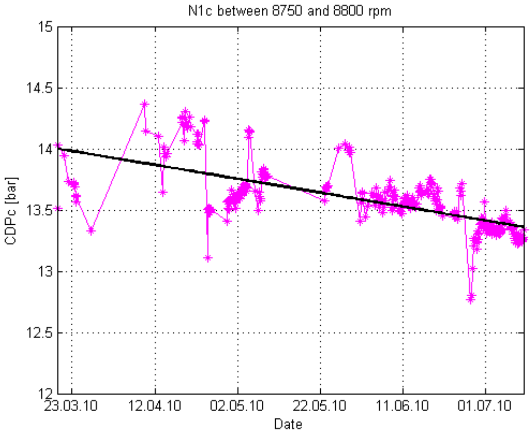


Figure 12-22: CDPc vs Date - N1c between 8750 and 8800 rpm - Machine 3

Corrected CDP versus Date for a 5 K interval

Machine 1: N1c – 1028–1033 K, Machine 3: N1c – 983,15-988,15K

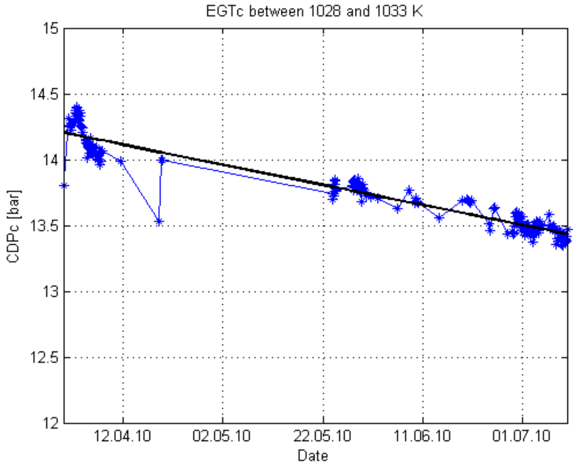


Figure 12-23: CDPc vs Date - EGTc between 1028 and 1033K - Machine 1

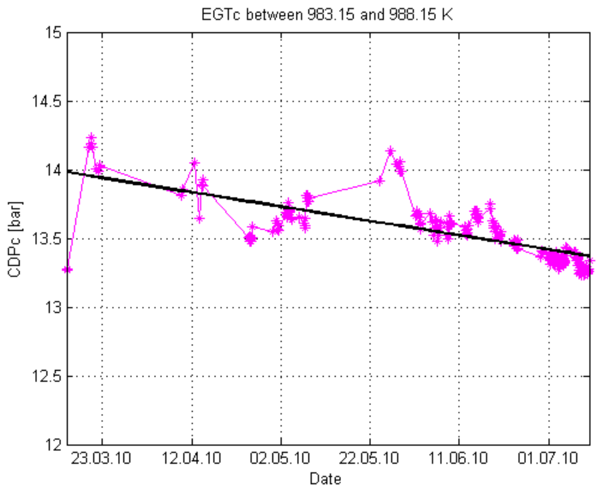


Figure 12-24: CDP vs Date - EGTc between 983.15 and 988.15K – Machine 3

Corrected compressor efficiency versus Date over a 5K interval
Machine 1: N1c – 1028–1033 K, Machine 3: N1c – 983,15-988,15K

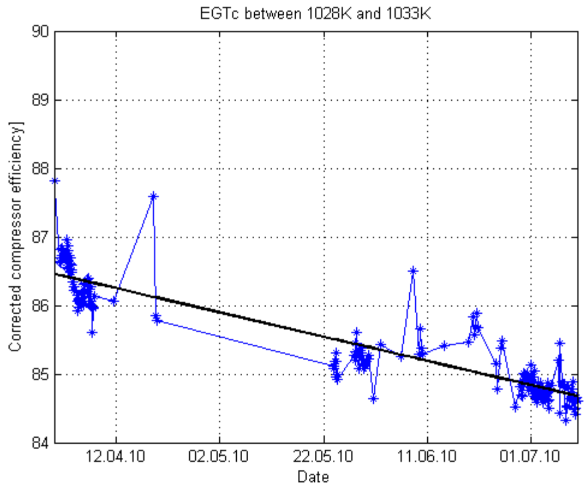


Figure 12-25: Corrected compressor efficiency vs Date- EGTc between 1028 and 1033K- Machine 1

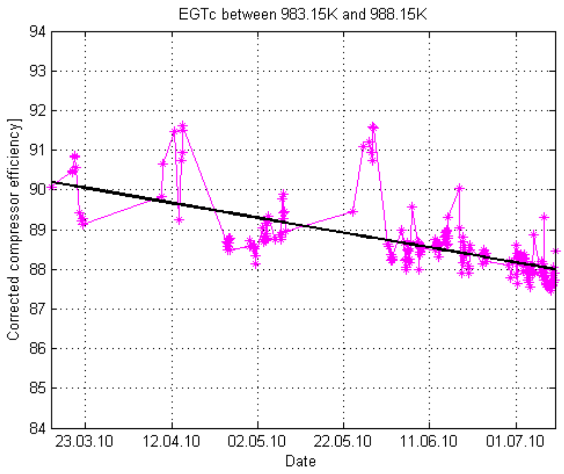


Figure 12-26: Corrected compressor efficiency vs Date - EGTc between 983.15 and 988.15K - Machine 3

Corrected compressor efficiency versus date over a 50 rpm interval
Machine 1: N1c – 8710–8760 rpm, Machine 3: N1c – 8750-8800 rpm

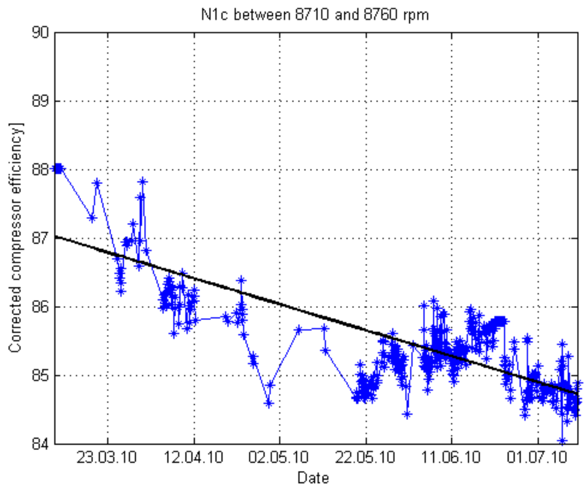


Figure 12-27: Corrected compressor efficiency vs Date - N1C between 8710 and 8760 rpm - Machine 1

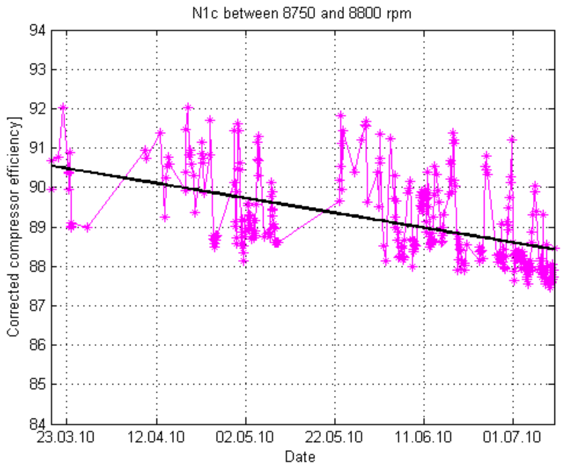


Figure 12-28: Corrected compressor efficiency vs Date -N1c between 8750 and 8800 rpm -Machine 3

Machine 1 versus Machine 3 – 10.03.10-10.07.10

Corrected compressor efficiency versus Date for a 200 rpm interval
Machine 1: N1- 8650-8850 rpm, Machine 3:N1- 8700 and 8900 rpm

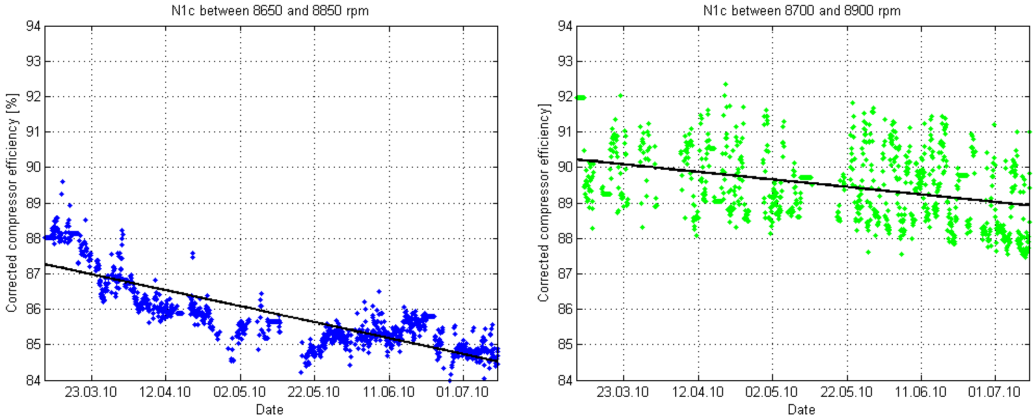


Figure 12-29: Corrected compressor efficiency vs Date- Machines 1(blue) and 3 (green)

Corrected mass flow versus Date for a 200 rpm interval
Machine 1: N1- 8650-8850 rpm, Machine 3:N1- 8700 and 8900 rpm

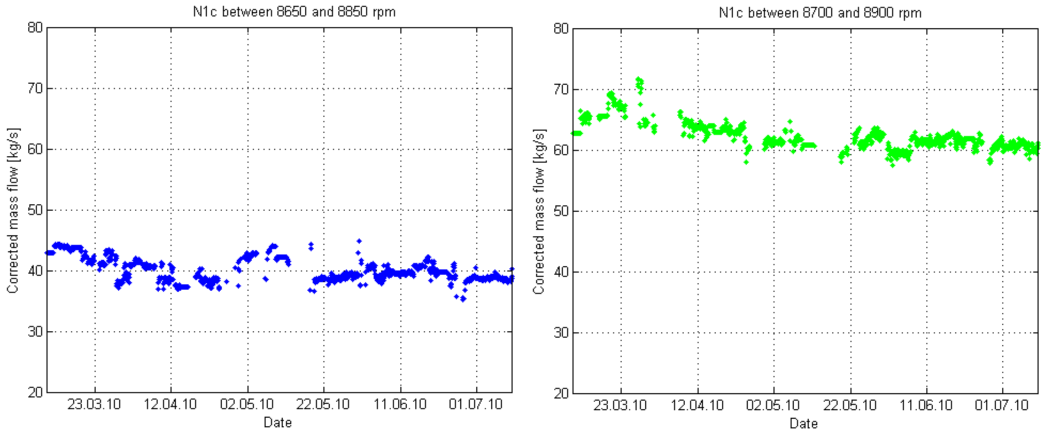


Figure 12-30: Corrected compressor airflow vs Date - Machine 1 (blue) and 3 (green)

Machine 1 versus Machine 2 - 10.03.10-10.07.10

Corrected compressor efficiency versus Date for a 200 rpm interval
Machine 1: N1- 8650-8850 rpm, Machine 3:N1- 8700 and 8900 rpm

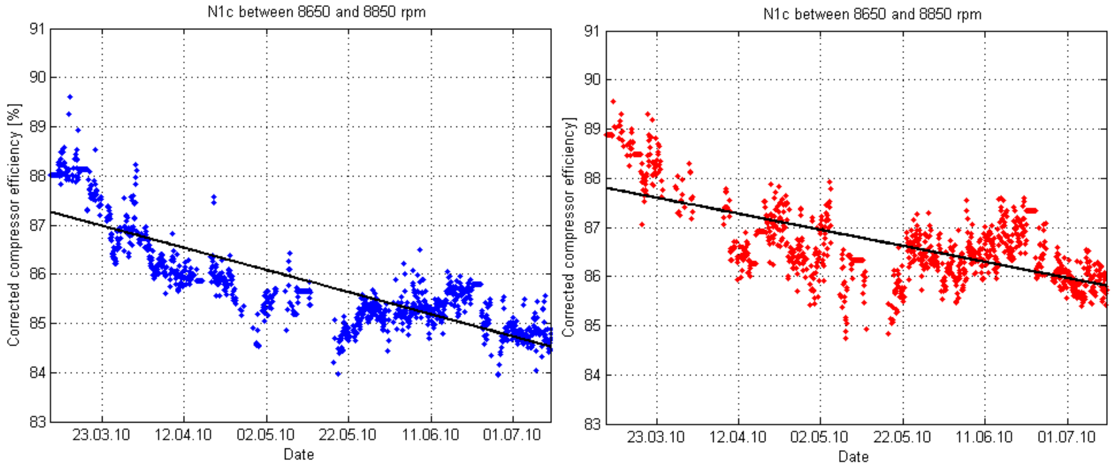


Figure 12-31: Corrected compressor efficiency vs Date - Machines 1 (blue) and 2 (red)

Corrected compressor air flow versus Date for a 200 rpm interval
Machine 1: N1- 8650-8850 rpm, Machine 3:N1- 8700 and 8900 rpm

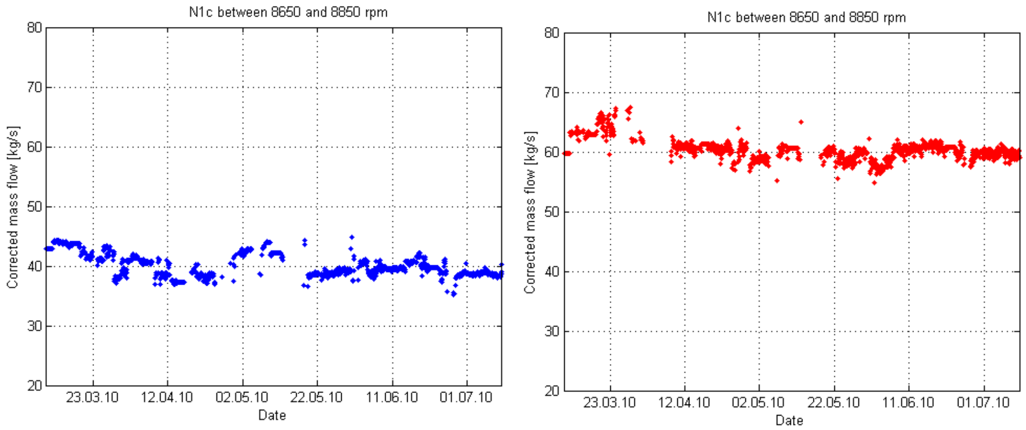


Figure 12-32: Corrected compressor airflow vs Date- Machines 1 (blue) and 2 (red)

Machine 1 versus Machine 2 - 15.07.10-12.11.10

Corrected compressor efficiency versus Date for a 200 rpm interval
Machine 1: N1- 8650-8850 rpm, Machine 3:N1- 8700 and 8900 rpm

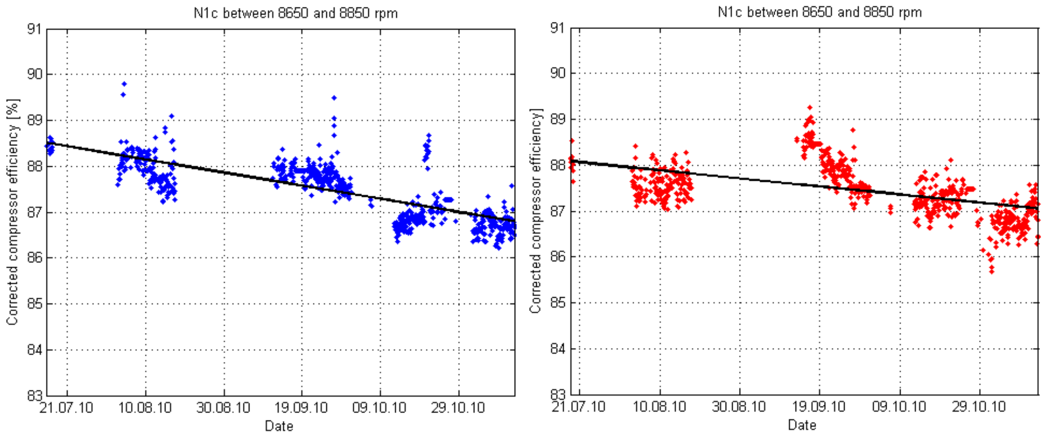


Figure 12-33: Corrected compressor efficiency vs Date - Machines 1 (blue) and 2 (red) – 15.07-12.11.10

Corrected mass flow versus Date for a 200 rpm interval
Machine 1: N1- 8650-8850 rpm, Machine 3:N1- 8700 and 8900 rpm

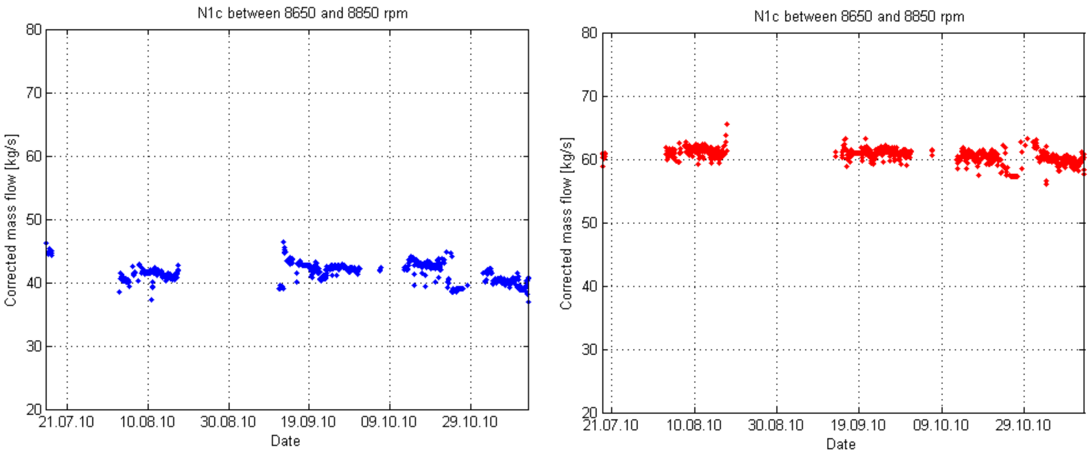


Figure 12-34: Corrected compressor airflow vs Date - Machines 1 (blue) and 2 (red) -15.07-12.11.10

K: Gas generator speed for Machine 3 – 10.03-10.07.2010

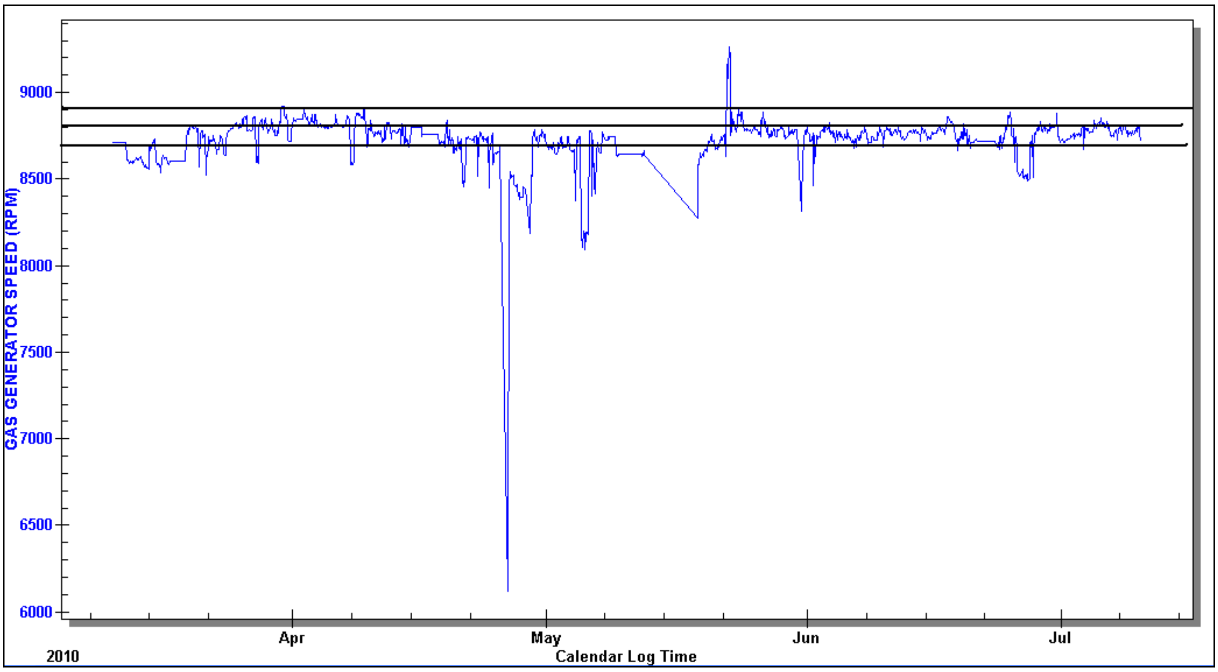


Figure 12-35: GG speed - Machine 3 – 10.03-10.07.2010