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Impact of Degradation on the Operational Behaviour of a Stationary Gas Turbine and in Detail on the Associated Compressor

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Holding a master degree in mechanical engineering and involved in compressor design and operational issues for more than 20 years. Interacting with all involved functions and disciplines as aerodynamic, mechanical design, mechanical integrity, manufacturing, testing and field support. Having been involved in various acting and managerial functions for multiple compressor upgrades/major modifications.

ABSTRACT

Degradation of stationary gas turbines is a quite often observed burden and can impact financial revenues of end-users significantly. In addition to the well-known occurrence of increased pressure losses in the intake filter systems due to clogging (filter loaded with deposits), the compressor has been shown to be one of the main contributors to this change in gas turbine performance. In case of compressor blading and flow path degradation, additionally to a compressor efficiency reduction, the inlet mass flow can be significantly reduced as well. As a consequence also the gas turbine pressure ratio is decreasing. Therefore, compressor degradation acts detrimentally twice on the overall gas turbine power.

Degradation of a compressor in the context of this paper is understood as the behavior of a stationary gas turbine, where intake filter systems are getting clogged and/or dirt deposits on the compressor blading surface, which impact the aerodynamics, and therefore the overall gas turbine performance.

The paper focuses on compressor and gas turbine performance parameters for degraded engines with varying ambient conditions. Furthermore the internal aerodynamic loading distribution and possible operational limitations (e.g. surge) for certain degradation rates and gas turbine operating conditions are assessed. For this purpose a simplified CFD model is applied, capable to simulate certain degradation rates for every individual compressor blade row. In addition a simplified gas turbine model is applied, to study the compressor degradation impact with its overall consequences.

Two basic case studies are shown. The 1st one is addressing filter clogging. The 2nd one takes compressor blading degradation into consideration while assessing the impact of different fouling distribution rates. The case studies are substantiated and supported by field measurements. Finally recommendations are given for possible activities to be taken by the end-users to ensure efficient and safe gas turbine operation.

INTRODUCTION

The incentive for gas turbine operators (end-users) to understand, minimize and control performance deteriorations of their plants has significantly increased over the last years. Decentralization and increasing fuel costs are seen as driver for this change. Furthermore, reliability, availability, maintenance intervals, and associated costs are getting more significant and become a focus. Engine degradation reduces possible commercial benefits and is meanwhile handled with strong attention.

Gas turbine degradation can be driven by multiple effects. On one side by turbine blading degradation, which is most significantly seen with oil operation. On the other side the intake filter systems and the compressor have been identified often to be significant root causes (see e.g. Meher-Homji et al., 2009 [1]). The effect of filter degradation on the overall gas turbine behavior is e.g. intensively discussed by Kurz et al., 2009 [2]. The different degradation effects for the compressor blading can be identified as

- Fouling/deposits stick-up on surfaces
  (only considered mechanism inside this paper)
- Clearance increase
  (e.g. by strong rubbing)
- Mechanical blading deformation
  (e.g. by foreign object damages)

All these effects have an aerodynamic impact, which results in a blading efficiency reduction, and often also a reduction of the flow turning and the maximum possible incoming flow angle variation capabilities.

Fouling is understood as particles sticking on the blading surface and therefore influencing its aerodynamic behavior. These particles sticking on the surface often result from two causes. Firstly from environmental conditions as heavy industries and/or chemical industries in the neighborhood. These particles are often so small that they can pass the filter system and get to the compressor front stages. The compressor front stages dominantly define the incoming mass flow and small impacts on their surface start to reduce the mass flow capability already. Humidity supports the effect that particle stick on the surfaces. Furthermore, stand still periods, when the gas turbine is shut off, without closing the filter system properly will lead to slight corrosion, which does not affect the mechanical integrity but the aerodynamics. This type of corrosion is a significant part of degradation mainly for the mid and rear stages. Examples and background information is given e.g. with Kurz et al., 2011 [3] or Kappis et al., 2012 [4].

Monitoring activities on the overall gas turbine deterioration are documented in various papers e.g. Schneider et al., 2009 [5] or Hepperle et al., 2011 [6]. These papers mainly focus on the overall gas turbine degradation and are partly proposing consequential mitigation measures as on-line washing, but do not assess the loading shifts inside the different components in detail. Other papers study the effect of e.g. the compressor degradation for itself only, whereby Meher-Homji et al., 2009 [1] clearly states that it has to be assessed in combination with the overall gas turbine. This can be easily explained while looking for the effect of compressor fouling. As a consequence the compressor efficiency and the inlet mass flow are reduced. Keeping a constant firing temperature it consequently decreases as well the pressure ratio of the compressor. Therefore it is clearly seen that the compressor degradation acts even three times on the gas turbine performance, on the one hand side with the reduced compressor efficiency, furthermore with a reduced mass flow and additionally with a reduced gas turbine pressure ratio. All three parameter significantly impact the overall gas turbine performance. As a consequence the following scheme was developed for being capable to analyze compressor degradation effects on internal compressor load shifts and overall gas turbine behavior:

- An aerodynamic tool with the capability to simulate degradation impact for individual blade rows
- A simplified performance tool to simulate the overall gas turbine behavior (defining the back pressure for the compressor)

PRINCIPLE OF COMPRESSOR DEGRADATION

A typical engine degradation behavior is shown in Fig. 1. Assuming the performance parameter is interpreted as the engine power it is seen that over time the power output decreases for constant operational engine conditions. Different washing and cleaning activities can support to regain a significant part of the performance loss, however, a certain amount of permanent performance degradation will continuously build up. Therefore it is important for the end-users to understand these degradation effects and possible mitigations, but as well at which point in time the gas turbine operation becomes critical and mitigation measures have to be taken.

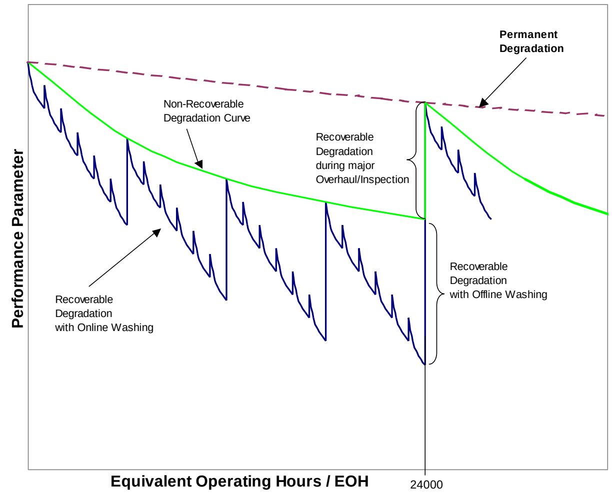

Fig. 1: Typical gas turbine degradation chart (Hepperle et al., 2011 [6])

COMPRESSOR FLOWPATH & BLADING CLEANING

The main mitigation measures to reduce gas turbine degradation are:

- On-line washing, during standard gas turbine operation
- Off-line washing, where the engine is disconnected from the grid
- Manual hand cleaning, during a standard outage/inspection

as intensively and detailed discussed by e.g. Schneider et al., 2009 [5].

Compressor on-line washing as a first mitigation measure is normally done during gas turbine base load operation, where the variable inlet guide vanes (VIGVs) are close to a fully open position. A corresponding compressor on-line washing system is usually installed at the air inlet of the gas turbine, in the intake system facing the compressor front blading. Generally, demineralized water is injected through this configuration of small nozzles. In addition to pure demineralized water a detergent can be used to increase the cleaning effect of compressor on-line washing. A small drawback is the full evaporation of the washing water after a certain number of compressor stages. Consequently the cleaning effect for the following blade rows will disappear. Therefore the effect of on-line washing is limited to the VIGVs and the front stages of a compressor only, as discussed by Demircioglu, 2008 [7], Bromley et. al., 2004 [8] and Stalder et al., 1994 [9]. The front stages cleaning ensure that nearly the original mass flow is re-achieved by the engine, again. Unfortunately, the on-line washing has the negative impact of dirt from the front stages to be transported to the mid and rear stages, where it can stick to these blade rows, as the washing water evaporates in the mid and rear stages due to the high temperatures. This reduces the efficiency and the working ranges of these blade rows and has to be carefully observed to avoid critical operating conditions, which could in a worst-case even cause surge phenomena (see Kappis et al., 2012 [4]). Compressor on-line washing can diminish the degree of compressor fouling only to a certain amount. Therefore, from a performance recovery point of view, it is recommended to carry out a compressor off-line washing from time to time, because this shows a higher effectiveness for performance recovery.

A ***compressor off-line washing*** procedure usually consists of several steps. Firstly the engine has to be shut down and cooled. Afterwards the gas turbine compressor is flushed with demineralized water, where as well detergents can be used. A gas turbine driving system is used, e.g. the start-up system, to bring the gas turbine to 20-50 [%] of its nominal speed and afterwards allowed to coast to a stop. This flushing and speeding up procedure may be repeated several times. Finally several rinse-cycles are applied until the water that pours out of the compressor through the opened blow-off valves and the drainage system does not indicate any more signs of contamination. The whole process is intensively discussed by e.g. Boyce et al., 2007 [10] and Demircioglu, 2008 [7]. The drawback of this procedure is that the engine has to be disconnected from the grid and cooled down, creating an interruption in production. Therefore, compressor off-line washing is at least recommended after scheduled inspections to minimize non-availability of the gas turbine.

Abrasive Cleaning respectively the use of abrasive materials for compressor cleaning, e.g. the injection of rice or walnut-shells into the compressor, is suspended in most plants (see Schneider et al., 2009 [5]). Reason for it is that the erosion effects on the blading can be too high and there is the danger that these non-liquid materials collect in the gas turbine and clog the passages of the secondary air system. Particles may enter seals and the lubrication system, or they may damage the coating of the compressor blades as discussed by e.g. Boyce et al., 2007 [10] and Demircioglu, 2008 [7]. Nonetheless, abrasive compressor cleaning may be performed on a removed rotor during a major overhaul.

Compressor Hand Cleaning is very effective for removing particles sticking to the blade surface, but has the drawback that it is very time consuming and requires the shutdown and cooling of the gas turbine beforehand (see e.g. Demircioglu, 2008 [7]). Usually brushes and a detergent are used for the hand cleaning procedure. The procedure should be applied for every major outage where the gas turbine is opened. Unfortunately at these outages the rotor is not always removed, which gives only limited access to the complete blading, but at least the rotor can be turned and all rotating blade rows properly cleaned. An additional hand cleaning of the VIGVs and the front blades can be executed at any shutdown. It is recommended to execute this limited hand cleaning of the accessible front stages in combination with a compressor off-line washing whenever possible.

COMPRESSOR AND GAS TURBINE MODELING

From an aerodynamic perspective, degradation changes the loss and flow turning capabilities of effected blade rows. The loss change is mainly driven by a higher surface roughness leading to an increase in friction. For the flow turning discussion, Fig. 2 is used to explain the different flow angles to be considered and their relationships.

With reference to the metal inlet angle, the incoming flow angle, which yields minimum profile losses, is called the nominal incidence angle (i*). Flow angles outside this nominal incidence flow angle are named as working incidence flow angles (i) and are limited by the stall and the choke flow angle.

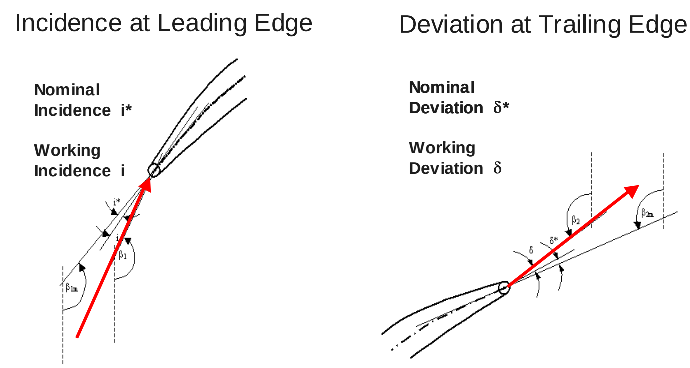

Fig. 2: Basic flow angles along a profile section

These two flow angles are physical limits, which define the minimum (stall) and the maximum (choke) mass flow passing through the blading passage. At the blading outlet a similar nominal deviation angle (δ*) exists for reference design conditions, which changes for varying inlet conditions with an add-on deviation angle (Δδ). The inlet and the exit flow angle together define finally the flow turning and therefore also the pressure rise of an individual profile section.

The fouling effect along a blading section increases the profile boundary layer thickness due to increased surface roughness. Consequently the blading losses increase and as well the deviation angle, which results in less turning of the flow. Furthermore the blading section will become more sensitive to incidence angles. This is driven by the fact that the original stall and choke incidence angles can’t be further provided, due to the weakened profile boundary layer behavior.

For numerical simulation so called loss and deviation models are used to predict the flow around a profile section (see e.g. Horlock, 1975 [11] or Cumpsty, 1989 [12]). These aerodynamic loss and deviation characteristics have to be adapted to the impact of fouling. The model developed consists mainly on a reference value increase and another parameter reflecting the reduced operating range capabilities, which means stall and choke conditions. The theory is more intensively described in Kappis et al., 2012 [4] and substantiated by Morini et al., 2010 [13] and Zaba, 1981 [14]. The basic principles for the loss model are shown in Fig. 3.

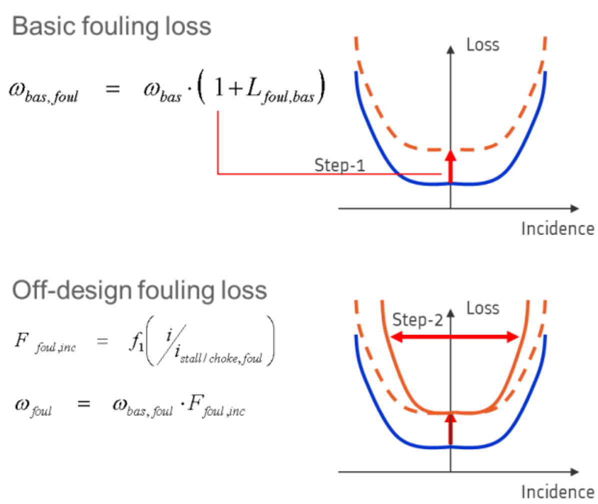

Fig. 3: Two step approach for degradation modeling

A similar approach is used for the deviation correction namely:

Basic fouling deviation

$$\delta_{bas,foul} = \delta_{bas} \cdot \left(1 + D_{foul,bas}\right)$$

Off-design fouling deviation

$$i_{stall/choke,foul} = {}^{i_{stall/choke}}\!/\!\left(1 + D_{foul,inc}\right)$$

$$E_{foul,inc} = f_2\left({}^{i}\!/_{i_{stall/choke,foul}} \right)$$

$$\delta_{foul} = \delta_{bas,foul} + \delta_{inc} \cdot E_{foul,inc}$$

To simulate different gas turbine operating points with and without degradation effects the overall gas turbine behavior needs to be predicted. For this purpose a simplified approach by Stodola as e.g. reference by Traupel , 1982 [15] is used, assuming that the reduced turbine inlet mass flow stays constant.

$$m_{Turb,red} = const. = \dot{m}_{Turb,inl} \cdot \frac{\sqrt{R_{Turb,inl} \cdot T_{tot,Turb,inl}}}{p_{tot,Turb,inl}}$$

While neglecting changes of relative extraction mass flows, relative fuel mass flows and pressure losses for the combustion system, a simplified operating line equation for constant firing temperature can be derived in the form:

$$\frac{\Pi_{comp}}{\Pi_{comp,ref}} = \frac{\dot{m}_{comp}}{\dot{m}_{comp,ref}} \cdot \sqrt{\frac{R_{Comp,inl,ref} \cdot T_{amb,ref}}{R_{Comp,inl} \cdot T_{amb}}}$$

This simplified operating line equation shows that for constant ambient conditions the relative compressor inlet mass flow change goes linear with the combined compressor pressure ratio change.

THE IMPACT OF INTAKE FILTER DEGRADATION

The overall compressor performance behavior is best seen inside the compressor map. This map has to be drawn with the normalized aerodynamic similarity parameters to be valid for changed ambient conditions. The aerodynamic similarity parameters are:

Aerodynamic speed

$$n^*_{aer} = \frac{n}{n_{ref}} \cdot \sqrt{\frac{R_{ref} \cdot T_{t,inl,ref}}{R \cdot T_{t,inl}}}$$

Reduced inlet mass flow

$$\dot{m}^*_{aer} = \frac{\dot{m}}{\dot{m}_{ref}} \cdot \frac{p_{tot,inl,ref}}{p_{tot,inl}} \cdot \sqrt{\frac{R \cdot T_{tot,inl}}{R_{ref} \cdot T_{tot,inl,ref}}}$$

Reduced blading pressure ratio

$$\Pi^*_{aer} = \frac{\Pi}{\Pi_{ref}}$$

A typical corresponding compressor map is shown in Fig.4, describing the reduced mass flow behavior as a function of the aerodynamic speed and the reduced blading pressure ratio.

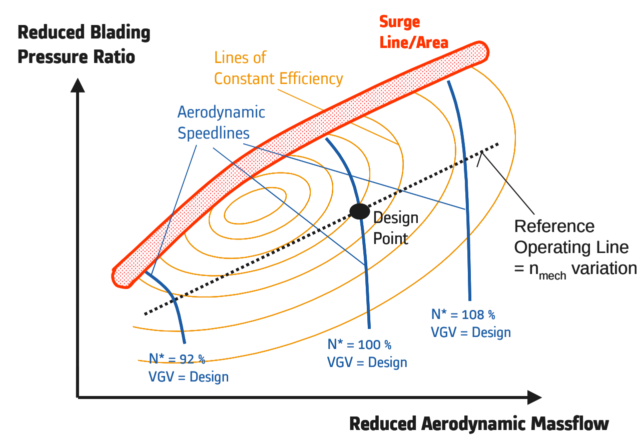

Fig. 4: Typical compressor map for const. VIGV position

Assessing the reference design point only and assuming that ambient temperatures and the mechanical speed do not change the aerodynamic speed stays constant. Due to the steepness of the speed lines for multistage compressors the reduced aerodynamic mass flow changes neither. Therefore it can directly be concluded that the mass flow changes linearly with the pressure change at the compressor inlet.

$$\frac{\dot{m}_{Comp}}{\dot{m}_{Comp,ref}} = \frac{p_{tot,inl}}{p_{tot,inl,ref}}$$

Based on the given simplified gas turbine operating line equation (for constant fuel composition and firing temperature) the overall compressor ratio turns down to:

$$\frac{\Pi_{comp}}{\Pi_{comp,ref}} = \frac{p_{tot,inl}}{p_{tot,inl,ref}}$$

Taking into consideration some typical field pressure losses of an intake system, which can start at 10 [mbar] and easily degrade to 20 [mbar] pressure loss the following relative changes can be calculated assuming ISO conditions as a reference:

$$\frac{p_{tot,inl}}{p_{tot,inl,ref}} = \frac{1{,}01330 - 0{,}02}{1{,}01330 - 0{,}01} \approx 0{,}99$$

This implies correspondingly a mass flow and a pressure ratio reduction of about 1 [%] can be expected.

Studying this effect by physical simulation and assuming that the intake loss is driven by 20 [%] on static and by 80 [%] on dynamic losses, Fig. 5 shows a typical gas turbine power degradation schematic for such a case with varying ambient temperatures. Thereby for this figure and following ones deterioration data are calculated by following equation:

$$Change = \frac{F_{actual} - F_{ref}}{F_{ref}}.100$$

This approach ensures the understanding of the change with respect to a corresponding reference value.

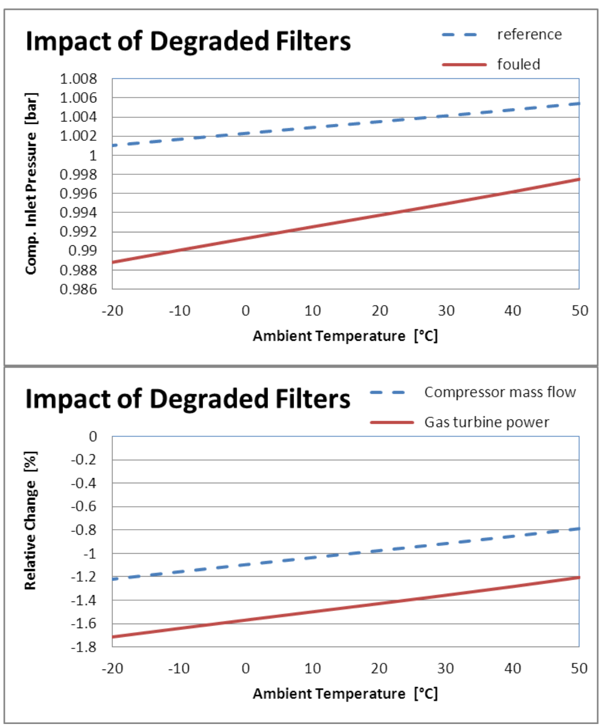

Fig. 5: Typical power reduction curve due to filter clogging

With a compressor operating line pressure ratio reduction of about 1 [%] no critical loading re-distribution inside the compressor blading is expected. More important issues for safe gas turbine operation is a clear monitoring of the under pressure at blading inlet. Reason is that very often stationary gas turbines have an implosion door, which opens if the under pressure is too strong. In this case un-filtered air can pass to the compressor blading and fouling, as well as foreign object damage can bring severe risk to the gas turbine.

To resolve the filter degradation, most of the time the different filter patches have to be exchanged. Nevertheless due to the architecture of the filter housing, not all filter patches are similarly loaded and therefore degrade differently. Sometimes the filter exchange in the main flow path (usually lower part of the filter house) can bring already a significant improvement.

THEORETICAL PRE-STUDIES ON THE IMPACT OF COMPRESSOR BLADING DEGRADATION/FOULING

From an aerodynamic perspective, degradation changes the blading turning and efficiency. To study these effects with the previously described model a 2 step approach was chosen.

In a 1st step an analytical analyses is done to understand the impact of different fouling distributions on the compressor aerodynamic re-matching and the overall compressor and associated gas turbine behavior. Furthermore this analyses is used to confirm basic trends as intensively discussed in literature (see e.g. Meher-Homji et al., 2009 [1], Kurz et al., 2011 [3] and Schneider et al., 2009 [5]).

Taking as a reference a typical 18 stage stationary gas turbine compressor (see Kappis et al., 2012 [4]) and applying a fouling factor distribution as shown in Fig. 6 the following compressor and gas turbine trends, shown in Fig. 7, can be observed as a function of changed ambient temperatures.

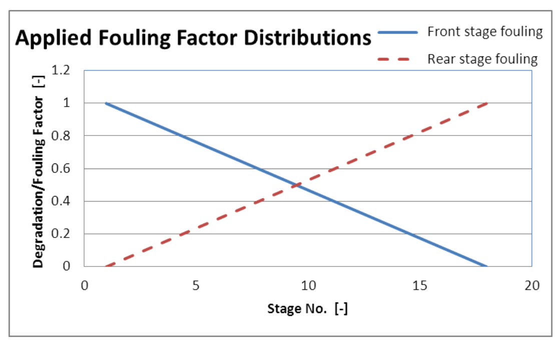

Fig. 6: Applied fouling factor distributions for basic trend studies

The calculations clearly show that front stage fouling reduces the compressor inlet mass flow to a much higher degree as the rear stage fouling. This clearly confirms trends from the literature. The mass flow reduction behavior itself is explained by the fact, that mainly the front stages define the inlet mass flow. For multistage axial compressors the steep speed line characteristics make the mass flow even independent from the back pressure and therefore no change happens. Furthermore it is seen that for high ambient temperatures or low aerodynamic speeds the mass flow reduction effect increases. This is driven by the point that for low aerodynamic speeds the front stages see a higher positive incidence angle and therefore have to work harder. This makes front stage fouling more critical for these conditions. As well this trend is confirmed by the applied model. For the overall gas turbine modulation the consequences of the different fouling factor distributions on the pressure ratio and the gas turbine power are physically correct. Therefore the model can be assumed to predict correct trends for the chosen 18 stage stationary compressor and the overall associated gas turbine.

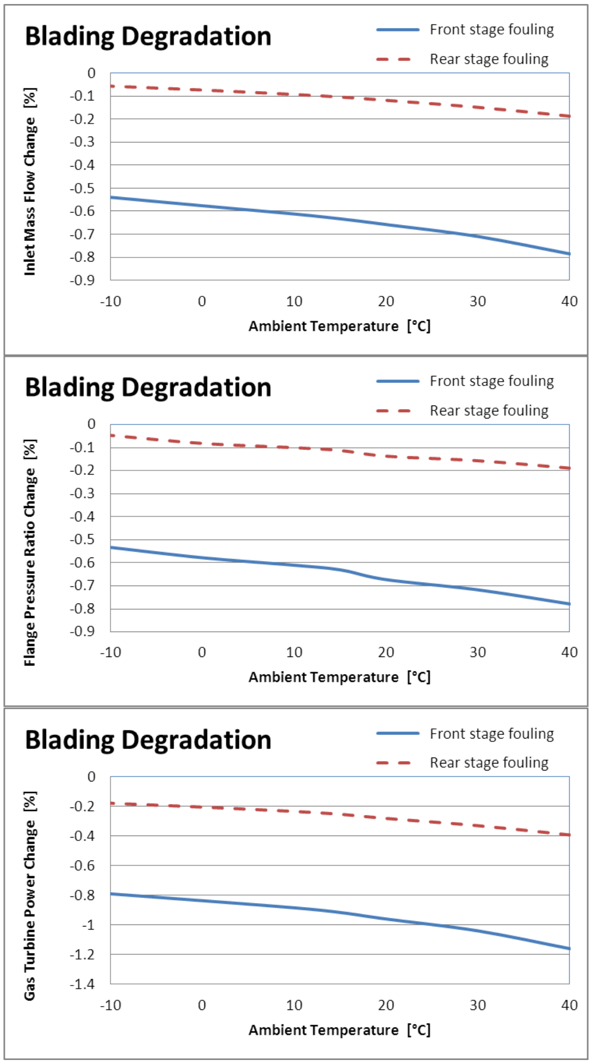

Fig. 7: Compressor and gas turbine performances for different basic fouling factor distributions

Another parameter of interest for assessment is the impact of the blading type, namely if fouling on the rotating blades or the none-rotating vanes have a differently strong impact on the gas turbine behavior. Due to the stronger impact of the front stage fouling characteristic this distribution characteristic was applied independently to the rotating blades and the none-rotating vanes only. Main results, as shown in Fig. 8, clearly indicate that the impact of fouling on the rotating blades is nearly twice than for the none-rotating vanes. One physical explanation for this effect is that the blades due to working in the rotating frame generate an absolute pressure rise and mainly the front stages with their fairly high inlet Mach numbers generate as well a static pressure rise by going subsonic at the blading outlet. The none-rotating vanes can only build up a static pressure rise. This interaction is strongly design dependent and typical reaction numbers confirm this different loading of the front stages (usually quite high in the front stages and decreasing to a nearly identical loading for blades and vanes in the rear stages).

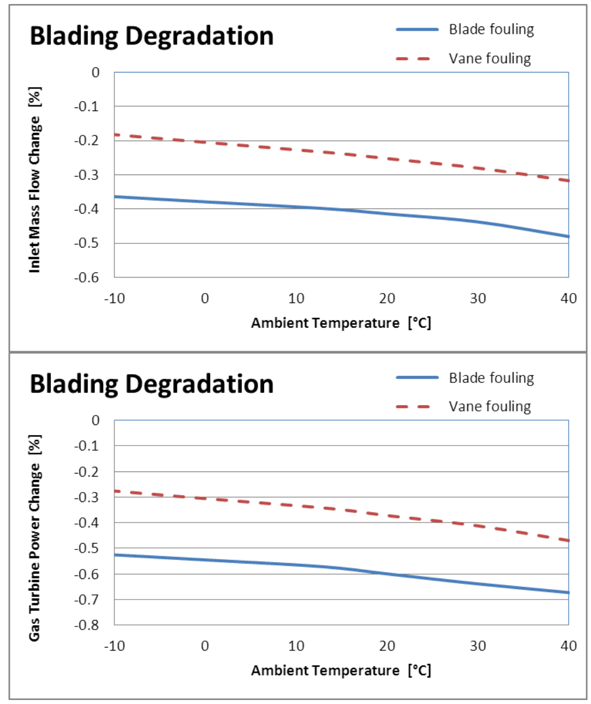

Fig. 8: Compressor and gas turbine performances for fouling simulation on blades or vanes only

As a consequence of this theoretical studies it is clearly seen that the fouling factor distribution over the compressor length, but as well the variation for rotating blades and none-rotating vanes has to be clearly considered for a realistic degradation simulation.

THE IMPACT OF COMPRESSOR BLADING DEGRADATION/FOULING

Field observations documented e.g. by Tarabrin et al., 1998 [16] for a two-shaft Frame 5 engine clearly indicate that mainly the front stages of the compressor are effected by fouling as shown in Fig. 8. Inside these graphs it is as well quite obvious, that a difference between rotating blades and stationary vanes has to be taken into consideration. This different blading type behavior is expected to be associated with the acting centrifugal force inside the rotating system, which allows only less deposits to stick to the blading surface. As a consequence for real case simulation the fouling factor distribution has to be adapted accordingly taking significantly higher front stage fouling and a significant difference between rotating blades and none-rotating vanes into consideration.

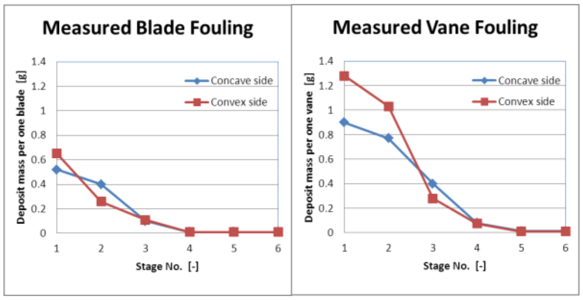

Fig. 9: Measured distribution of deposits on a compressor of a two shaft Frame 5 gas turbine (Tarabrin et al., 1998 [16])

Furthermore additional field experience on the overall gas turbine degradation is taken into consideration as e.g. documented by Schneider et al., 2009 [5] and shown in Fig. 10. Different engines at a same site have been monitored. The power plant (named PB in following) consists of three GT13E2 gas turbines in a combined-cycle multi-shaft arrangement. This reference is taken, because it is one of the rare examples where overall engine power and compressor efficiencies are assessed over a long term period.

The environmental conditions of the monitored site can be summarized as follows:

- Industrialized area
- High air humidity
- High particles concentration < 10 µm
- Maritime conditions

Performance figures are monitored over time, but as well for different washing intervals. In detail washing was applied as follows:

- **PB1**: Twice a week compressor on-line washing with a detergent
- **PB2**: Daily compressor on-line washing with water
- **PB3**: No compressor online washing

Analyzing the gas turbine power data of Fig. 10 and taking the previously discussed learnings of the compressor inlet mass flow change due to front stage fouling into consideration, it can be clearly seen that the on-line washing is quite efficient for the front stages. This is as well confirmed by papers e.g. from Schneider et al., 2009 [5] and Kurz et al., 2009 [2]. Furthermore considering the long term trend, some performance deterioration can be observed, which seems to be connected with a continuous deterioration of the whole compressor.

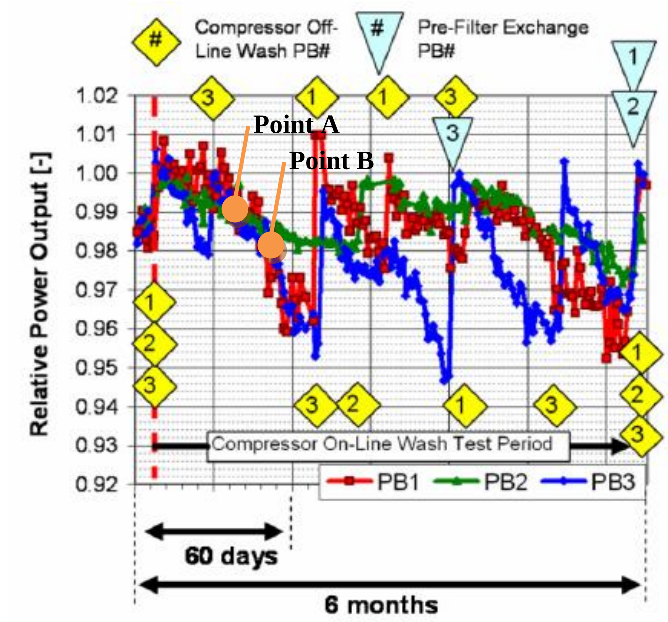

Fig. 10: Relative gas turbine power output comparison (Schneider et al., 2009 [5])

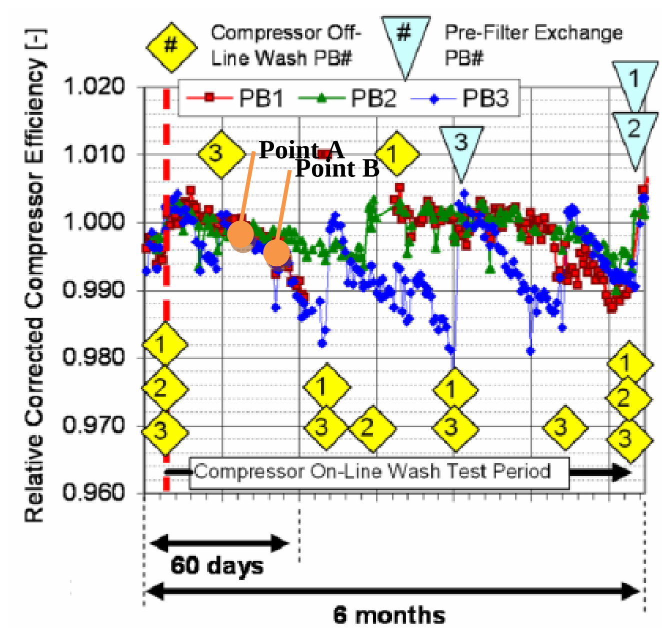

Fig. 11: Relative compressor efficiency comparison (Schneider et al., 2009 [5])

For realistic compressor degradation simulation , and while assessing the compressor behavior in between two off-line washing intervals without on-line washing (PB3), a model has been developed on the change of the fouling factors to be applied for the numerical simulation, which is shown in Fig. 12. The difference between rotating blades and none-rotating vanes was taken into consideration by just multiplying the fouling factor curve with a factor of 1.8 for the short term fouling and by 1.6 for the mid term fouling. This different application was chosen, because based on the long term data shown in Fig. 10

& 11 a kind of fouling saturation starts after a certain time or fouling rate.

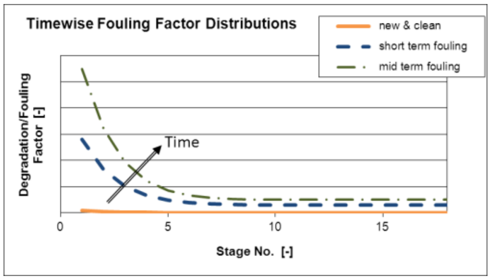

Fig. 12: Fouling factor distributions inside two off-line washing intervals

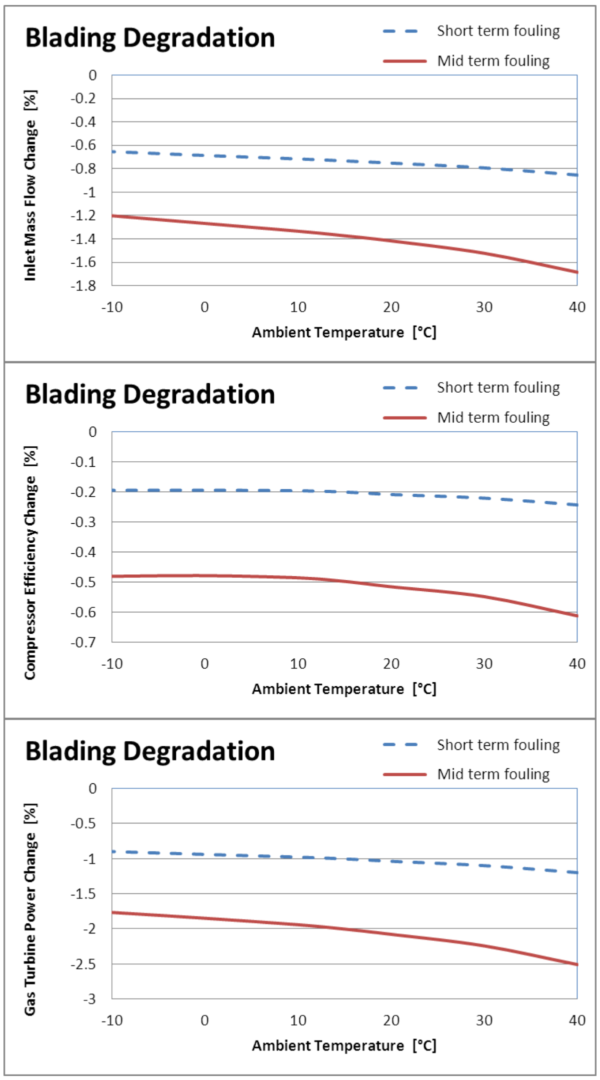

Fig. 13: Calculated compressor and gas turbine performance figures in between two off-line washes

The fouling factor distributions have to be applied to the individual blade row simulations for the loss and the flow exit angle (deviation) calculation models. The loss model mainly acts on the compressor efficiency, whereby the deviation model dominantly defines the inlet mass flow. To distinguish between these two separated fouling factor distributions the model has been calibrated for ISO conditions and certain associated performance deterioration figures as graphically shown in Fig. 10 & 11 (Point A & B). The calibration numbers applied are:

- Point A / short term fouling
  1 [%] power loss, 0.2 [%] compressor efficiency loss
- Point B / mid term fouling
  2 [%] power loss, 0.5 [%] compressor efficiency loss

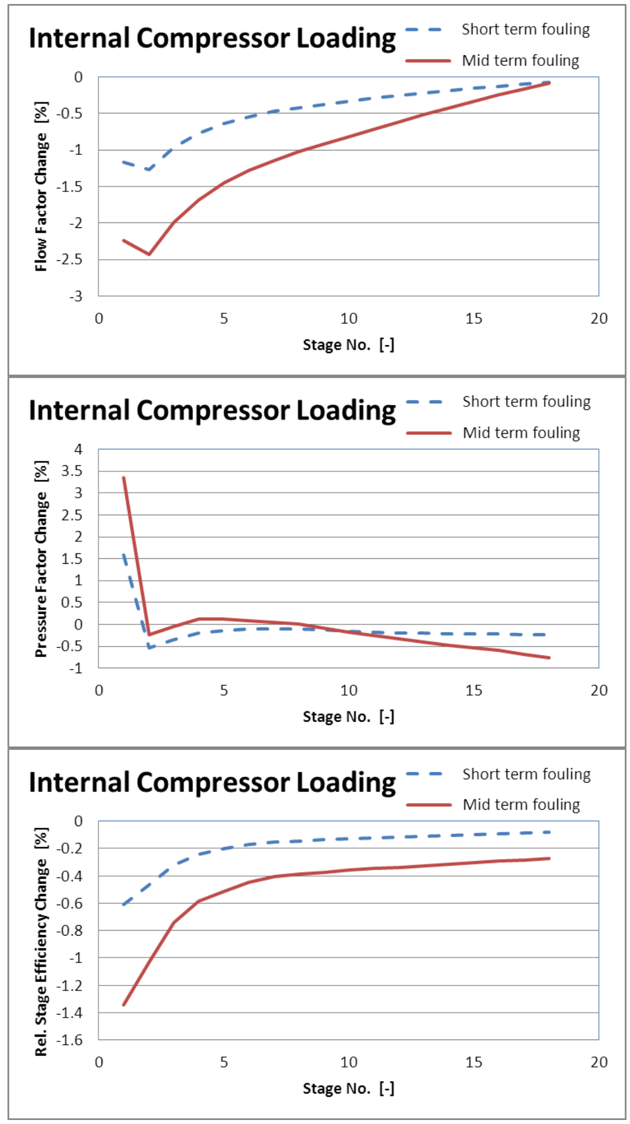

Fig. 14: Stage loading figures for fouling conditions at 30 [°C] ambient temperature

After all this calibration work the models have been applied for the performance degradation calculations for different ambient conditions and changed fouling rates. The resulting performance figures of the compressor and the gas turbine are shown in Fig. 13. Detailed internal compressor loading figures for an operating point of 30 [°C] ambient conditions are shown in Fig. 14 for the compressor stage data and in Fig. 15 for the data of the rotating blades.

Based on the calculation results the impact of fouling with respect to ambient temperature change for the overall compressor and gas turbine (Fig. 13) can be discussed as follows:

- Compressor inlet mass flow changes basically as a function of the fouling rate. The effect is less severe for cold ambient conditions, whereby it can become quite significant for high ambient temperatures
- Compressor efficiency changes basically as a function of the fouling rate. For low ambient temperatures it stays nearly constant , whereby it can become quite significant for high ambient temperatures
- Gas turbine power changes basically together with the compressor mass flow rate and the compressor efficiency. The reduced mass flow rate changes pressure ratio additionally. As a rough guess it can be seen that the gas turbine power reduction is approximately 40 [%] higher as the compressor mass flow reduction.

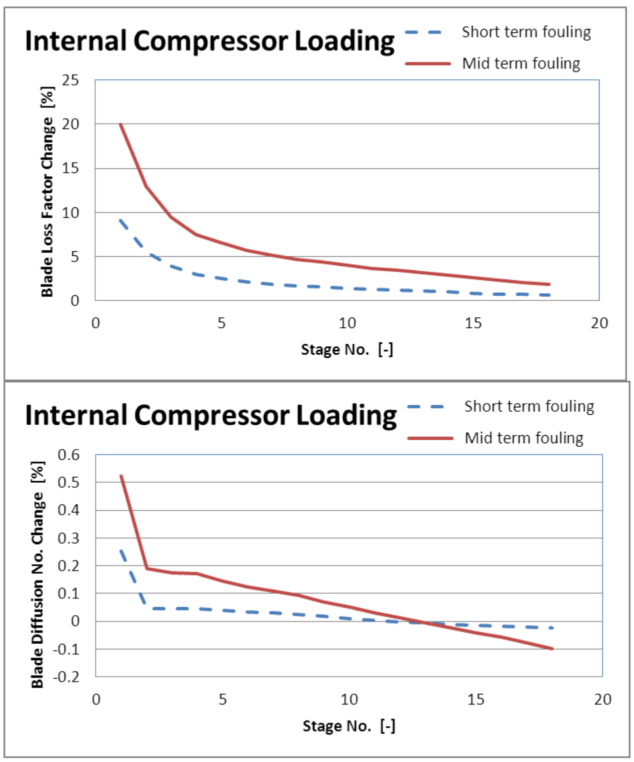

Fig. 15: Rotating blading loading figures for fouling conditions at 30 [°C] ambient temperature

To discuss detailed stage loading figures, the 30 [°C] ambient temperature operating point at was chosen. Criteria for this decision was that already a quite significant decrease in overall performance data and trend gradient can be observed. Thereby the following variable definitions have been applied:

Flow factor:

$$\varphi = \frac{c_{ax,inl,stage}}{u_{inl,stage}}$$

Pressure factor:

$$\Psi = \frac{\Delta h_{tot,abs,stage}}{u^2_{inl,stage}}$$

As a general observation on the stage data (Fig. 14) a trend or strong gradient change can be observed in some characteristics at the second stage. This is to a huge extent linked with the strong fouling behavior of the first vane, which gives quite disturbed inlet conditions for the second stage. The loading figures of the first stage are getting dominantly defined by the overall mass flow rate. Therefore the front stages need special consideration by the end-user because they might drive the engine to a surge event. In general the following discussion applies:

- The flow factor significantly decreases for the front stages and afterwards recovers nearly to the original value. This is linked with the strong mass flow decrease, which mainly acts on the front stages.
- The pressure factor shows a quite intensive increase for the first stage and stays afterwards nearly around original values. This additionally confirms that the front stages are with degradation closer to the surge limitations.
- The efficiency curves show a strong decrease in the front stages and in the mid and rear stages a nearly constant reduction, based on given fouling rates.

Breaking even further down to understand blading data Fig. 15 was introduced, showing for an operation point of 30 [°C] ambient temperature and different fouling rates the data of the rotating blading only. Thereby the following variable definitions have been applied:

Loss Factor

$$\omega = \frac{\Delta p_{tot,row}}{p_{tot,inl} - p_{inl}}$$

Diffusion Number

$$D = \frac{w_{suc,\max} - w_{exit}}{w_{inlet}}$$

$$\text{with} \quad \frac{w_{suc,\max}}{w_{inlet}} \approx 1 + \frac{\Delta w_{\text{tang}}}{\sigma \cdot w_{inlet}}$$

Assessing the parameter for the rotating blading it can be seen:

- The blading losses confirm the trend of the stage efficiencies, being quite high in the front of the compressor due to the strong re-matching and falling down to reasonable figures in the mid and rear stages of the compressor just depending on the fouling rate.

- The blading diffusion numbers are showing a clear increase for the front stages, whereby for mid and rear blades they stay at a low level. This confirms the possible criticality of the front stages for surge. Nevertheless the change is in absolute figures quite small and has to be interpreted very carefully. The used definition for the diffusion numbers don't reflect the increased surface roughness and the weakened boundary layer, which is seen as the main contributor of fouling. Therefore the absolute numbers under predict reality.

All parameter assessments executed indicate a clear mass flow and power reduction due to fouling but as well an increased risk for front stage surge under fouled conditions and high ambient temperatures.

CONCLUSIONS

An empirical model was developed to simulate the degradation effects for clogged filters and fouled compressor blading.

In a first case study clogged filters have been assessed and consequences on the gas turbine power reduction have been calculated for different ambient conditions. The internal compressor re-matching was assessed as negligible but reasonable filter patch exchanges from time to time are recommended.

In a second case study compressor blading fouling has been assessed. Simulation models have been applied and calibrated based on field observations. Main effects hereby have been strong front stage fouling and a significant higher degradation rate for none-rotating vanes as for rotating blades. Application of the model for different ambient conditions and fouling rates delivered overall gas turbine and compressor data, but as well stage and even rotating blading loading figures for an operating point of 30 [°C] ambient temperatures. Inside these analyzed data the front stages have been identified to have the risk for getting critical for surge. Mitigation would be frequent compressor on-line washing, which mainly cleans the front stages. Unfortunately erosion of the front stages can occur, which has to be carefully observed e.g. by boroscope inspection. Nevertheless, from a performance point of view, off-line washing including hand cleaning of the accessible front stages is recommended for degraded engines.

As a general summary it has to be concluded that fouling rates and even the compressor design has a significant impact on gas turbine degradation. Fouling rates are very site and even engine specific due to local ambient and applied operating conditions. The dependency of the compressor design is strongly linked with the flow turning trends and capabilities mainly of the front stages. Therefore models to predict degradation have to take all these effects into consideration.

NOMENCLATURE

EOH	= Equivalent operating hours	(-)
VIGV	= Variable inlet guide vane	(-)
c	= Velocity in absolute frame	(m/s)
F	= Arbitrary factor	(-)
h	= Specific enthalpy	(J/kg)
i	= Flow incidence angle	(deg)
m	= Mass flow	(kg/s)
n	= Rotor shaft speed	(rpm)
p	= Pressure	(N/m^2)
R	= Specific gas constant	(J/kg/K)
T	= Temperature	(K)
u	= Tangential rotor speed	(m/s)
w	= Velocity in relative frame	(m/s)
β	= Flow angle	(deg)
δ	= Flow deviation angle	(deg)
Δ	= Value difference	(-)
φ	= Flow factor	(-)
Ψ	= Pressure factor	(-)
Π	= Pressure ratio	(-)
σ	= Solidity (chord/pitch)	(-)
ω	= Total pressure loss factor	(-)

Indexes:

abs	= Absolute frame of rotation
amb	= Ambient value
inl	= Inlet
ref	= Reference
stage	= Compressor stage
suc	= Suction side
ax	= Axial
row	= Compressor blade row
tang	= Tangential/circumferential direction
tot	= Stagnation value
Comp	= Compressor
Turb	= Turbine

REFERENCES

[1] Meher-Homji, C. B., Chaker, M., Bromley, A.F., 2009, "The fouling of axial flow compressors – Causes, effects, susceptibility end sensitivity", ASME Paper GT2009-59239.

[2] Kurz, R., Brun, K., Wollie, M., 2009, "Degradation Effects on Industrial Gas Turbines", ASME J.Eng. Gas Turbines Power, 131.

[3] Kurz, R., Brun, K., 2011, "Fouling Mechanisms in Axial Compressors", ASME Paper GT2011-45012.

[4] Kappis W., Guidati G., 2012, "Detailed Compressor Degradation Effect Modelling for Single Blade Rows while Assessing its Local and Overall Consequences", ASME Paper No. GT2012-68475.

[5] Schneider E., Demircioglu S., Franco S., Therkorn D., 2009, “Analysis of Compressor On-Line Washing to Optimize Gas Turbine Power Plant Performance”, Proceedings of ASME Turbo Expo 2009, GT2009-59356, Orlando, Florida, USA, June 8-12.

[6] Hepperle N., Therkorn D., Schneider E., Staudacher S., 2011, “Assessment of Gasturbine and Combined Cycle Power Plant Performance Degradation“, ASME Paper No. GT2011-45375.

[7] Demircioglu, S., 2008, “Detail Analysis of Compressor On-line Wash Impact in the Gas Turbine Performance”, Diploma Thesis, FH Offenburg.

[8] Bromley, A. F., Meher-Homji, C. B., 2004, “Gain a Competitive Edge with a Better Understanding of GT compressor Fouling” , Washing. Comb. Cycl. J., 4th quarter, pp. 37-41.

[9] Stalder, J. P., van Oosten, P., 1994, “Compressor Washing Maintains Plant Performance and Reduces Cost of Energy Production”, Int. GT. Aeroeng. Con. Expo., 94-GT-436.

[10] Boyce M. P., Gonzales F., 2007, “A Study of On-line and Offline Turbine Washing to Optimize the Operation of a Gas Turbine”, Journal of Engineering for Gas Turbines and Power, Vol. 129, pp. 114-129.

[11] Horlock, J. H., 1975, “Axialkompressoren“ (Axial Compressors), Verlag G. Braun, Karlsruhe.

[12] Cumpsty, N. A., 1989, “Compressor Aerodynamics“, Verlag Longman Group UK Limited.

[13] Morini, M., Pinelli, M., Spina, P. R., Venturini, M., 2010, “Computational Fluid Dynamics Simulation of Fouling on Axial Compressor Stages”, ASME J. Eng. Gas Turbines Power, 132(7), 072401.

[14] Zaba T. , 1981, “Losses in Gas Turbines Due to Deposites on the Blading”, Brown, Boveri & Cie, Publication No. CH-T 113433E.

[15] Traupel, W. , 1982, “Thermische Turbomaschinen” (Thermal Turbo Engines), Bd. II, Kap. 11.4, Dritte Aufl., Springer Verlag.

[16] Tarabrin, A. P., Bodrov, A. I., Schurovsky, V. A. and Stalder, J-P., 1998, “Influence of Axial Compressor Fouling on Gas Turbine Unit Performance Based on Different Schemes and with Different Initial Parameters”, ASME International Gas Turbine and Aeroengine Congress, Stockholm Sweden, June 2-5, ASME paper No. 98-GT-416, 1998.