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Norwegian University of
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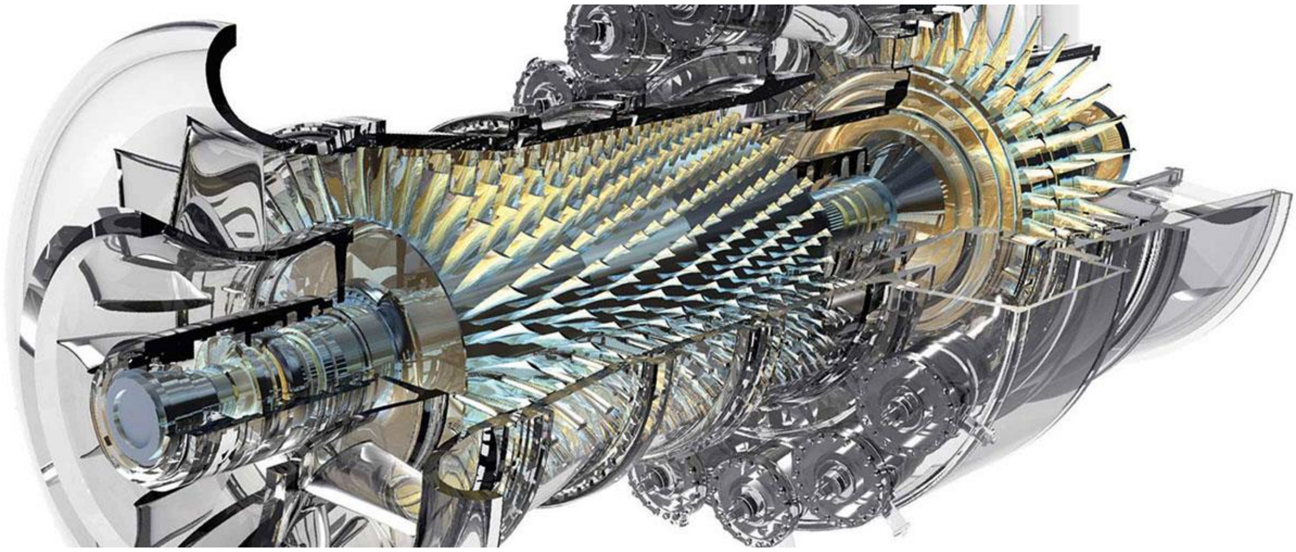
Edvard Aamodt

# Gas Turbine Deterioration and Recovery



Master of Science in Natural Gas Technology

Submission: February 2018

Supervisor: Lars Eirik Bakken, EPT

Co-supervisor: Stian-Mikael Madsen, Statoil

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

# Project description

**Norwegian University of Science and Technology**

**Faculty of Engineering**
**Department of Energy and Process Engineering**

EPT-P-2017-22

NTNU

**PROJECT WORK**

for

student Edvard Aamodt

Autumn 2017

***Gas Turbine Optimization***
*Optimalisering av gassturbiner*

**Background and objective**
Increased focus on energy efficiency and reduced emissions to air from gas turbine power production applies on both design and operation. Operating experience demonstrate a large incentive to analyse and understand the phenomenological effects related to deterioration and different cleaning processes to ensure optimum operation.

The main objective is to document "state-of-the-art" technology related to specific gas turbine water washing, including increased knowledge on the fundamental mechanisms related to deterioration and performance recovery of offshore gas turbines.A close cooperation with Statoil and PhD project "Optimum Gas Turbine Performance Offshore (online water wash and inlet air filtration)" is required.

**The following tasks are to be considered:**

Based on literature review and operating data the focus areas are:

1. Document different techniques to detect deterioration and recover performance of gas turbines. Focus on specific gas turbine type(s) is accepted.
2. Conduct performance analyses and validate condition parameters against site test/operational data. This includes polytropic flow path analysis utilizing Hysys.
3. Establish procedure for compressor efficiency variation and trend analyses to explore site and operating conditions impact on performance/performance deterioration.

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The project work comprises 15 ECTS credits.

The work shall be edited as a scientific report, including a table of contents, a summary in Norwegian, conclusion, an index of literature etc. When writing the report, the candidate must emphasise a clearly arranged and well-written text. To facilitate the reading of the report, it is important that references for corresponding text, tables and figures are clearly stated both places.
By the evaluation of the work the following will be greatly emphasised: The results should be thoroughly treated, presented in clearly arranged tables and/or graphics and discussed in detail.

The candidate is responsible for keeping contact with the subject teacher and teaching supervisors.

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

According to "Utfyllende regler til studieforskriften for teknologistudiet/sivilingeniørstudiet ved NTNU" § 20, the Department of Energy and Process Engineering reserves all rights to use the results and data for lectures, research and future publications.

The report shall be submitted to the department via Blackboard.

<u>Submission deadline:</u> 15 February 2018.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department for Energy and Process Engineering, August 22nd, 2017.

L E Bakken
Lars E Bakken
Supervisor

Co-Supervisor(s):
Stian Madsen, PhD

# Acknowledgements

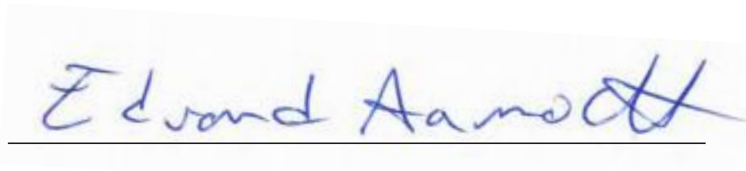
This thesis is the first part in a degree in Master of Science in Natural Gas Technology at Norwegian University of Science and Technology (NTNU), with specialisation in Energy, Process, and Flow Engineering.

This project work that has been reported in this thesis is conducted in the autumn 2017 and January 2018. The work has been performed in collaboration with NTNU and has taken place in Trondheim.

I would like to thank my supervisors Stian-Mikal Madsen and Lars Eirik Bakken for their guidance through this work and teachings in gas turbine operation. Some thanks also go out to several of the staff working in the department for their support during technical and practical challenges.

I would also like to thank my fellow students for their valuable support and help throughout the autumn and early spring. I would not have done it without you guys.

Trondheim 15. February. 2018



# Abstract

The harsh ambient conditions of offshore industry heavily influence the performance of offshore gas turbines. The main reason for gas turbine deterioration and the subsequent performance loss is found to be compressor fouling especially salt from the sea air. Ways to prevent deterioration and recover performance includes inlet filtration and offline and online water wash. To ascertain reliable monitored readings for compressor performance have proved to be difficult due to factors such as varying ambient conditions, inlet systems, inaccurate measuring instruments and gas turbine load. These factors are hampering the task of identifying the source of the performance loss.

Two gas turbines installed in the North Sea are documented and operational data from the two GE LM2500s compressors are collected. The data are validated through performance analysis of the operational data given from the monitoring equipment and software offshore. The polytropic efficiency calculated by the monitoring software are validated in simulation tools and are shown to be applicable for further investigation of deterioration trends.

Variation in ambient conditions such as temperature, pressure and humidity are found to contribute to the performance variation. Parameters like gas generator (GG) speed, discharge and exhaust temperatures corrected for compressor inlet temperature $T_2$ are found to reliably reflect performance trends and can be used for load-correction procedures in later studies.

Offline water wash is identified to be conducted every 6-months through plots generated in Excel and the performance recovery given in increasing polytropic efficiency is clearly indicated.

# Abstract in Norwegian

De ekstreme forholdene forbundet med offshore industri har stor innflytelse på gas turbiners ytelse. Den største grunnen til forverring av gas turbiner og deretter nedsatte ytelse er dokumentert å være beleggdannelse på kompressordelens komponenter, spesielt sjøsalt. Metoder for å forhindre forverring og gjenopprette ytelsen omfatter filtrering av innløpsluft og offline og online vannvask. Å fastslå sikre målinger for kompressorforverring viser seg å være vanskelig grunnet faktorer som varierende omgivelser, innløpssystemer, unøyaktige instrumenter og varierende last. Disse faktorene gjør også jobben med å identifisere kilden til ytelsestap vanskelig.

To gassturbiner installert i Nordsjøen er dokumentert og operasjonsdata fra to GE LM2500 kompressorer er innsamlet. Dataene valideres gjennom ytelsesanalyse for operasjonsdata gitt fra målingsinstrumenter og programmer offshore. Den polytropiske virkningsgraden beregnet av målingsprogrammet er validert i simuleringsverktøy og viser seg å være brukbare for videre undersøkelse av forverringstrender.

Data for omgivende varierende forhold som temperatur, trykk og luftfuktighet tyder på at kan disse parameterne kan gi ytelsesvariasjon. Parametere som gas generatorfart (GGspeed), utløps- og eksostemperatur korrigert for innløpstemperatur $T_2$ er funnet til å være sikre indikatorer som reflekterer ytelsestrendene for kompressorene. Disse kan derfor brukes i en metode for lastkorreksjon i en videre undersøkelse.

Gjennom grafer generert i Excel er offline vannvask identifisert og gjenopprettelsen av ytelse er dokumentert ved polytropisk effektivitet.

# List of figures

# Nomenclature

## Symbols

| | | |
|---|---|---|
| X/Y | Compressibility Function | - |
| η | Efficiency | % |
| T | Temperature | °C |
| P | Pressure | bara |
| k/κ | Heat Capacity Ratio | - |
| $C_p$ | Heat Capacity at Constant Pressure | $J/K$ |
| $C_v$ | Heat Capacity at Constant Volume | $J/K$ |
| θ | Correction Factor | - |
| δ | Correction Factor | - |
| m | Polytropic Temperature Exponent | - |
| n | polytropic Volume Exponent | - |
| R | Gas Constant | $J/molK$ |
| Z | Compressibility Factor | - |
| a | Molecule Factor | - |
| b | Molecule Factor | - |
| ω | Acentric Factor | - |
| W | Work | $J/s$ |
| N | Rotational speed | rpm |

## Abbreviations

| | |
|---|---|
| $CO_2$ | Carbon Dioxide |
| $NO_x$ | Various Nitrous Oxides |
| GE | General Electric |
| ISO | International Standard Organisation |
| GG | Gas Generator |
| HPT | High Pressure Turbine |
| PT | Power Turbine |
| VIGV | Variable Inlet Guide Vane |
| EGT | Exhaust Gas Temperature |
| DLE | Dry Low Emission |
| CFF | Compressor Front Frame |
| CRF | Compressor Rear Frame |
| TMF | Turbine Mid Frame |
| TRF | Turbine Rear Frame |
| EOS | Equation of State |
| SRK | Soave-Redlich-Kwong |
| PR | Peng-Robinson |
| RH | Relative Humidity |

## Subscripts

| | |
|---|---|
| c | Critical |
| c | Corrected |
| i | Inlet |
| o | Outlet |
| p | Polytropic |
| 2 | Compressor Inlet |
| 3 | Compressor Outlet |
| 5.4 | Low Pressure Turbine Inlet |

# Contents

# 1. Introduction

This chapter is presented to give the reader an introduction to the project work the author has been conducting. It briefly describes why this topic has been chosen, the scope of the work and how this report is structured.

## 1.1 Background

Most of the power consumption processes and equipment drivers needed on the offshore processing plants today are run by gas turbines. A critical problem for the reliable and efficient supply of energy is gas turbine deterioration. Deterioration can happen in different parts of the gas turbine and depend on the ambient conditions surrounding the machine. For gas turbines installed on platforms in the North Sea, compressor fouling has the greatest impact in the performance of the compressor and causes the most offline stops. The compressor acts like an effective air filter, collecting and depositing solids such as salts from the ambient air. This effect is also strengthened by the subsequent temperature rise in the compressor as the ambient air is compressed [3].

Statoil is a company operating in over 30 countries and are currently running over 100 gas turbines world-wide [1]. There is a significant economic gain in trying to minimize the compressor fouling as it leads to higher fuel consumption and subsequently higher $CO_2$ and $NO_X$ emissions. Also, as the Statoil gas turbines are responsible for nearly 85 % of the company's combined fossil fuel emissions, the company is set on reducing the fuel consumption where this is possible [2].

As can be seen in the plot given below (figure 1) the efficiency of a LM2500 compressor decrease about 1 % along the 6-month operating period due to fouling. The sudden recovery of efficiency, at the start and end of the operating period, is due to the offline water wash and is, alongside inlet filtration, contributing to a significant reduction in performance losses.

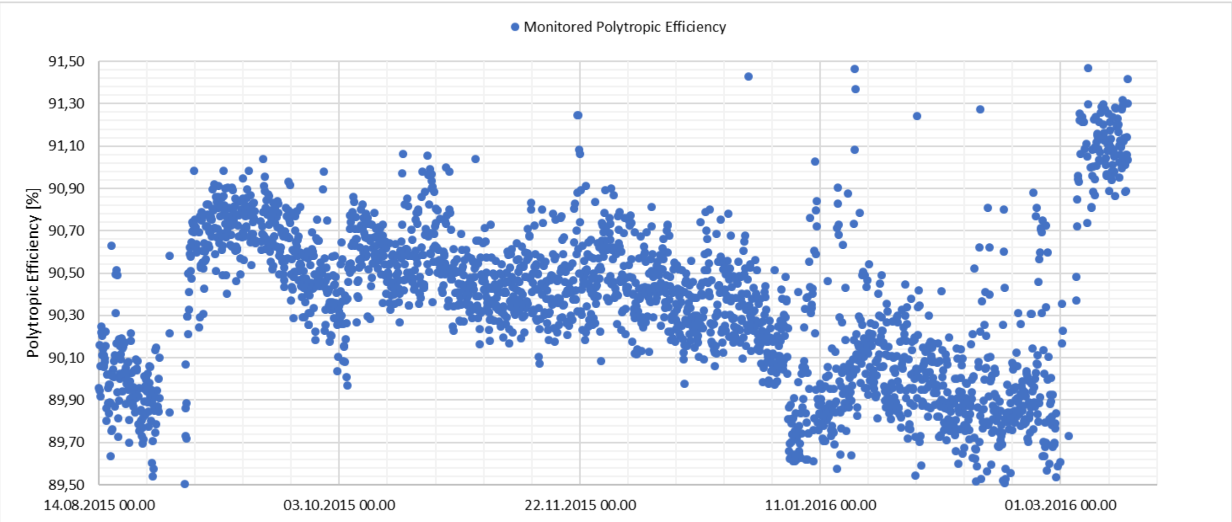



*Figure 1: Monitored 6-month efficiency trend*

Since the introduction of online water wash the aforementioned operating intervals have been increased due to decreasing deterioration rates. The efficiency line is displaying a trend reoccurring after each offline wash and is measured by the current instrumentation and monitoring software.

Therefore, a vast interest in research surrounding online water wash has been shown from the industry and especially Statoil. This project report is meant to serve as ground work for future research by the author on the matter. It will document different techniques for detecting deterioration and performance recovery by offline and online water wash in two LM 2500 gas turbines. Simulation tools and monitored conditions will be used for performance evaluation of the compressor and validation of the operational data.

## 1.2 Scope of work

It is agreed that this project work is going to focus on two gas turbines with given operational data taken from an offshore platform. The emphasis would be on the compressors as they are the most exposed part of the offshore gas turbines. The two engines, called A and B, where chosen as to point out the difference between a gas turbine running a generator (A) with steady load as opposed to gas turbine running a mechanical drive (B) for process compressors experiencing varying load.

The gas turbines are part of an ongoing study on air filtration and online water wash where the latter is to be documented alongside the impact on offline water wash intervals. In addition to varying load conditions, the two engines are equipped with two different sets of instrumentation, engine A with new and B with old. This allows for further comparison and evaluation of problems related to instrumentation, process, and performance. Inlet systems such as anti-icing and inlet filtration will be briefly covered and documented.

A performance analysis is carried out. The parameters are corrected to ISO standards found in literature. The monitored data will be validated in Aspen Hysys using the excel add-inn Aspen Simulation Workbook which allows for handling of large amounts of data. The performance of the engines are agreed to be highlighted through polytropic efficiency alone.

## 1.3 Thesis structure

Chapter 1 consists of an introduction to the project thesis, the scope of work and how the thesis is structured.

Chapter 2 gives a brief introduction to gas turbine theory and operation.

Chapter 3 gives a brief introduction to gas turbine deterioration, the resulting performance loss and how it affects the operation.

Chapter 4 explains how deterioration is currently prevented and how performance is recovered.

Chapter 5 covers the performance analysis and the validation of operational data.

Chapter 6 includes a summary of the work and the conclusion

Chapter 7 are suggesting work for further investigation and ideas on what the author can work further on.

# 2. Gas turbines

Gas turbines are used on offshore plants both for power generation and drivers for mechanical equipment such as compressors. They follow the thermodynamic principles of the Brayton cycle. This cycle involves a compression of the gas, the subsequent heating of the gas by burning of fuel, followed by an expansion of the hot compressed gas. The first three steps are referred to as a gas generator (GG) comprised of a compressor, a combustion chamber, and a high-pressure turbine (HPT). Utilizing the compressed, heated gas for expansion the HPT drives the compressor and a power turbine (PT) is using the remaining energy in the gas for shaft power output. More on thermodynamic principles in gas turbines, and how the machines work is extensively covered in Gas Turbine Theory in Saravanamuttoo [4].

Following is a brief description of the evaluated gas turbines.

## 2.1 Statoil gas turbines

On one of the Statoil operated platforms in the North Sea utilizing the LM2500 gas turbine from General Electric (GE). These gas turbines are delivering power to the platform as well as the nearby platforms. In addition, eight of the gas turbines are used for operating the compressor drives on the field. This project has focus on one of the gas turbine of the power generating GE LM2500 referred to as gas turbine/compressor A and in addition on a compressor drive gas turbine referred to as gas turbine/compressor B.

### 2.1.1 General Electrics LM2500

The GE LM2500 is one of the top selling gas turbines in its class, dating back to the first runs on the 1960s. This is due to its high reliability, relatively high simple cycle efficiency and the fact that it can deliver up to 24 MW of power depending on its configuration. It is a two shaft aeroderivative gas turbine consisting of gas generator and a power turbine, see figure 2 and 3.

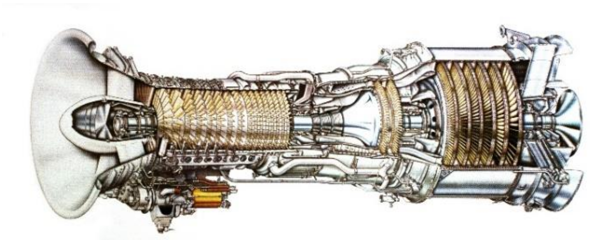


*Figure 2: LM2500 cutaway*

#### Compressor section

The axial compressor section consists of 16 stages. A stage is made up of a row of rotor blades followed by a row of stator blades. The rotor blades are accelerating the working fluid which transfers its kinetic energy on to the rotor blades which converts it into static pressure [4]. The first six stages of the LM2500 compressor are variable inlet guide vanes (VIGV) to vary the entering flow angle and through the stages a compression ratio of 20:1 is generated [10].

### Combustor section

After the working fluid is compressed it is sent to the annular combustion chamber. There are 30 nozzles for gas fuel injection which is mixed and thereafter burned [10]. The exhaust gas comprising of combustion-deposits and -gases and excessive air is sent to the high-pressure turbine (HPT) at a temperature of about 1200 °C.

### High pressure turbine

The high-pressure turbine (HPT) consists of 2 stages and delivers shaft power to the compressor and gearbox. The high energy combustion gas and excess air flows through the stator and rotor blades of the turbine producing rotational energy which is transferred to the compressor as they both are attached to the same shaft. Higher turbine inlet temperatures (TIT) and pressures result in higher turbine performance.

### Power turbine

This section is a six-stage turbine which utilizes the left-over pressure and temperature from the HPT to create shaft power output. It is not connected to the other components but rather rotates on its own shaft at a different speed. As mentioned before this shaft power is used for driving a generator or a process compressor drive.

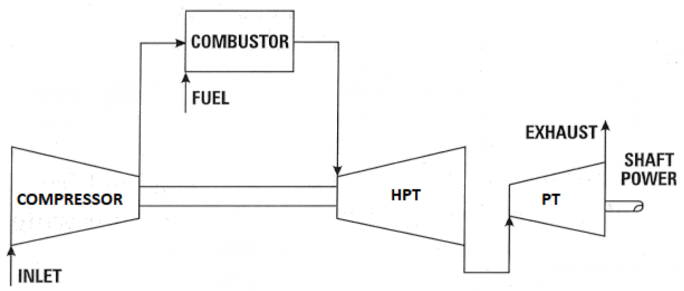



*Figure 3: Schematic of a gas turbine with Gas Generator and Power Turbine*

## 2.2 Compressor characteristics

This section gives a very brief introduction to compressor characteristics which will become important at later stages in this report.

### 2.2.1 Polytropic analysis

When calculating for polytropic flow in a compressor it is important to account for real gas relations. In this study the real gas calculations of Schultz [18] are chosen for the calculation procedure. It differs from ideal-gas relations by supplementing compressibility factor Z with compressibility functions X and Y. It also introduces a polytropic head factor f to adjust the results for deviations from ideal-gas behaviour. Schults' method proposed a simple iterative way of accounting for real-gas relations in the calculations, which is given below:

I. Note inlet η, $T_i$ and $P_i$. Calculate $V_i$ with $k = \frac{c_p}{c_v}$.

II. Estimate outlet temperature $T_o$:

$$T_o = T_i \left(\frac{P_o}{P_i}\right)^{\frac{k-1}{k}} \tag{1}$$

III. Estimate average pressure $\bar{P}$:

$$\bar{P} = P_i \sqrt{\frac{P_o}{P_i}} \; or \; \bar{P} = \frac{P_o + P_i}{2} \tag{2}$$

IV. Estimate average temperature $\bar{T}$:

$$\bar{T} = \frac{P_o + P_i}{2} \tag{3}$$

V. Estimate average heat capacity ratio $\bar{k}$:

$$\bar{k} = \frac{k_i + 2k_{\bar{T},\bar{P}} + k_o}{4} \tag{4}$$

VI. Estimate compressibility function $\bar{X}$:

$$\bar{X} = \frac{T}{V}\left(\frac{\delta V}{\delta T}\right)_P - 1 \; at \; \bar{P}, \bar{T} \tag{5}$$

VII. Estimate compressibility function $\bar{Y}$:

$$\bar{Y} = \frac{T}{V}\left(\frac{\delta V}{\delta P}\right)_T - 1 \; at \; \bar{P}, \bar{T} \tag{6}$$

VIII. Estimate polytropic temperature exponent $\bar{m}$:

$$\bar{m} = \frac{\left(\frac{\bar{k}-1}{\bar{k}}\right)\left(\frac{1}{\eta} + \bar{X}\right)\bar{Y}}{(1+\bar{X})^2} \tag{7}$$

IX. Estimate polytropic volume exponent $\bar{n}$:

$$\bar{n} = \frac{(1 + \bar{X})}{Y\left[\frac{1}{\bar{k}}\left(\frac{1}{\eta} + \bar{X}\right) - \left(\frac{1}{\eta} - 1\right)\right]} \tag{8}$$

X. Estimate new outlet temperature $T_o$:

$$T_o = T_i\left(\frac{P_o}{P_i}\right)^{\bar{m}} \tag{9}$$

XI. Estimate outlet volume $V_o$:

$$V_o = V_i\left(\frac{P_o}{P_i}\right)^{-\frac{1}{\bar{n}}} \tag{10}$$

XII. Verify n from EOS:

$$\bar{n} = \frac{\ln\left(\frac{P_o}{P_i}\right)}{\ln\left(\frac{V_i}{V_o}\right)} \tag{11}$$

XIII. Compare n-values and adjust $T_o$ estimates or subdivide steps and return to step IV.

XIV. Estimate work W:

$$W \approx \frac{1}{\eta}\left(\frac{\bar{n}}{\bar{n} - 1}\right)\frac{Z_i R T_i}{MW}\left[\left(\frac{P_o}{P_i}\right)^{\frac{\bar{n}-1}{\bar{n}}} - 1\right] \approx \frac{f}{\eta}\left(\frac{\bar{n}}{\bar{n} - 1}\right)\frac{Z_i R T_i}{MW}(Z_o T_o - Z_i T_i) \tag{12}$$

### 2.2.2 Map

The compressor map is a diagram created to display the characteristics of the compressor. It is unique to every gas turbine model and the machine must be driven by an external drive and go through extensive testing to have its map created. The manufacturer seldom gives such a plot to the operator.

This map is a typical characteristic for the LM2500 gas turbine and the operating line is not specified to the evaluated compressors in this study. The compressor pressure ratio is plotted against the corrected mass flow, but it also contains constant speed, efficiency, and surge lines. A compressor following a constant speed line will, with reduced or increased mass flow, deviate from the best efficiency point which follows the operating line [4].

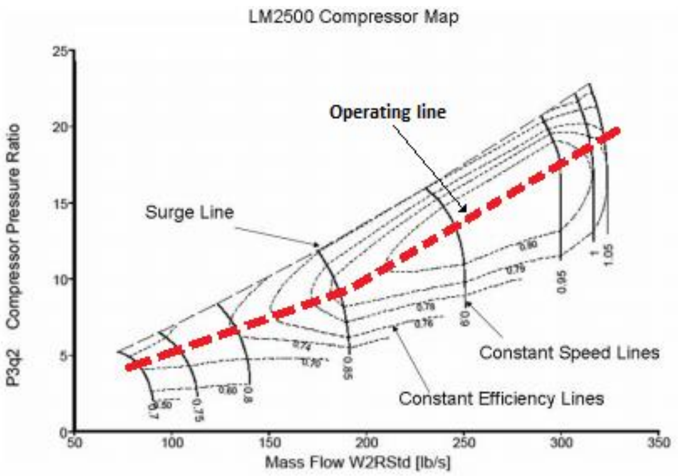



*Figure 4: An GE LM2500 compressor map, derived from Dahl [12]*

### 2.2.3 Surge/stall

The operating point can, if not monitored correctly, move beyond the surge line which indicates the development of surge in the compressor. Compressor surge is a phenomenon occurring when there is a sudden drop in flow and subsequently delivery pressure to the compressor and the downstream pressure does not follow the drop. This forces the air to reverse its direction and the pressure ratio drops rapidly making the flow pulsate back and forth.

Airfoil or rotating stall occurs when operation close to the surge line produces unstable flow. This induces boundary layer separation resulting in turbulence over the airfoils, which are hard to notice.

Both phenomena result in instability and poor performance and should be closely monitored and avoided. Vibrations caused by these phenomena can also have non-recoverable results including fatigue failures throughout the gas turbine [4].

### 2.2.4 Control systems

To avoid unwanted or critical operational situations, a control system controls the gas turbine. The system must ensure that the limits of gas generator (GG) speed ($N_1$), exhaust gas temperature (EGT) [$T_{5.4}$] and the above-mentioned compressor surge are not exceeded. Fuel consumption, and in this study, VIGV's, are examples of parameters that can be used to ensure steady operation of the gas turbine at any ambient condition or operating mode.

# 3. Gas turbine deterioration

In this chapter, general gas turbine deterioration will be covered. It will first touch in on parameters contributing to performance variation and what types of deterioration there is with special emphasis on recoverable deterioration as it is most relevant for this study. Documentation from the two analysed gas turbines will be provided.

## 3.1 Offshore operating environment

The environment surrounding an offshore platform in the North Sea can be harsh and the impact on the platform equipment thereafter. It is mainly the gas turbine compressor that is affected by the ambient conditions and contaminants from the processing equipment. This includes exhaust gas and fumes from other combustion processes, cooling towers aerosol drifts and gas leaks from transport pipes [6]. After analysing the compressor fouling of two offshore gas turbines in the North Sea, the dominating contaminant was found to be salt [5].

### 3.1.1 Humidity

Weather conditions resulting in fog formation around the platforms have proven to affect the performance on gas turbines negatively. Studies regarding this phenomenon shows that humidity in the air resulted in an increased pressure drop over the inlet filtration system and a later decrease in compressor efficiency [7]. A theory regarding this also suggests that there is an unloading of salts and other contaminants into the air stream making them deposit in the compressor flow path. This has been supported by the fact that compressor cleaning recovered the performance loss. Humidity data are one of the unavailable parameters within this study.

### 3.1.2 Ambient conditions

Ambient conditions such as temperature and pressure are important parameters to address and they vary a great deal. In Appendix B (figure 29 and 30) the impact of time of year and its effect on the inlet conditions can be seen. It is also interesting to see how accuracy and parts of obvious mismeasurements can increase the unreliability of the results. This is easy to see around 31.12.2016 where a far too low ambient pressure is affecting the simulated performance for compressor A as depicted in figure 23 in chapter 5.3. In theory, lowering the ambient pressure will decrease the gas turbine performance as the control systems will compensate with higher fuel rate. An increase in temperature will result in decreased air density and therefore lower the performance.

## 3.2 Non-recoverable deterioration

Non-recoverable deterioration is due to mechanical wear in the gas turbine components. These problems are due to erosion and corrosion of the components, especially the compressor and turbine blading. By impingement of particles on the flow surfaces removal of material occurs, this is called erosion. Water droplets that are injected into the gas turbine are also eroding the blades, which are so carefully designed that even the slightest change in blade profile can affect the machine performance. Corrosion is a more complicated phenomenon where several mechanisms can contribute to reduction in p erformance. Usually oxygen in the air reacts to the metal surface, but other chemicals can also create corrosion. Temperature alongside contaminants in the inlet air, fuel or water contributes as well [3]. Non-recoverable deterioration will not be covered further in this report.

## 3.3 Recoverable deterioration

Recoverable deterioration is mainly due to compressor fouling [4]. As will be described in chapter 4.2, cleaning of the compressor is the primary solution to the performance loss. This report will only cover compressor fouling as it is a common problem for the gas turbines on the platforms in the North Sea.

### 3.3.1 Compressor fouling

Fouling of the compressor section is as mentioned the main catalyst for gas turbine deterioration on the platform in the North Sea. It is due to the gradual build-up of deposits on the compressor blades and its edges [4]. This layer of contaminants increases the roughness and change the shape of the airfoil and the annulus to some extent. It is especially the pressure side of the airfoil that experiences the largest increase in roughness [8]. Salt is especially prominent in the offshore compressor fouling and leads to the formation of chloride deposits in microscopic surface cracks. It also leads to the formation of subsurface corrosion, as can be observed as small pits or holes on the surface of the blades [3].

In 2009 a study [5] done on two offshore gas turbines showed that considerable contamination could be found on the compressor inlet or bellmouth, on the inlet guide vanes and the first rotor stage.

## 3.4 Operating with deteriorated compressor

As the compressor becomes fouled the operating line is found to be shifted and this results in a lower flow rate and compression ratio in the compressor map described in chapter 2.2.1. This means that the efficiency will be reduced as the operating point moves closer to the surge line [8].

### 3.4.1 Performance loss

To compensate for the reduced air mass flow, the fuel flow and GG speed N1 must be increased. Increasing the fuel flow will give a higher exhaust gas temperature (EGT) and can damage the turbine blades if not restricted. As the reduced air flow continues and the control system prohibits higher EGT, the output power of the gas turbine, and subsequently the performance, will decrease. Increase in GG speed $N_1$ and EGT is clearly seen in the plots given in chapter 5.2.1 for corrected parameters when ambient conditions are accounted for.

### 3.4.2 Friction losses and reduced flow area

The surface roughness is found [9] to have a great significance to the inlet volume flow reduction, which will move the operating point away from design point and optimal efficiency. This is in addition to frictional losses and an increase in boundary layer thickness.

### 3.4.3 Other losses

Increased fuel consumption results in an increase in air pollution and fuel costs. The contaminants from the combustion can deposit on the turbine blading and cause increased friction and clogging.

# 4 Deterioration prevention and recovery

This chapter will cover the methods of deterioration prevention and how they affect the performance of the gas turbine, as well as how the lost performance is recovered in the compressor. This will be documented by data from the two gas turbines analysed.

## 4.1 Preventative methods

Different methods to prevent gas turbine, and especially compressor deterioration are many. The methods relevant to this study are described below.

### 4.1.1 Inlet Air Filtration

Layout of the gas turbine inlet filtration is given in figure 5. Filtration systems for gas turbines are important for successful operation. It minimizes the occurrence of foreign object damage, erosion, fouling and corrosion in the inlet air stream. Choosing the right filter configuration for gas turbines can be a difficult task. Site-specific contaminants and how they vary in the future and with seasons are challenging to predict. Also, the filter must fit with the operational philosophy and goals of the gas turbine [6].

As the filter will restrict the air flow, there will be pressure loss occurring at the intake. This pressure loss will affect the gas turbine performance as decreasing inlet pressure must be compensated with higher fuel consumption and is lowering the power output. This effect seems to only increase as the filter loading increases and inevitably the saturation limit will be reached. Although this is generally not a desired effect, studies have shown that poorer inlet air quality still contributes more to gas turbine degradation. It is a trade-off that must be taken into consideration when choosing inlet filtration.

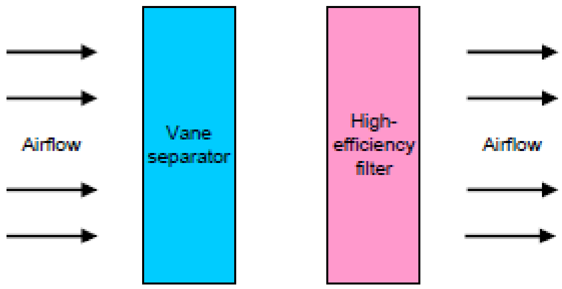



*Figure 5: Layout of the gas turbine inlet filtration system*

### 4.1.2 Anti-Ice systems

A recurring problem when operating in varying conditions offshore is the formation of ice at low temperatures and high humidity. Ice can build up on the inlet components and increase the pressure drop across the inlet section, causing performance loss. In extreme cases it causes foreign object damage and compressor surge. Ice formation forms in the inlet section at humidity greater than 65 % and temperatures less than 4.4°C [10].

### Bleed air

Gas turbine A comes equipped with an anti-icing unit drawing warm air (referred to as bleed air) from different stages in the compressor to prevent icing. Bleeding off the warm air from these compressor stages does not significantly impact the performance. The inlet air is usually heated up 5-10°C for safe operation from ice but warming up the compressor inlet air will decrease its efficiency as the inlet density decrease.

### Radiator

Gas turbine B comes equipped with a liquid media heat exchanger prior to the inlet utilizing the waste heat coming from the turbines. This method of preventing ice formation does have a tendency of overheating the inlet air, affecting the inlet air flow just as the bleed air method. Figure 6 given below is indicating that the sudden rise in inlet air temperature ($P_2$) at low ambient temperatures is resulting in an efficiency drop. This too reflects the time of year and the winter season.

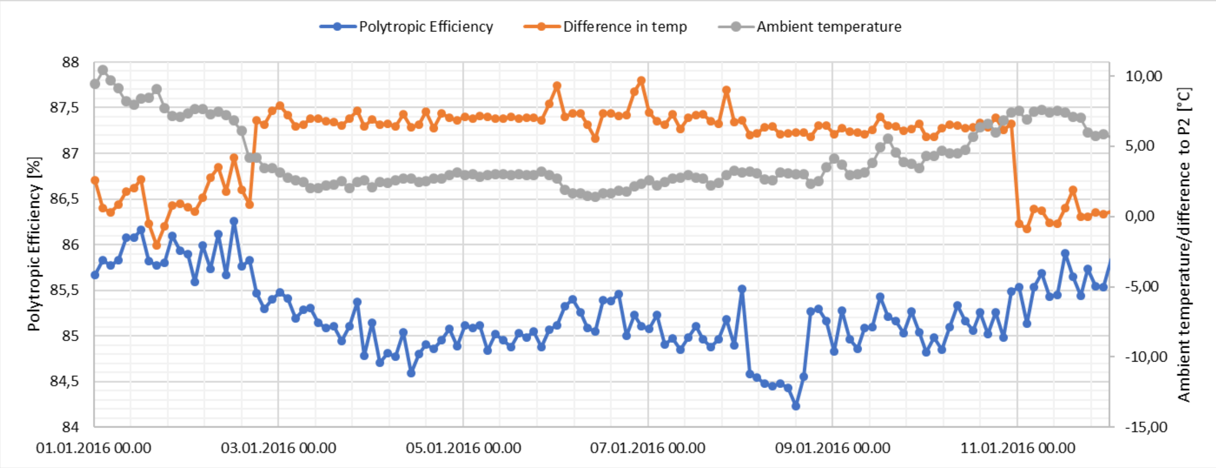



*Figure 6: 11-day monitored efficiency vs. temperature difference at the inlet of compressor B*

### 4.1.3 Coating

Special coating and surface finish of the blading in gas turbines have been shown to decrease the roughness of the surface. This reduces the non-recoverable deterioration of corrosion/erosion and the deposit of contaminations' ability to stick to the blades. This subsequently improves the washing efficiency. Costly coating and surface finish must be a trade-off against the blade lifetime.

## 4.2 Performance Recovery

To recover the performance lost due to compressor fouling there are several methods of cleaning, but due to superior efficiency and economical concerns the water wash techniques are the most common. Techniques like hand and foam cleaning will not be covered in this study.

### 4.2.1 Offline water wash

Also called crank washing this technique requires the gas turbine to be shut down and cool before wash. This method is by far the most effective of the two water wash techniques, which can often achieve extreme performance recovery. Due to the strict requirements of gas turbine shut down and low temperature (approx. 90 °C), this method is always a trade-off against costly downtime [3]. As the gas turbine drives key processes on platforms (power generation, compressor drive etc.), offline water wash operation is seldom favoured and usually done during other necessary maintenance tasks [11].

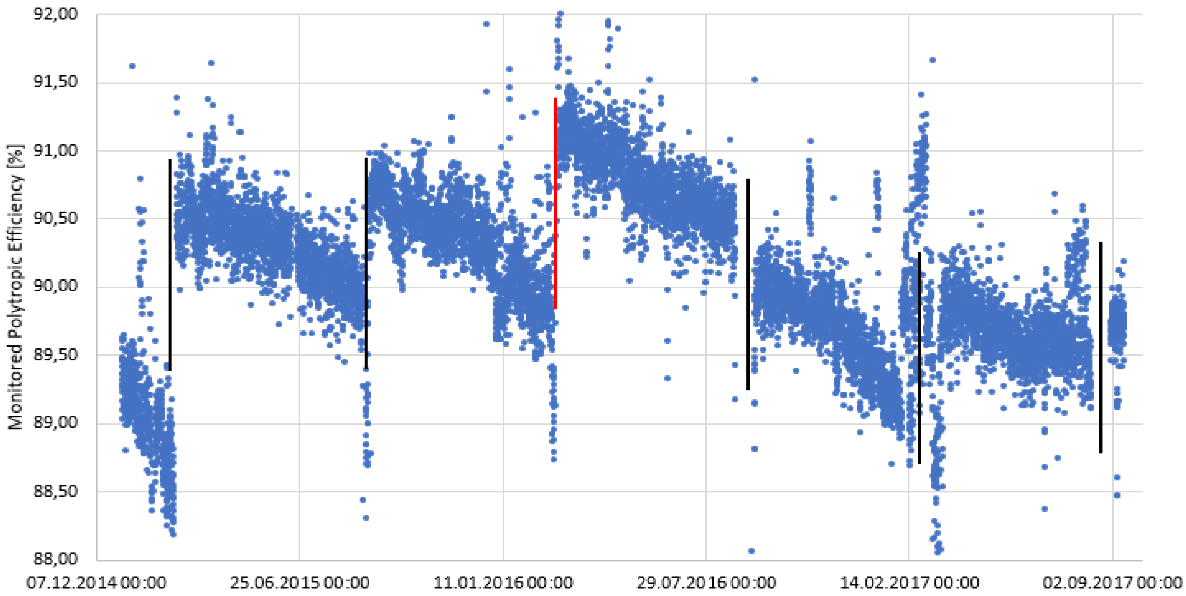



*Figure 7: 2-year monitored efficiency with indication of offline wash of compressor A*

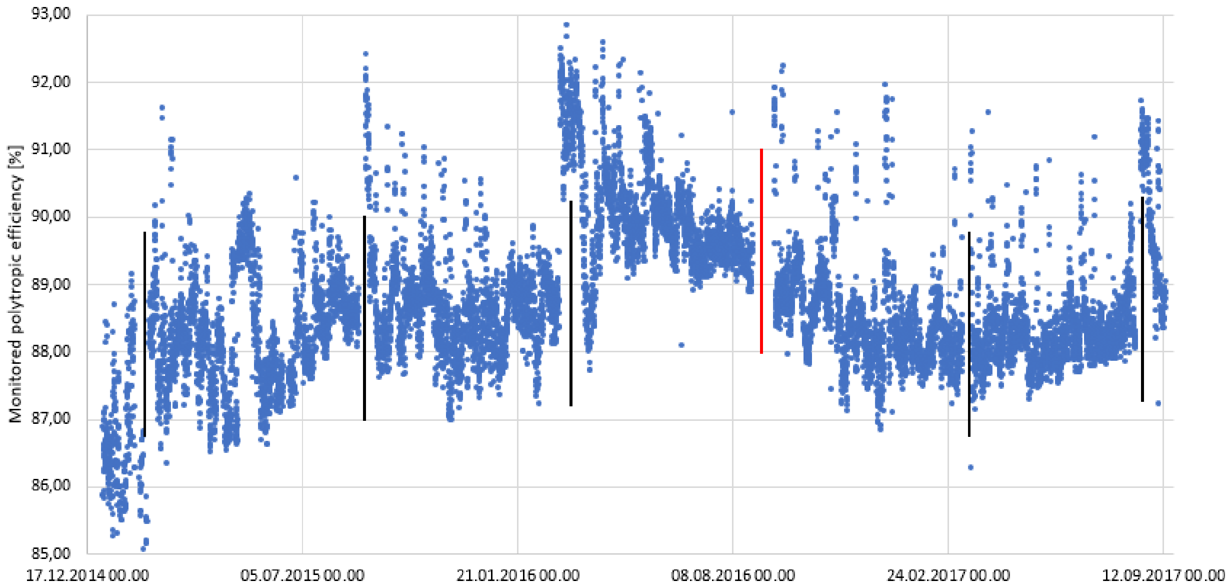



*Figure 8: 2-year monitored efficiency with indication of offline wash of compressor B*

As can be seen on figures 7 and 8 given above offline wash is done in 6-month intervals through January 2015 to September 2017. The black lines indicate washing and the performance recovery is easy to notice for compressor A, but not for compressor B due to variable load and poor instrumentation. This will be covered more in chapter 5. The red line in figure 7 and 8 indicates that there has been a GG change and the sudden jump in performance is evident. These changes are done every 25000 to 30000 hours. The change of a PT is every 75000 to 90000 hours [11].

### 4.2.2 Online water wash

A washing regime not requiring down time is the online water wash, which is a considerable advantage. It is not a widely used technique but is gaining ground and are shown to decrease the need of offline water wash significantly. This technique injects water at 60°C in to the inlet stream daily. The water-to-air ration is between 0.5-1.0% (mass fraction) which is recommended from the GE guidelines [11]. Below (figure 9) the performance recovery of online water wash in a 6-month operating period between offline wash is shown. The deterioration of performance is about 1% which is relatively low compared to no online wash which is reported to deteriorate by as much as 4.5 % [11]. Other studies [17] have operated with a 4-month offline water wash regime which have significantly higher costs as the gas turbine must go offline 3 times a year, instead of 2.

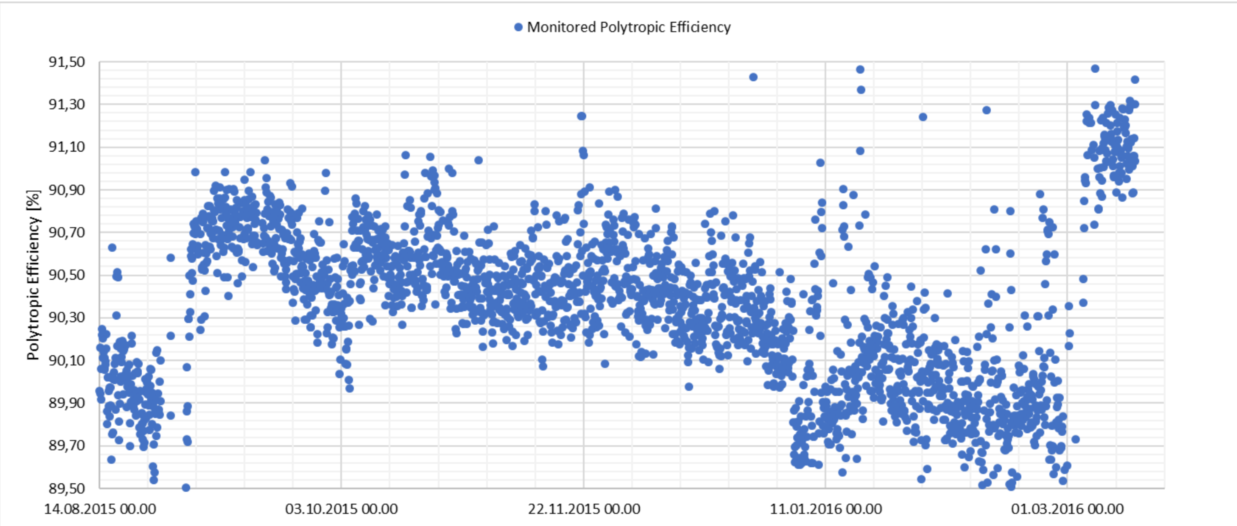



*Figure 9: 6-month monitored efficiency plot showing online wash*

### Challenges of online water wash

Even though there are considerable advantages from online water wash there are a few risks that needs to be mentioned. For DLE combustors the burning mode can change, for cold ambient conditions icing can occur at the compressor inlet, the extinction of the combustor flame (also called Flame out) and high water ratios in the long term can cause corrosion and erosion [11]. There is evidence to support higher ratios of water from 0.8 to 2% without problems [14].

# 5 Performance analysis

This chapter will cover aspects of compressor performance analysis using data sets from the two evaluated compressors and validate the performance trends in simulation software. It will explore how site and operational data impacts the overall performance and its deterioration and look at the compressor instrumentation.

## 5.1 Monitoring instruments

Operational parameters and how they are measured are an important aspect to include when looking at the compressor performance. Whether the measurements are static or dynamic, where in the gas turbine the sensors are located, and the accuracy of the instruments are factors which are affecting the measurements.

### 5.1.1 Location of the measuring instrument

The figure below shows where the operational parameters are located on the gas turbine. The location is standard for all GE LM2500 machines and the two evaluated gas turbines' parameters are therefore assumed comparable.

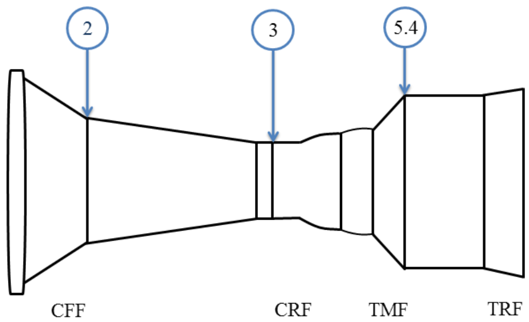



*Figure 10: Instrument location on evaluated gas turbines [17]*

#### Compressor inlet pressure ($P_2$)

The compressor inlet pressure probe is located just behind the bellmouth in the compressor front frame (CFF) and will therefore account for all changes due to inlet systems. Both gas turbine sensors measure static pressure but differs as operating data for engine A and B are given in absolute and gauge respectively. This is important to account for as gauge means relative to ambient conditions which is varying.

#### Compressor inlet temperature ($T_2$)

The temperature of the inlet stream is measured together with $P_2$ in the same sensor at the CFF to account for changes in the filter systems and bellmouth. As described in chapter 4.1.2 the temperature is expected to increase when the anti-ice systems are active.

#### Compressor discharge pressure ($P_3$)

Sensors for measuring the pressure discharged from the compressor are located in the rear frame of the compressor (CRF). This is also measured in static conditions which is GE standards and given in gauge values. The discharge pressure is expected to decrease as the compressor deteriorates due to lower pressure ratios as mentioned in chapter 4.2.

### Compressor discharge temperature ($T_3$)

The discharged temperature is alongside $P_3$ measured in the CRF and is expected to increase as the compressor fouling increase the friction in the flow area.

### Exhaust gas temperature ($EGT/T_{5.4}$)

Measured in the turbine mid frame (TMF) the exhaust gas temperature (EGT), also referred to as $T_{5.4}$, is expected to rise as the deterioration increase. This is, as explained in chapter 3.4.1, due to increased fuel flow as the control system must compensate for the reduced air flow.

### Gas generator shaft speed ($N_1$)

The GG speed is measured in the gear box and is expected to vary with the ambient conditions and gas turbine load. When the air flow is restricted the GG speed will normally increase if the load is steady. This is further discussed in chapter 5.2.1 of corrected parameters.

## 5.2 Analysis of parameters

### 5.2.1 Thermodynamic analysis

Throughout this study the monitored performance of the compressors and how parameters affect them are calculated in the monitoring program and given in polytropic efficiency by this equation:

$$\eta_p = \frac{\ln\left(\frac{T_{3i}}{T_2}\right)}{ln\left(\frac{T_3}{T_2}\right)} = \frac{ln\left[\left(\frac{P_3}{P_2}\right)^{\frac{(\kappa-1)}{\kappa}}\right]}{ln\left(\frac{T_3}{T_2}\right)} \tag{13}$$

The real air heat capacity ratio (κ) is calculated using the mean temperature over the compressor ($T_2$,$T_3$).

### 5.2.2 Corrected parameters

Varying ambient conditions such as temperature and pressure makes comparison of operating parameters hard due to the change in air flow gas properties. Due to this a correction of parameters to a reference condition is a widely used method. These methods have been extensively covered through time and documented by Volponi [16] and Krampf [15].

The method of correction is based on the International Standard Organization (ISO) correction method. By bringing the operational data to a reference point of temperature, pressure and humidity based on the Buckingham pi-theorem and non-dimensional analysis of gas and energy laws, correctional formulas can be developed. ISO conditions are defined as 288.15 K of temperature, 101.325 kPa of pressure and 60 % relative humidity. Accounting for change in conditions through the inlet section inlet parameters ($P_2, T_2$) are utilized and not ambient.

Krampf [15] discuss some weaknesses in the method as it is not considered "perfect". Concerns about the accuracy at high variations from the given ISO conditions, mechanical limits at low ambient conditions and the fact that varying RH is not considered are important to remember. The compressor section deals with air alone and are therefore more suited for performance prediction than other sections. This study will take a closer look at the application of corrected speed ($N_{1c}$), discharge pressure ($P_{3c}$) and temperature ($T_{3c}$) to see if they reflect the compressor deterioration satisfactory. Manufacturers exponent factor is not included in this stage of the study in order to avoid complications as the goal of this report is to only show the principal of the method.

### Corrected speed

Corrected GG speed is related to rotational speed (rpm) and the acoustic velocity by Mach number. The corrected speed, denoted as $N_{1c}$ in this study, which corrects for varying inlet temperature $T_2$ is given by:

$$N_{1c} = \frac{N_1}{\sqrt{\theta}} \tag{14}$$

where the correction factor is:

$$\theta = \frac{T_2\ K}{288{,}15\ K} \tag{15}$$

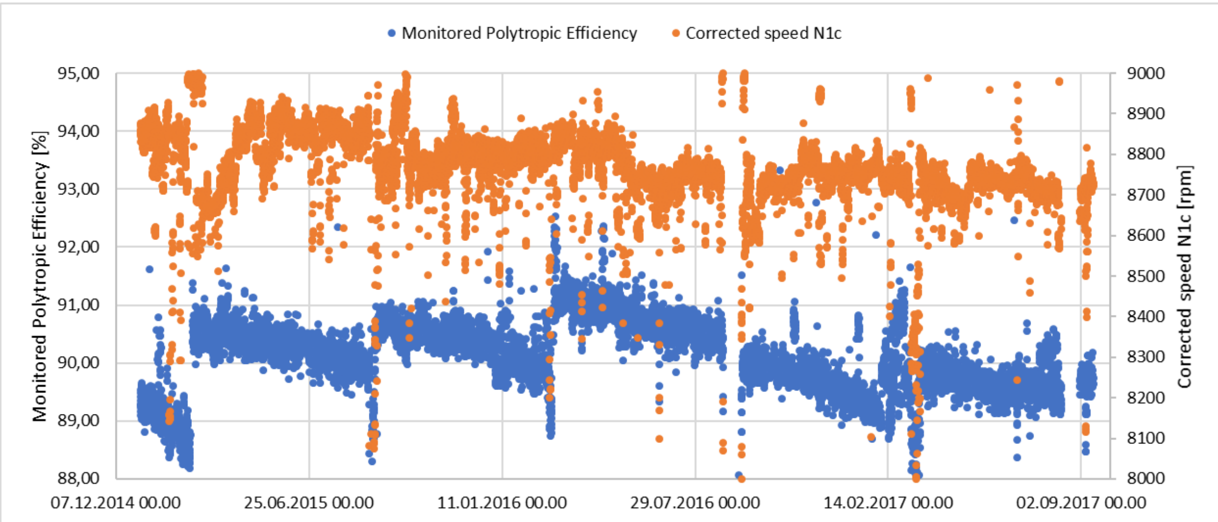



*Figure 11: 2-year polytropic efficiency vs. corrected speed N1c of compressor A*

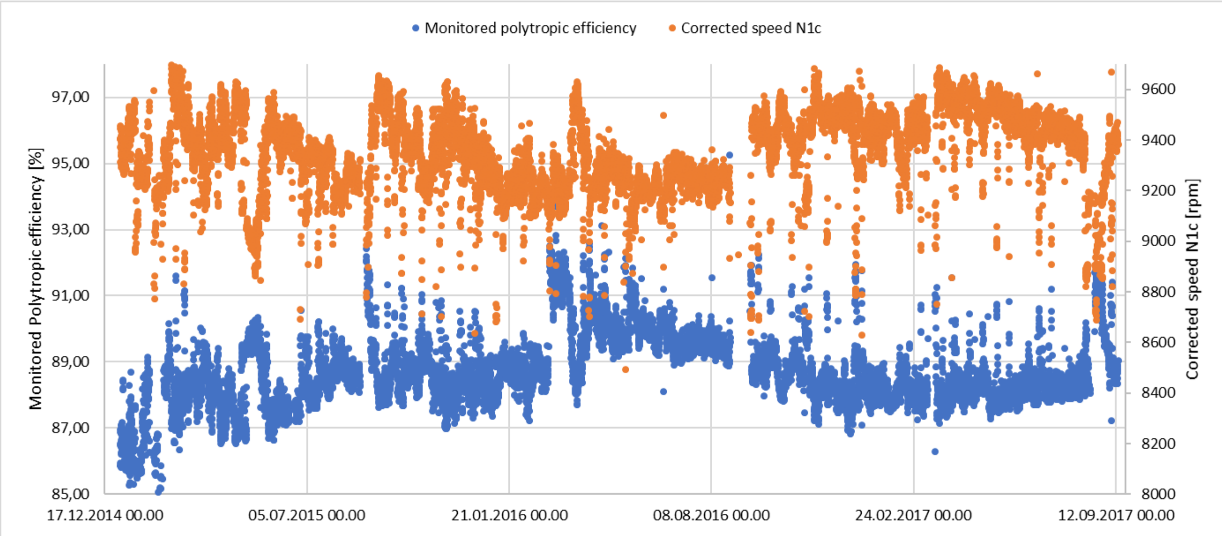



*Figure 12: 2-year polytropic efficiency vs. corrected speed N1c of compressor B*

It is expected that the GG speed of the compressor will increase with the deterioration due to restricted mass flow. This is evident in figure 11 and 13 that compressor A experience steady load which is a requirement for using GG speed for trend comparison.

Figure 12 and 14 shows large irregularities in the trend comparison for compressor B. This is expected as the gas turbine experiences large variations in load. Figure 14 is therefore a very good example of how difficult it can be to get reliable performance data from engines like this.

To emphasise how well the corrected speed reflects the efficiency trends, especially in compressor B, an interval of one operating period has been chosen in Figure 13 and 14 from July 2015 - March 2016.

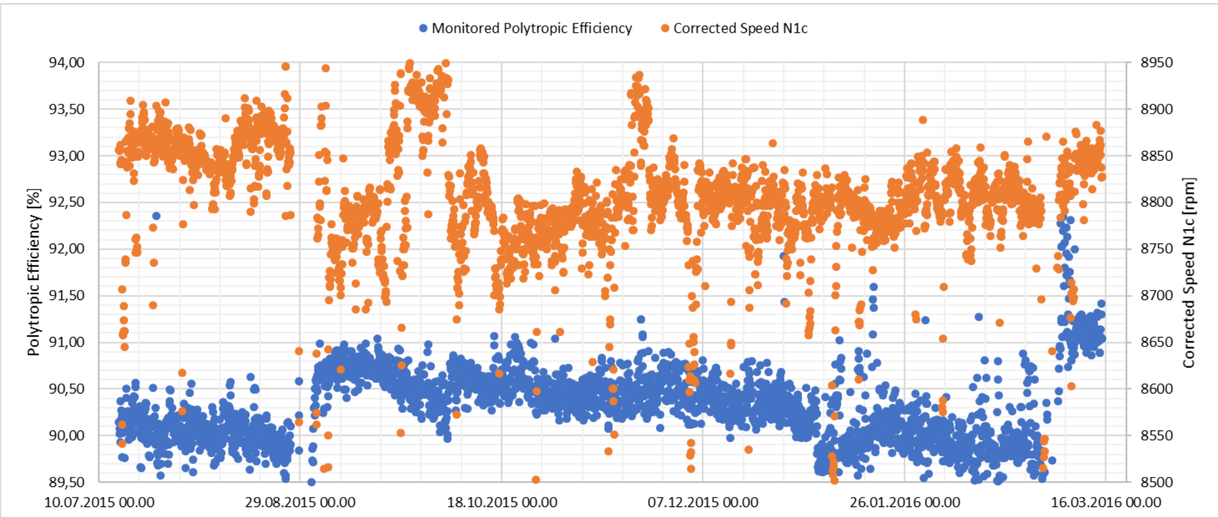



*Figure 13: 6-month polytropic efficiency vs. corrected speed N1c of compressor A*

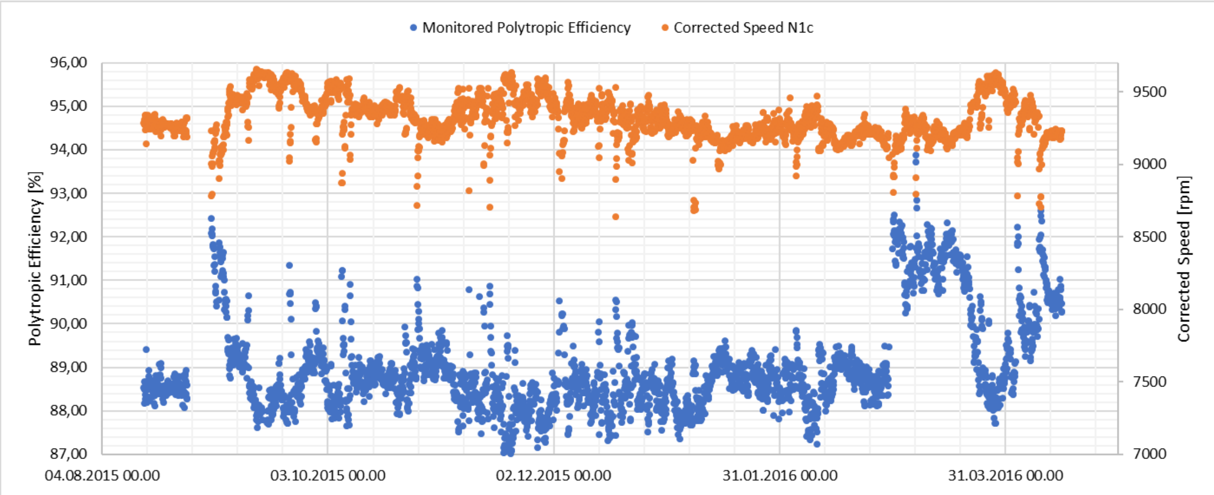



*Figure 14: 6-month polytropic efficiency vs. corrected speed N1c of compressor B*

## Corrected discharge pressure

Corrected pressure, called $P_{3c}$ in this study, will consider the varying compressor inlet pressure ($P_2$) and is given by:

$$P_{3c} = \frac{P_3}{\delta} \tag{16}$$

where the correction factor is:

$$\delta = \frac{P_2\ kPa}{101.325\ kPa} \tag{17}$$

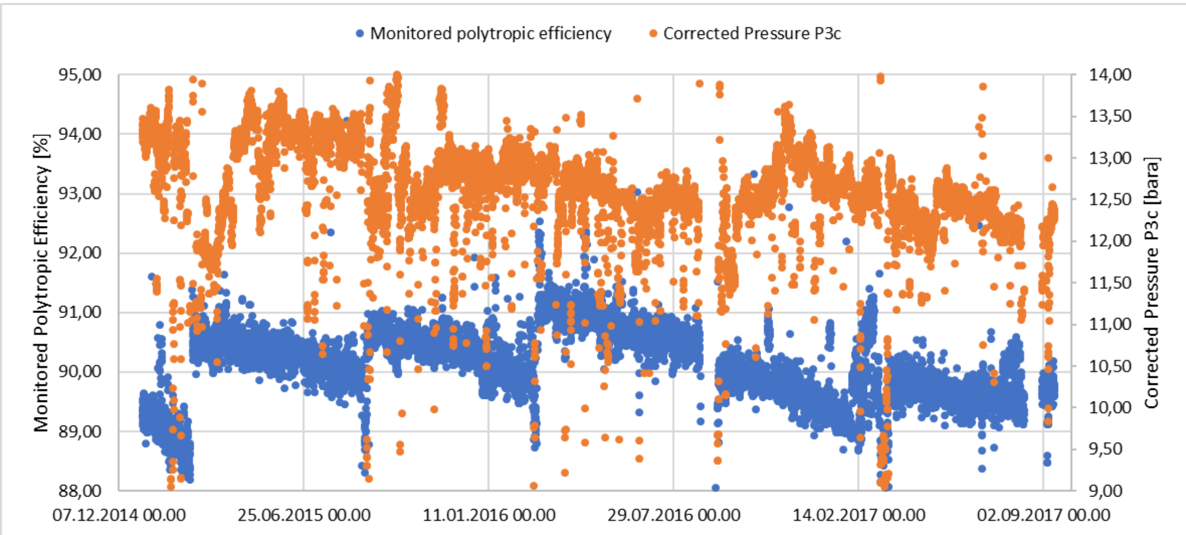



*Figure 15: 2-year polytropic efficiency vs. corrected pressure P3c for compressor A*

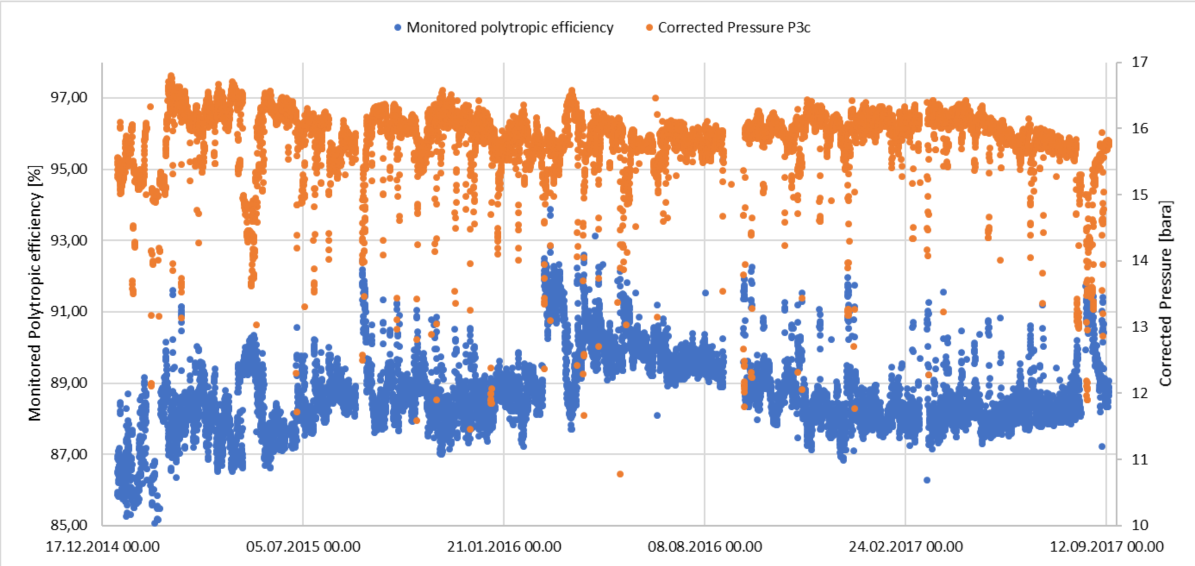



*Figure 16: 2-year polytropic efficiency vs. corrected pressure P3c for compressor B*

Again, the figure of gas turbine A (15) is showing more accurate results than B (16), but not accurate enough. The expected decrease in discharge pressure is not sufficient for reliable trend comparison and are therefore not further investigated in this study.

### Corrected compressor discharge temperature

The corrected temperature, called $T_{3c}$ in this study, will take into consideration the varying compressor inlet temperature ($T_2$), and it seems only reasonable that $T_3$ should vary with it. It is given by:

$$T_{3c} = \frac{T_3}{\theta} \tag{18}$$

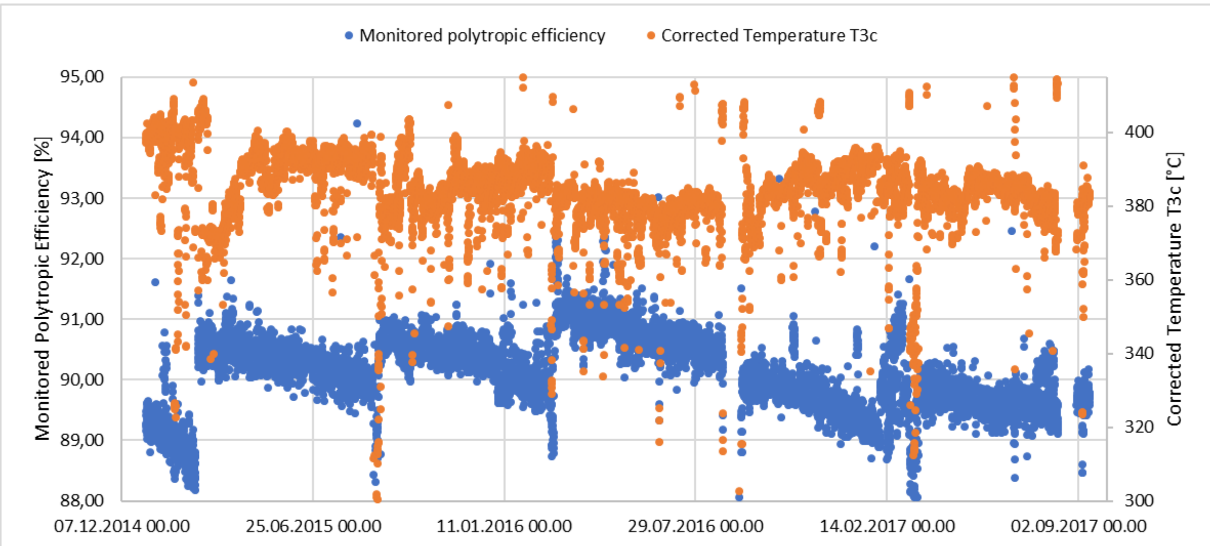



*Figure 17: 2-year polytropic efficiency vs. corrected temperature T3c for compressor A*

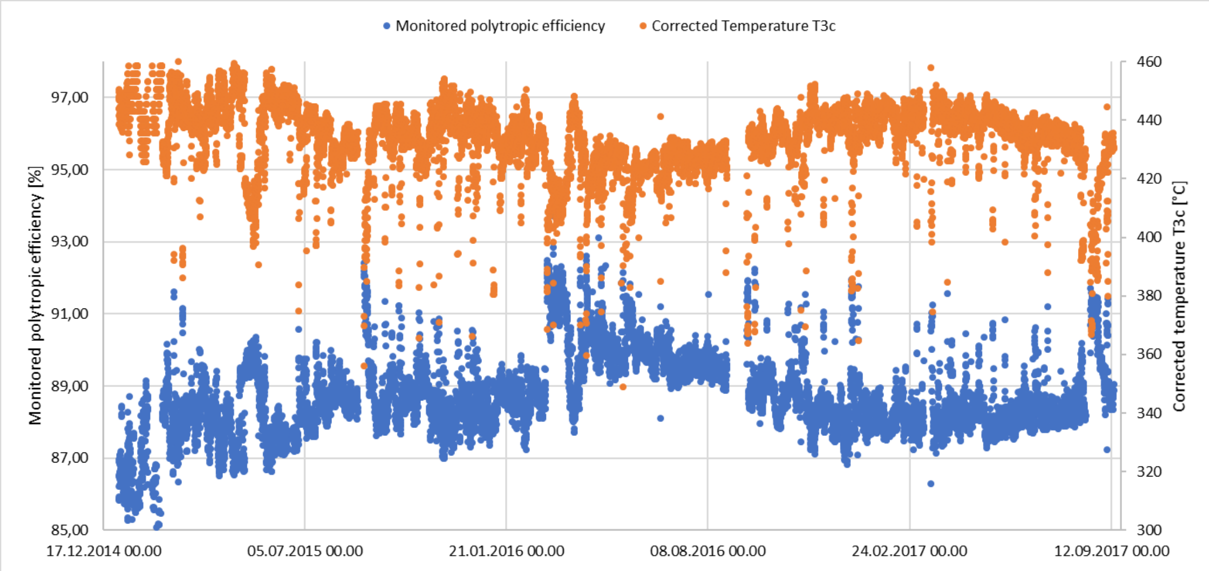



*Figure 18:2-year polytropic efficiency vs. corrected temperature T3c for compressor B*

Both Figure 17 and 18 are showing comparable trends as the expected increase in temperature correlates with the decrease in efficiency caused by deterioration.

### Corrected exhaust gas temperature

The corrected exhaust gas temperature, called $T_{5.4c}$ in this study, will also consider the varying inlet temperature ($T_2$) and have the same correction formula as $T_{3c}$:

$$T_{5.4c} = \frac{T_{5.4}}{\theta} \tag{19}$$

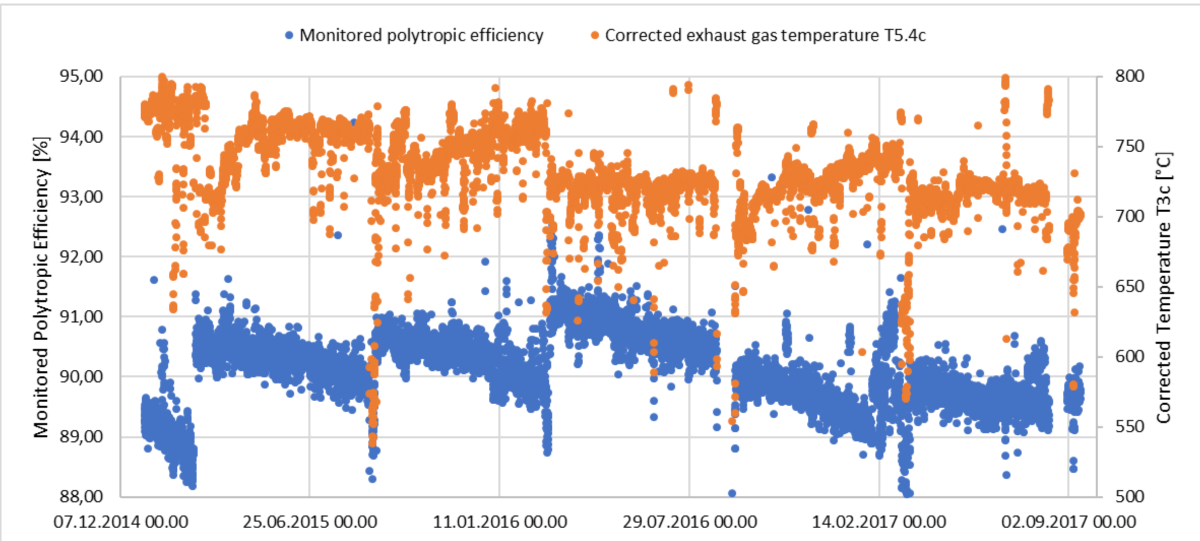



*Figure 19: 2-year polytropic efficiency vs. corrected temperature T5.4c for compressor A*

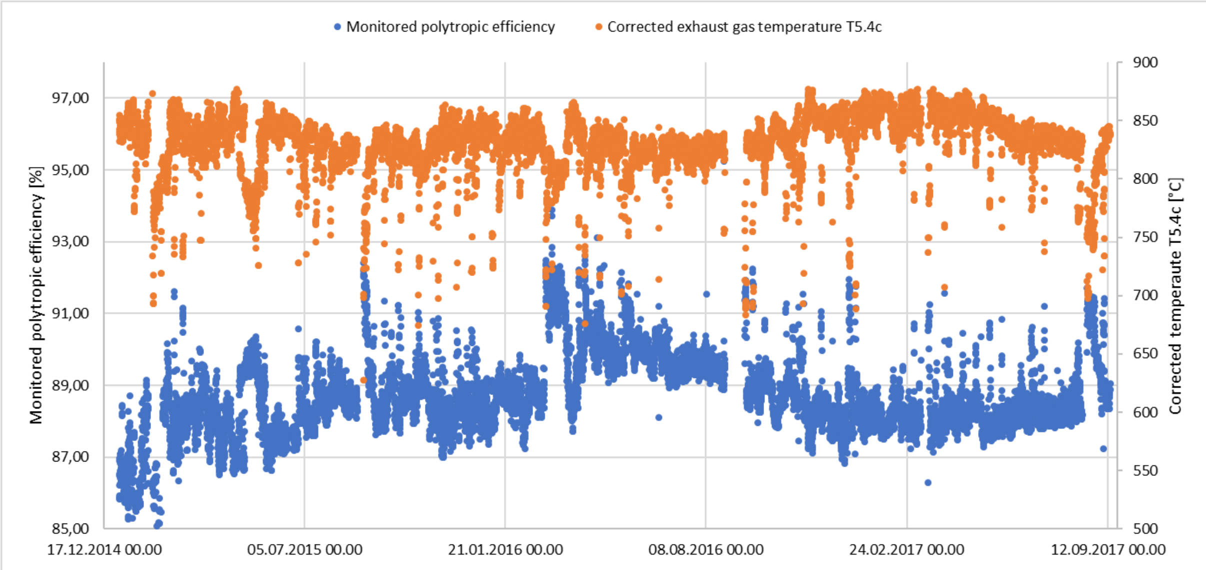



*Figure 20: 2-year polytropic efficiency vs. corrected temperature T5.4c for compressor B*

The expected rise in exhaust gas temperature from deterioration, explained in chapter 3.4.1, is clear to see for both engines in Figure 19 and 20. It reflects the efficiency trend better than $T_{3c}$ especially for compressor A and is therefore regarded as more reliable.

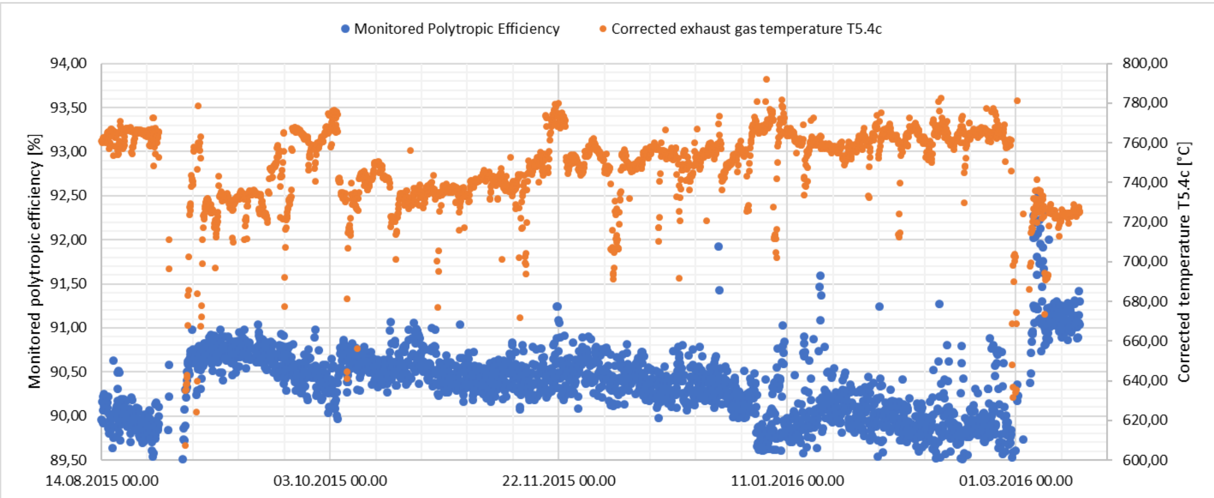



*Figure 21: 6-month polytropic efficiency vs. corrected EGT, T5.4c, of compressor A*

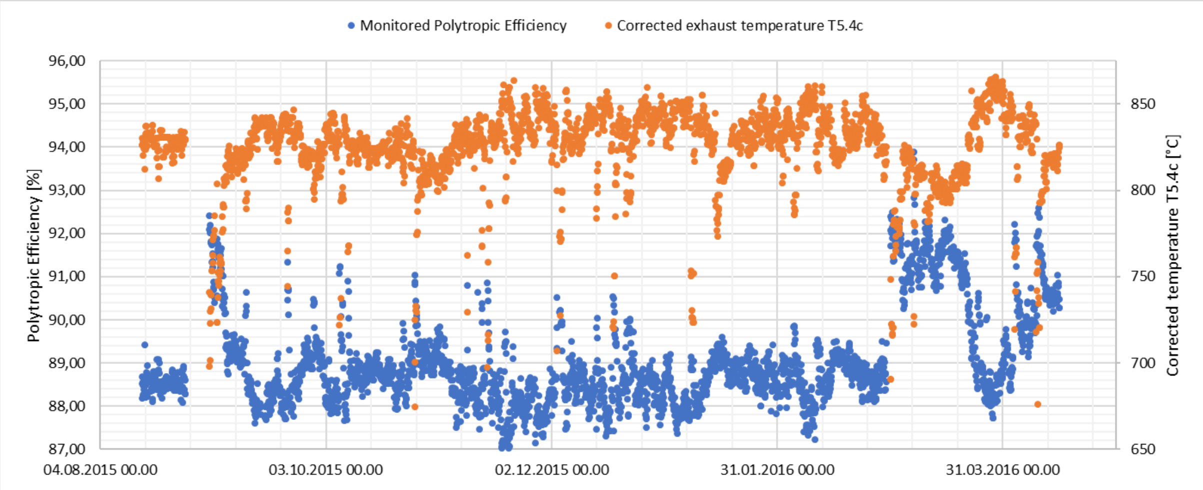



*Figure 22: 6-month polytropic efficiency vs. corrected EGT, T5.4c, of compressor B*

A closer look at one operating period, August 2015- April 2016, is shown in Figure 21 and 22 in order to show how accurately the efficiency trend is reflected in the corrected exhaust temperature

## 5.3 Performance validation

A process simulation is established in Aspen HYSYS of a LM2500 compressor with humid air inlet to simulate real air conditions offshore. The analysis is performed to validate the compressor response to operational data and to better predict the gas turbine performance.

The simulation done in this study is processing large numbers of operational data spanning over a 2-year period. To implement the data Aspen Simulation Workbook add-in function in Excel has been utilized.

### 5.3.1 Equation of state (EOS)

Choosing an EOS is an important consideration as the different equations can give differing results in the simulation and can affect the performance prediction. An EOS is used to describe the relation between state variables and physical properties such as temperature, pressure, and humidity.

Over 100 equations of state have been proposed through history in an attempt to improve the ideal gas law. The inverse relation between gas volume and pressure where first supported by Robert Boyle and later formulated by Amedeo Avogadro into what we know today as the ideal gas law:

$$PV_m = RT \tag{20}$$

#### Van der Waals Equation

Improvements on the ideal gas law were made by Johannes D. Van der Waals who argued that intermolecular forces become significant when temperature and pressure increase. Therefore, Van der Waal improved the ideal gas law to incorporate correction to the volume of molecules and their interactions:

$$\left(P + \frac{a}{V_m^2}\right)(V_m - b) = RT \tag{21}$$

Where *a* refers to the degree of reaction of gas molecules, the term $V_m - b$ represents the "free volume" where molecules can move around. *b* is linked to the volume of the gas molecules and their repulsive forces and both *a* and *b* are constant unique to each gas molecule and are independent of pressure and temperature. The equation can be re-arranged to:

$$P = \frac{RT}{V_m - b} - \frac{a}{V_m^2} \tag{22}$$

Which is balanced by the fact that $\frac{RT}{V_m - b}$ describes attraction pushing molecules together alongside *P* and $\frac{a}{V_m^2}$ representing the repulsive forces between molecules.

At higher pressures the repulsive forces prevail over the attractive ones and this equation, which is reasonable at middle pressures, presents inconsistencies. The constants of the equation and the critical parameters are given with:

$$P_c = \frac{a}{27b^2}, T_c = \frac{8a}{27bR}, V_c = 3b \tag{23}$$

For a single component a critical parameter is the highest value the parameter can have where liquid and vapour can coexist. Although, for multicomponent systems the two-phase region can extend beyond the systems critical point.

### Redlich-Kwong Equation (MRK)

The Van der Waals equation was modified to improve the equations ability to reproduce fluid parameters at higher temperatures and pressures. The first term in equation 21 was modified giving:

$$\left(P + \frac{a}{V_m(V_m + b)\sqrt{T}}\right)(V_m - b) = RT \tag{24}$$

This opened for utilizing the equation for pure gases and their mixtures, but also for *NaCl* and $H_2O - CO_2$ fluids.

### Soave-Redlich-Kwong (SRK)

The MRK EOS developed Soave-Redlich-Kwong equation made modifications to the correction factor:

$$P = \frac{RT}{V_m - b} - \frac{a}{V_m(V_m - b)} \tag{25}$$

Using the same equation as MRK, Soave made some adjustments to the *a* factor:

$$a = 0{,}42748\frac{R^2T_c^2}{P_c}[f(T)]^2 \tag{26}$$

The function of the reduced temperature $T_r$ and the acentric factor ω where incorporated into the equation:

$$f(T) = 1 + k\left(1 - \frac{T}{T_c}\right) \tag{27}$$

$$k = 0{,}480 + 1{,}57\omega - 0{,}176\omega^2 \tag{28}$$

Accounting for the molecules without a spherical form the acentric factor is introduced. The molecules without spherical form have $\omega = 0$. K.S. Pitzer introduced it and is calculated by:

$$\omega = -\log_{10}\left(\frac{P_{sat}}{P_c}\right)_{T_r=0,7} - 1 \tag{29}$$

Soave did not alter the volume correction factor *b*, and maintained it as:

$$b = 0{,}08664\frac{RT_c}{P_c} \tag{30}$$

The modification done in SRK presented a marked impact on calculations of hydrocarbons and the biggest advancements upon which later EOS where built. This includes among others Peng-Robinson (PR) a EOS very similar to SRK but are slightly altered to focus more on petroleum systems, especially gas/condensate systems [19].

The EOS chosen for the simulation in this study is the SRK (Soavo-Redlich-Kwong) and is the applied EOS by Statoil and NTNU for wet gas compressor analysis with air/water relations. It is also utilized in the ongoing doctoral study mentioned in the introduction and makes the results more comparable. Furthermore, by findings of Samnøy [17] the comparison of SRK to other common EOS gave insignificant differences.

### Hysys model

The HYSYS model utilized for this simulation was agreed [13] to be the same model used for the ongoing doctoral study for the sake of comparison and to to easier validate the work for supervisors and implementation in further work on the evaluated gas turbines.

Figure 28 in Appendix A is showing the layout of the model. A saturation operator is implemented to determine the humidity (RH) of the inlet air before it enters the compressor. Only temperature and pressure parameters are available to automatically implement, and RH are agreed to be manually set to 80% [13].

For the real gas calculations of polytropic flow the method documented by Shultz [18] was chosen. For closer inspection see chapter 2.2.1.

### Compressor A

The analysed monitored polytropic efficiency for compressor A is shown in Figure 23. As long as the interest is in the relative change of the efficiencies, they follow very similar trends. The deviation around 31.12.2016 in simulated efficiency is a result of mismeasurement and is addressed in chapter 3.1.2. Even when focusing on shorter intervals, as seen in figure 24, the trend is similar.

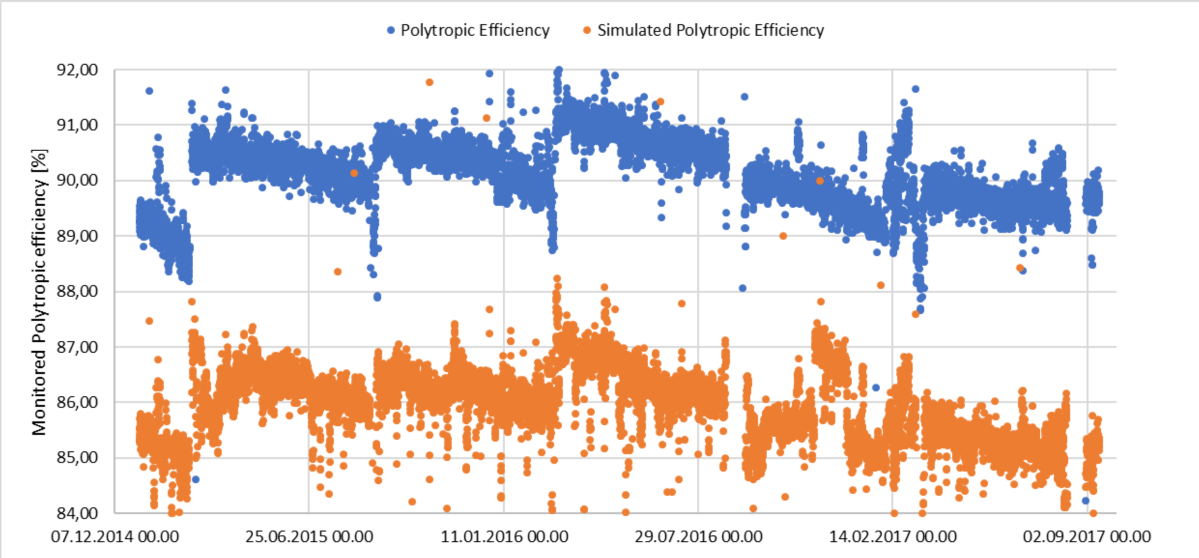



*Figure 23:2-year comparison of the monitored and simulated efficiency of compressor A*

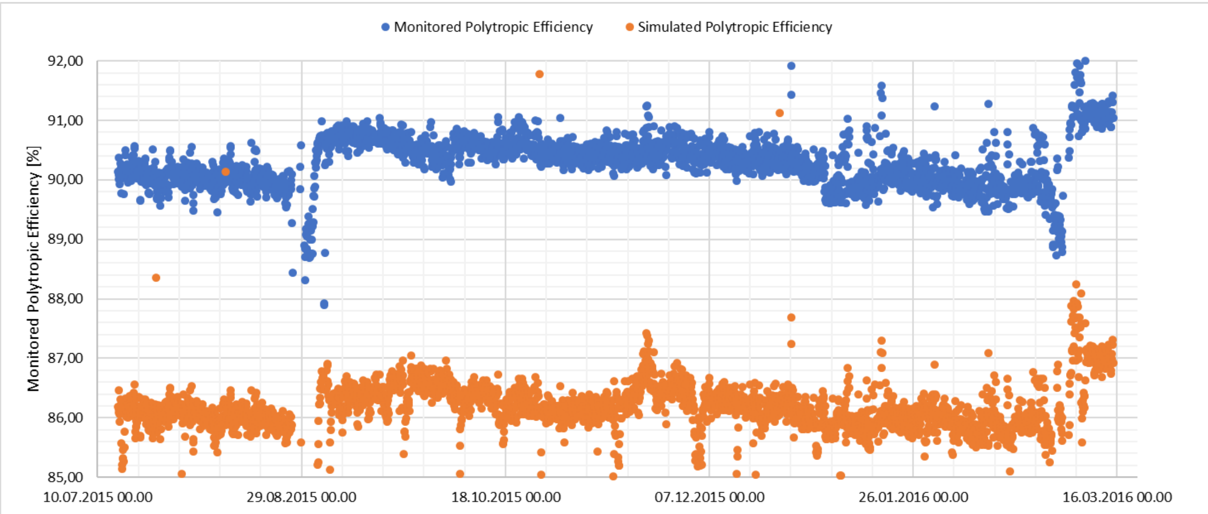



*Figure 24: 6-month comparison of the monitored and simulated efficiency of compressor A*

The monitored polytropic efficiency calculated by the monitoring program are therefore in accordance with the simulated polytropic efficiency for compressor A and hence assumed applicable for further analysis.

### Compressor B

For compressor B shown in Figure 25 the same conclusion can be made. The trend similarities are clear, even when focusing on only one operating period as in figure 26.

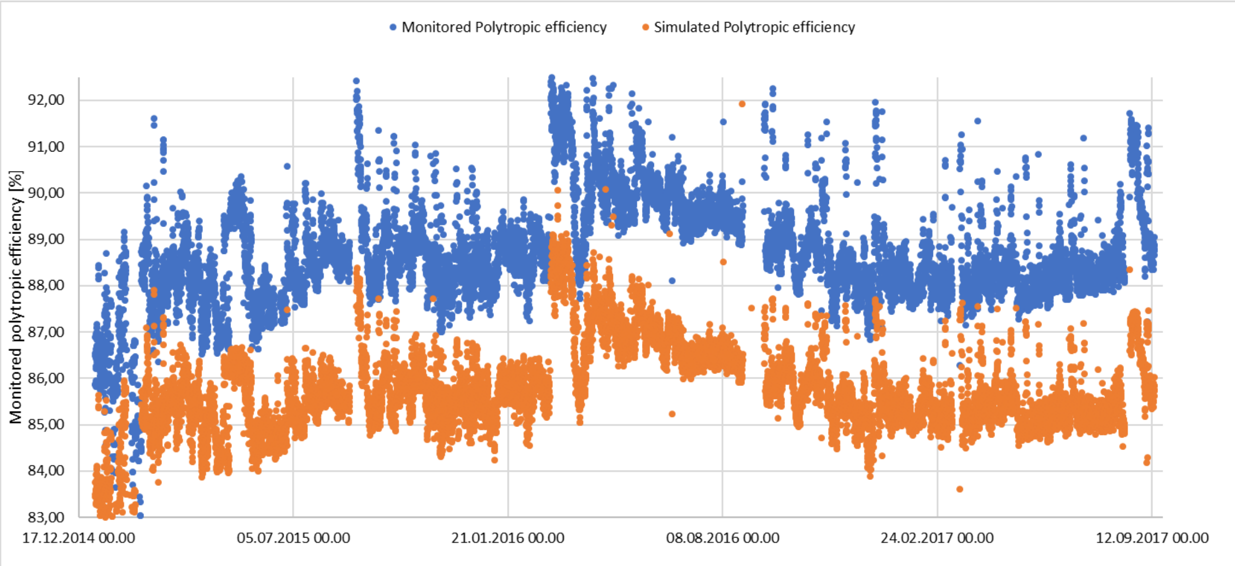



*Figure 25: 2-year comparison of the monitored and simulated efficiency of compressor B*

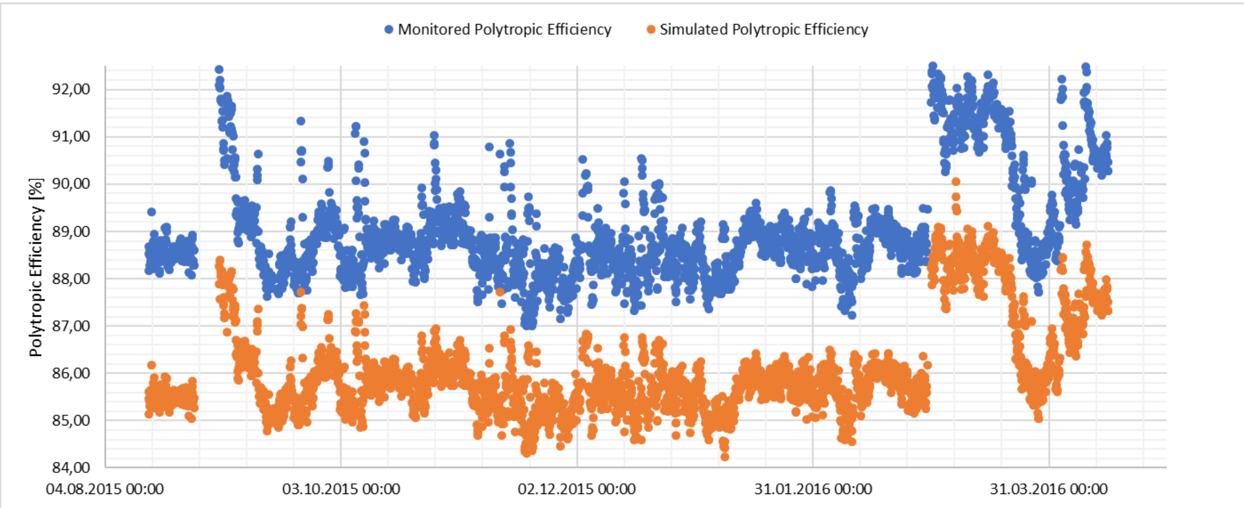



*Figure 26: 6-month comparison of the monitored and simulated efficiency of compressor B*

The polytropic efficiency calculated by the monitoring program is therefore in accordance with the simulated polytropic efficiency for compressor B and hence assumed applicable for further analysis.

### 5.3.2 Sensitivity analysis

As mentioned earlier in this chapter the accuracy of the instrumentations are important factors to consider. The two gas turbines are equipped with two different instruments that have relatively large difference in accuracy. Gas turbine A has the newest instrumentation and therefore a more precise reading of its measurements.

A sensitivity analysis has been conducted to see how this accuracy affects the operational data. The author has been given datasheets containing information on the pressure instruments of both engines, both new and old. The deviation in input, meaning the ± confidence interval given, is plotted against the resulting deviation in output parameter chosen to be polytropic efficiency.

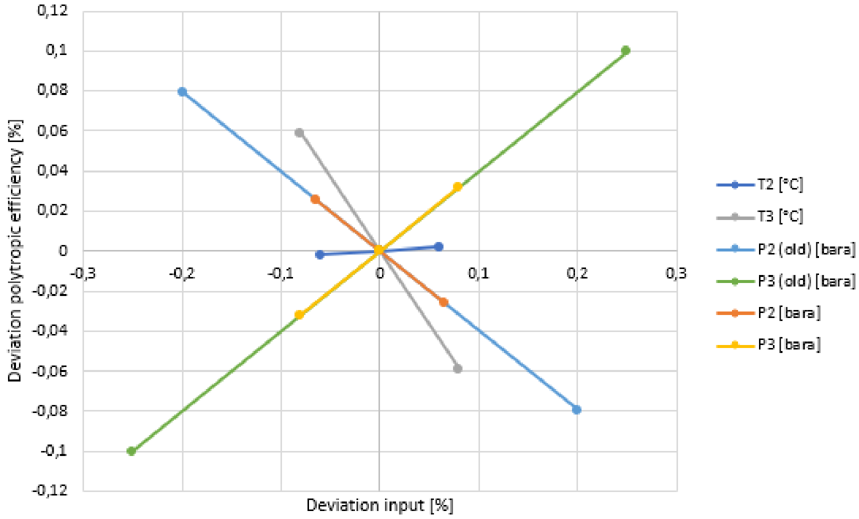



*Figure 27: Sensitivity analysis showing instrument accuracy and their impact on output*

The results are shown in Figure 27 where the old (light blue and green) and new (red and green) pressure instruments for the two gas turbines are clearly shown. This indicate how important it is to have state-of-the-art instrumentation to get reliable measurements and how difficult performance readings can be for gas turbine B.

It is also interesting to see how different the four parameters vary in impact on the polytropic efficiency and, in accordance with Samnøys [17] findings, especially $T_3$. It is important to emphasise that the confidence interval portrayed for the temperature measurements are assumed and not given and are included for the sake of comparison. They have the same values as the pressure instruments.

# 6 Summary and Conclusion

The purpose of this project thesis has been to document the operation of the gas turbines on an offshore platform. Through analyses of two different engines operating different load regimes and equipped with different monitoring instruments the emphasis was on gas turbine deterioration. What operational parameters and conditions contribute to performance deterioration, how the performance is recovered and validating the performance through polytropic flow path analysis utilizing Aspen Hysys. Exploring these points is done to establish a procedure for compressor efficiency variation and trend analysis.

In chapter 3 the different compressor deterioration and operational conditions contributing to performance loss was explored. It was found that ambient conditions such as pressure, temperature and humidity have great impact on the intake air and therefore the performance loss, but more data is needed to make more accurate conclusions. Compressor fouling due to salt has been documented to be the largest contributor to deterioration and the operational consequences are higher $N_1$ and $T_{5.4}$, which are confirmed by trend analyses in chapter 5.

Chapter 4 document prevention and recovery of the performance loss. Here the preventative methods of inlet filtration and anti-icing and their impact on the inlet parameters and subsequently the performance are covered. Pressure drop over the inlet filtration system alongside temperature increase to prevent ice formation in the inlet stream decreases the performance of the compressor. Recovery of performance loss due to fouling is covered through online and offline wash. Figures presented clearly depicts that the offline wash was conducted every six months and the jump in performance is evident for compressor A. Here the load variation of compressor B becomes evident as the recovery is not clear and should be corrected.

In chapter 5 a procedure was developed to analyse corrected parameters and how accurate they portrayed change in performance trends. The parameters found to be most reliable were $N_{1c}$, $T_{3c}$and $T_{4.5c}$ and will be used in further work. All of them having the same correction factor based on $T_2$. SRK was found to be the best EOS and the validation of the operational data concluded that the polytropic efficiency calculated by the monitoring software is applicable for further investigation. The sensitivity analysis pointed out the importance of new pressure instruments as the old instruments could have an impact of up to 0.1 % efficiency.

# 7 Further work

When it comes to the instrumentation of the two gas turbines it is recommended to replace the old instrumentation monitoring pressures in gas turbine B. This would afford more accurate monitoring of the performance.

Operational data containing more details would be recommended for the further work on these machines as they give a more reliable reading and procedures, which will be suggested below. This includes ambient humidity measurements to document the impact on compressor inlet. The resolution of the measurements should be higher to easier analyse the consequences of sudden water ingestion by online water wash. Although $T_{5.4}$ in this study is given to reflect the change in fuel flow it contains insecurities and the actual fuel flow measurement would be of interest. Data from additional gas turbines would be of interest for comparison and add to the accuracy of the work. As it proved difficult to locate where bleed air was utilized, in compressor A for anti-icing purposes, data on this will be of interest.

An ideal baseline was not included in this study and should be generated as part of the work for a procedure implementing load-correction to correct the efficiency for varying operational conditions. Here the parameters corrected for $T_2$ should be utilized.

It would be interesting to develop a complete simulation model in Hysys to get a more complete picture of the operational impact the water washing regimes and ambient conditions can have on the gas turbine as a complete system. This will also support validation of parameters for the entire gas turbine and their efficiency e.g. the high pressure and power turbine.

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# Appendix:

## A - Aspen Hysys Simulation model

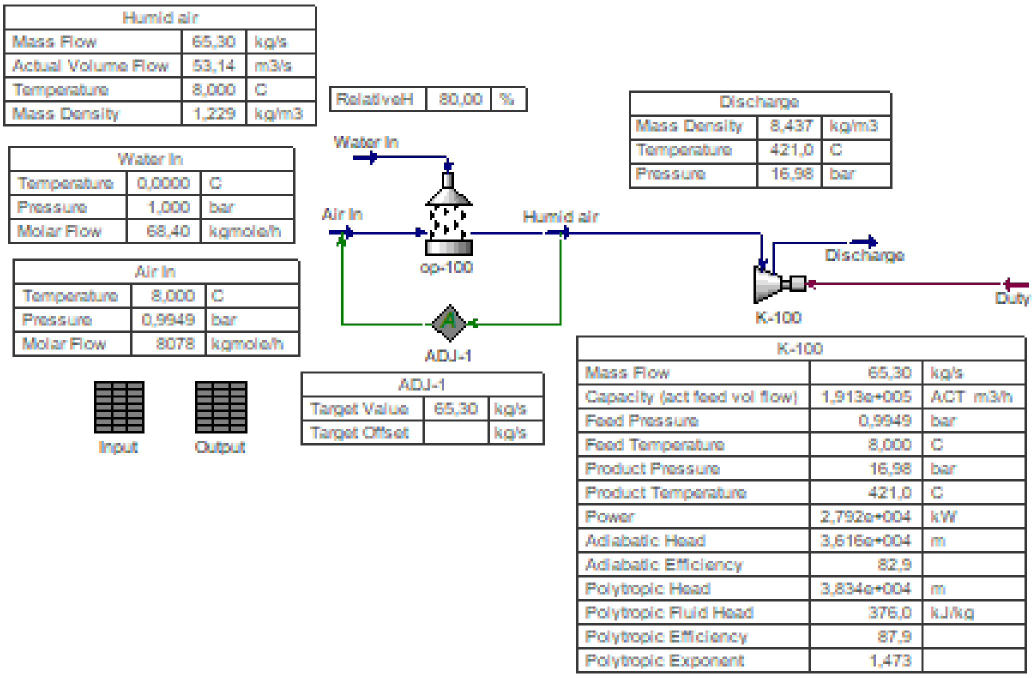


*Figure 28: Aspen Hysys Simulation Model*

## B - Ambient conditions

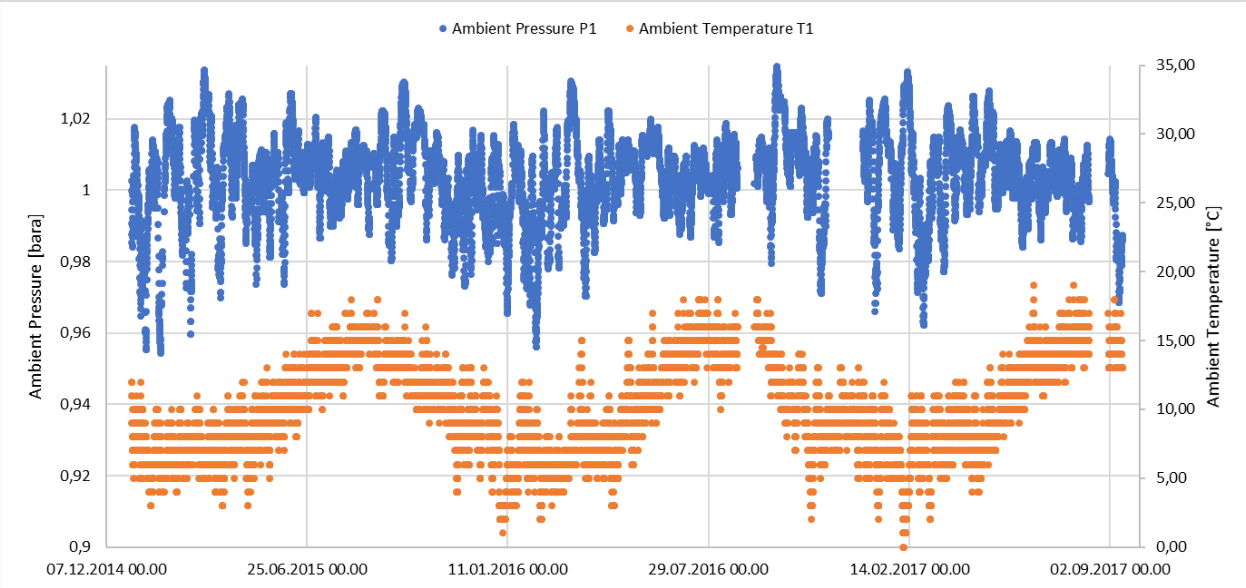


*Figure 29: 2-year ambient conditions for compressor A*

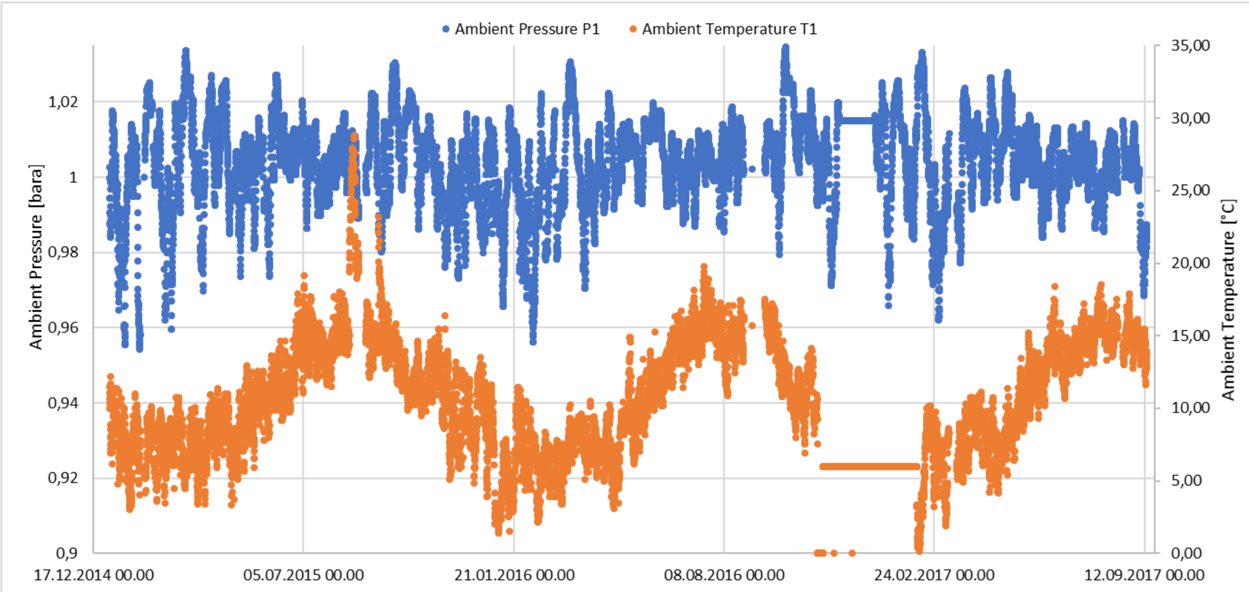


*Figure 30: 2-year ambient conditions for* compressor *B*

## C - Diagram of progress

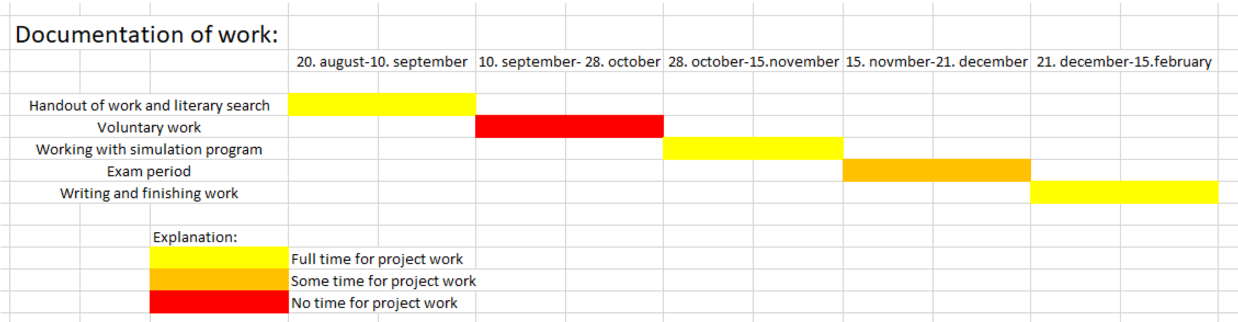


*Figure 31: Diagram of project progress*