Int.J. Applied Thermodynamics,
Vol.5, (No.1), pp.37-47, March-2002

ISSN 1301-9724

Variations in Gas-Turbine Blade Life and Cost due to Compressor Fouling – A Thermoeconomic Approach

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Abstract

The connection between gas turbine compressor fouling and expander blade lifetime is studied in a simplified manner for a gas turbine operating in a combined cycle with a constant power output. It is shown how blade materials and compressor fouling rate affect the blade lifetime, and with this background, based on a thermoeconomic approach, the economic aspects of compressor washing intervals and the possibility to find an economic optimum are studied. It is also discussed how it should be possible to employ on-line gas turbine monitoring based on artificial neural networks (ANN) in combination with a database containing blade life behavior in order to improve the blade life management strategy for an optimization of power plant life profitability.

Key words: compressor fouling modelling, compressor washing, gas turbine blade life, artificial neural networks

1. Introduction

Increasing the hot gas temperature in a gas turbine based power plant leads to an increase of the power output and, in general, also the thermal efficiency η_{th} of the plant increases. The high gas temperatures of modern gas turbines, however, are significantly above permissible materials temperatures, and hot parts, such as, turbine blades, vanes, disks and the combustion chamber must be cooled, usually with air bled off from the compressor. In modern gas turbines that are in operation today, with firing temperatures of over approximately 1530 K (2300°F), the requirements on the first stage rotating blades and also on the first stage vanes are rather extreme, which calls for expensive materials and complicated internal configurations, which, in turn, require complicated manufacturing processes. Altogether, this leads to significant costs for blades and vanes in modern gas turbines.

Making investments in new power plants is a risky business on the deregulated electricity market, and owners must take care of their investments over the entire life cycle. Ownership costs for the new high-temperature gas turbines have turned out to be very sensitive to operations and maintenance costs, compared to old technologies (Stambler, 2000), and first-stage blades and vanes are particularly sensitive and expensive, together with the burners in the combustion chamber.

The new and clean condition, which is given by gas turbine manufacturers in

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publications like the Turbomachinery Handbook, often constitutes the base for thermodynamic cycle studies, the work presented in Jordal et al. (2000) being no exception. However, gas turbine components begin degrading as soon as the turbine is put in operation. Examples of causes for gas turbine performance deterioration can be found in Razak and Carlyle (2000), including the well-known fact that compressor fouling is the most common cause of performance deterioration, and also one of the most easy to restore through washing of the compressor.

The deregulated electricity market has created new actors, in particular independent power producers (IPPs), and new behaviors for the production and sale of electric power. For instance, IPPs can sell electricity on a contract either as a certain number of kWhs per year or as a certain number of kilowatts (predefined power output). In the case with a contract on predefined output, the compressor fouling and decreased performance mean that the gas turbine firing temperature must be increased in order to maintain the contracted power output. An increase of hot gas temperatures will lead to increased blade temperatures, which, in turn, affect the blade life in a negative manner. In this case, care must be taken not to wash the compressor infrequently, since this will have a detrimental impact on blade lifetime. At the same time, compressor washing in itself brings a cost, and yet another cost will be added if the gas turbine must be stopped during normal operation hours when the IPP must pay a fine for contracted but undelivered power or buy electricity on the spot market to deliver what it cannot produce.

The present paper is an attempt to quantify the known, but little described, link between compressor washing and blade lifetime in a gas turbine. Also, the impact that compressor washing intervals may have on the cost of produced power is studied, based on a thermoeconomic approach. The work is completed with a discussion on artificial neural networks (ANN), which should be an economic and efficient way of on-line condition monitoring, and a useful tool for optimizing compressor washing intervals and other gas turbine maintenance measures.

2. Power Plant Model

A major part of the gas turbine based power plants built today are of the combined cycle type, since combined-cycle power plants provide a combination of high thermal efficiency, reliable technology and short construction time.

The power plant model used for the studies is a single-shaft dual-pressure combined cycle of 62.1MW and a thermal efficiency (including generator losses) of 52.5%. Basically, the same thermodynamic model was used in previous works (Jordal et al., 2000, Jordal and Torisson, 2000), but for a slightly smaller gas turbine. A control volume model over the entire plant is shown in *Figure 1*, and the performance of the power plant at full load in new and clean condition (ISO ambient conditions) is given in TABLE I.

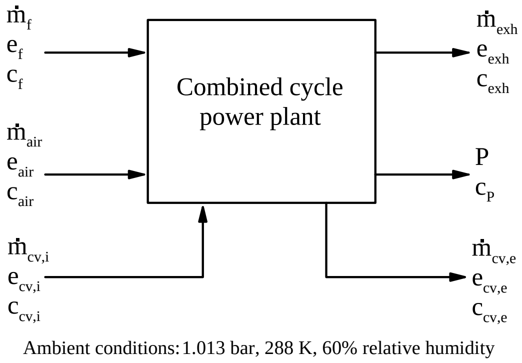

Figure 1. Control volume over the plant

All thermodynamic calculations were performed with the programmable equation-solving software IPSEpro (Anon., 1998). The off-design characteristics for the steam turbine and heat exchangers were taken from the APP_Lib component library that is provided with IPSEpro, whereas all gas turbine characteristics are in-house codes. The exergy of all streams is calculated with the additions to IPSEpro that were made by Sjögren and Westerberg (2000), based on Bejan et al. (1996). The exergy balance over the entire plant at the new and clean condition gives that the exergy lost to the environment is 29.6 MW, the exergy destroyed within the plant is 55.1 MW and the exergetic efficiency is 50.8%.

TABLE I. PERFORMANCE AND INPUT DATA FOR REFERENCE POWER PLANT

Power output, P	62.1 MW
Thermal efficiency, η_{th}	52.5%
Exergetic efficiency, η_{ex}	50.8%
GT Pressure ratio, π	20
Steam pressure HP/LP	80/8 bar
Steam temperature	813 K
Condenser pressure	0.04 bar
ISO TIT	1473 K
Fuel LHV	50000 kJ/kg
Fuel exergy, e_f	51760 kJ/kg
Mechanical efficiency	0.99
Generator efficiency	0.98

<u>Compressor degradation</u>

Depending on the location of a gas turbine power plant, the air that enters the compressor can contain various amounts of salt and other particles, which will form deposits on the compressor blades and contribute to the so-called fouling of the compressor. Compressor fouling disturbs the flow field and reduces the mass flow through the compressor, and sometimes also the compressor pressure ratio. Depending on the location, the fouling is a phenomenon that is specific for each and every gas turbine. To restore the performance of the compressor, washing is required and can be done either on-line or off-line. An extensive description of compressor fouling phenomena and the effect of compressor washing is given by Stalder (1998). Compressor washing and the mechanisms of compressor fouling are also described in Tarabrin et al. (1996) who also, with reference to Schurovsky and Levin (1986), give a semi-empirical exponential law for estimating changes in required compressor output.

Compressor fouling is most severe right after compressor washing; thereafter it slows down and eventually stabilizes. Hence, an exponential law like the one mentioned above was also applied to model the compressor fouling in this paper. However, the choice was made to consider the reduction in compressor mass flow as an exponential function:

$$\frac{\dot{m}_{cl} - \dot{m}_{d}}{\dot{m}_{cl}} = A\left(1 - e^{Bt}\right) \tag{1}$$

The value of the constant A is set to 0.05, i.e. the fouling is supposed to stabilize at a mass flow reduction of 5%. The value of the constant B determines the rate of the fouling. Three different rates of fouling were investigated, that lead to a compressor inlet mass flow reduction with 4% in 500, 1000 or 1500 hours. The resulting values of B are given in TABLE II. The reduction of the polytropic efficiency of the fouled compressor was assumed to be proportional to the inlet mass flow.

TABLE II. PARAMETER B FOR DEGRADATION RATE

4% $\dot{m}$ reduction at h	B
500	$3.22*10^{-3}$
1000	$1.61*10^{-3}$
1500	$1.07*10^{-3}$

Compressor fouling is a temporary kind of performance deterioration that is easily restored, but there are also many other permanent phenomena, such as, wear in seals, that lead to increased tip clearance. Permanent performance deterioration is very insignificant by how often the compressor is washed and will not be considered further in this paper since it is assumed that the marginal cost of produced electric power is unaffected by permanent performance deterioration.

Operation at constant power output

The assumed operation mode in this paper is that the power output of the combined cycle is maintained constant. This means that as the compressor inlet mass flow is reduced, the firing temperature must be increased to compensate for this. The result of this is that both (first law) thermal efficiency and (second law) exergy efficiency of the combined cycle remain almost constant, since the increased temperature in the exhaust gas is recovered in the steam cycle. In contrast to this, Stalder (1998) reports that in a combined cycle on “temperature control mode”, compressor fouling leads to decreased steam production.

Exergy balance over the plant

An exergy balance over the control volume in *Figure 1* gives that:

$$\begin{aligned} &\dot{m}_{air}e_{air} + \dot{m}_{f}e_{f} + \dot{m}_{cv,i}e_{cv,i} = \\ &P + \dot{m}_{exh}e_{exh} + \dot{m}_{cv,e}e_{cv,e} + \dot{E}_{D} \end{aligned} \tag{2}$$

where $\dot{m}_{exh}e_{exh} + \dot{m}_{cv,e}e_{cv,e}$ is the exergy lost to the environment.

In *Figure 2*, the difference in supplied exergy, lost exergy and destroyed exergy, with respect to the new and clean condition (t=0) of the power plant, are shown. It can be seen that the exergy supplied is reduced (due to a small decrease in fuel requirements), that the destroyed exergy is reduced (due to the increase in firing temperature in the gas turbine) and that the lost exergy is slightly increased due to even small increases in exhaust gas and condenser cooling water temperatures. Nevertheless, the changes are very small compared to the absolute values given above for the new and clean condition. This is due to the choice to study a power plant with fixed power output, which, more specifically, means that the decrease in compressor inlet mass flow due to fouling and the increase in firing temperature are more or less in balance so that the change in fuel flow is virtually zero.

3. Blade Temperatures and Equivalent Hours

Increased firing temperature means that the expander of the gas turbine engine must operate with increased gas temperature, which, in turn, will result in an increase in blade temperature. A simple linear correlation between T_g and T_b, derived from a diagram presented in Geipel et al. (1998), was employed for this purpose. As an example of the results for this correlation, a gas temperature increase of 40 K will result in an average blade temperature increase of approximately 10 K. The model employed for calculations of the cooled turbine uses a uniform blade temperature, T_b, for each blade row, which is assumed to be close to the average blade temperature.

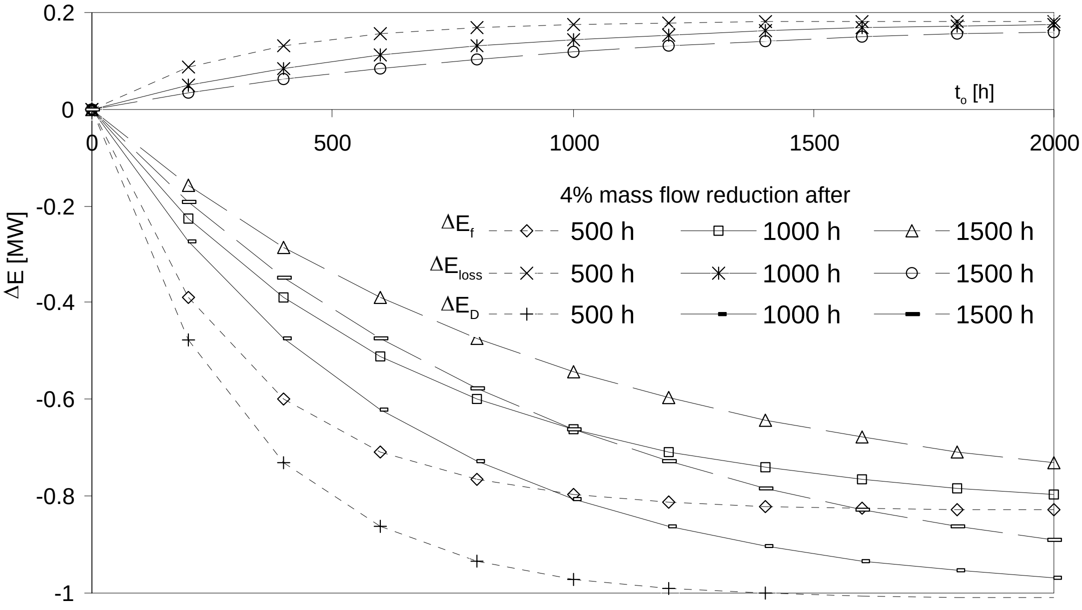

Figure 2. Difference in total fuel exergy, total exergy loss and total exergy destruction over time

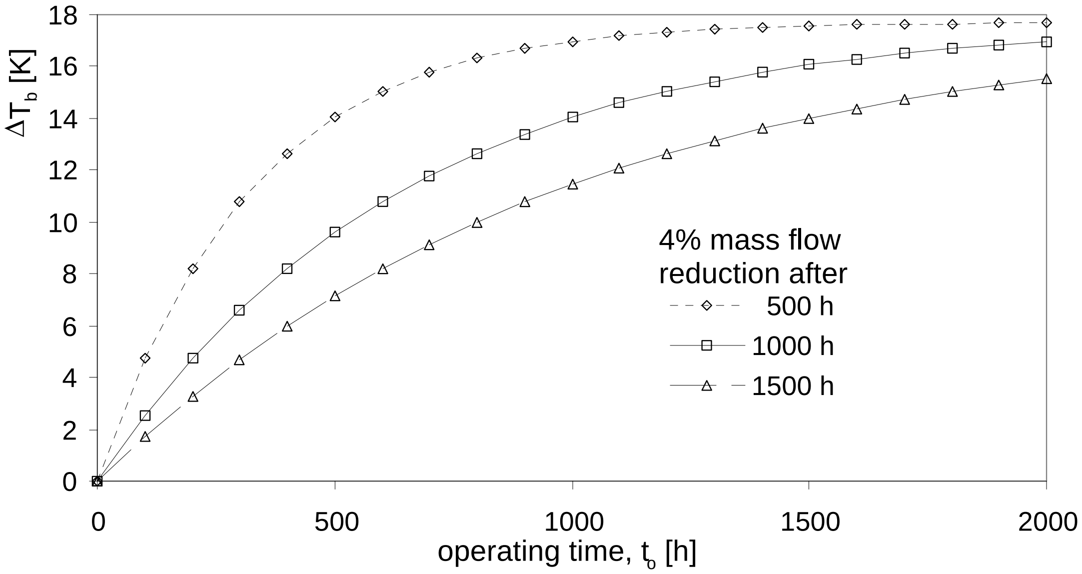

Figure 3. Increase in gas turbine model uniform blade temperature T_b as a function of operating hours for various degradation rates. No fouling at t_o=0.

The increase of uniform blade temperature with time is shown in *Figure 3* for the three different rates of degradation. For more details concerning the modelling of the cooled gas turbine, refer to Jordal (2001).

In all calculations, temperatures for the first stage vanes are used, but replacement costs for the entire first stage (vanes and rotating blades) are considered. Vanes experience a higher temperature than the rotating blades, but first-stage blades experience higher stress, and are usually the most critical blade row. In the very simplified analysis in this work, it is assumed that this leads to blades and vanes accumulating the same amount of equivalent hours.

Degraded components may eventually lead to gas turbine failure. A very detailed list of failures and damages observed in gas turbines is presented by Höxterman and Richter (2000), and for stator and rotor blades, overheating, hot gas corrosion and various coating failures are mentioned among other things. The two main blade failure mechanisms in the first stage of the gas turbine are creep (for base-load plants) and thermal mechanical fatigue (TMF) in peaking plants (Allen and Viswanathan 1999). Since the power plant in the present study is operating 6000 h/year, it is assumed that creep is the dominant mechanism that consumes blade life.

A typical design lifetime of a set of blades is 40000 equivalent hours, one equivalent hour meaning the blade lifetime that is consumed at blade design temperature operation at ISO conditions. Start-up, shutdown, trips and operation at increased blade temperature (caused e.g. by increased hot gas temperature) decrease the number of actual operation hours for the blading, and operation at lower blade temperatures can lead to the situation that during one actual hour, less than one equivalent hour of blade life is consumed.

It is difficult to assess the lifetime behavior and cost of recent advances in the design of cooled blades and in high-temperature materials, and the values employed in this paper must be regarded as (coarse) estimates. What is certain is that the correlation between material's temperature and resistance to creep is an exponential function, and it is, therefore, assumed in this paper that the number of blade equivalent hours, n_{eqh}, used per hour of operation is also an exponential function of the kind:

$$n_{eqh} = De^{C\Delta T} \tag{3}$$

where C and D are constants. Since at the time $t_o=0$ the compressor is clean, $\Delta T=0$ and one equivalent hour equals one actual hour, the value of D equals 1. The value of C varies with material properties. Three different modes of blade life consumption were studied: at an average blade temperature increase of 10 K, it was assumed that 1.2, 2 or 4 equivalent hours were consumed per operating hour. The corresponding values of the constant C can be found in TABLE III.

TABLE III. PARAMETER C FOR EQUATION (3)

Equiv. hours consumed at ΔT_b=10K	C
1.2	0.0182
2	0.0693
4	0.139

The number of equivalent hours consumed per hour, as described by equation 3 can be determined through an equation that describes $\Delta T(t)$. Numerical expressions for the three curves in *Figure 3* were elaborated on the form:

$$\Delta T = ax^3 + bx^2 + cx + d \tag{4}$$

where:

$$x = e^{-t/1000} \tag{5}$$

and, hence it was possible to calculate the total number of accumulated equivalent hours for the blading at a given hour through:

$$n_{eqh} = \int_0^t e^{C\Delta T(t)} \tag{6}$$

The results of calculations with equation 6 are shown in *Figure 4*. The legend in *Figure 4*

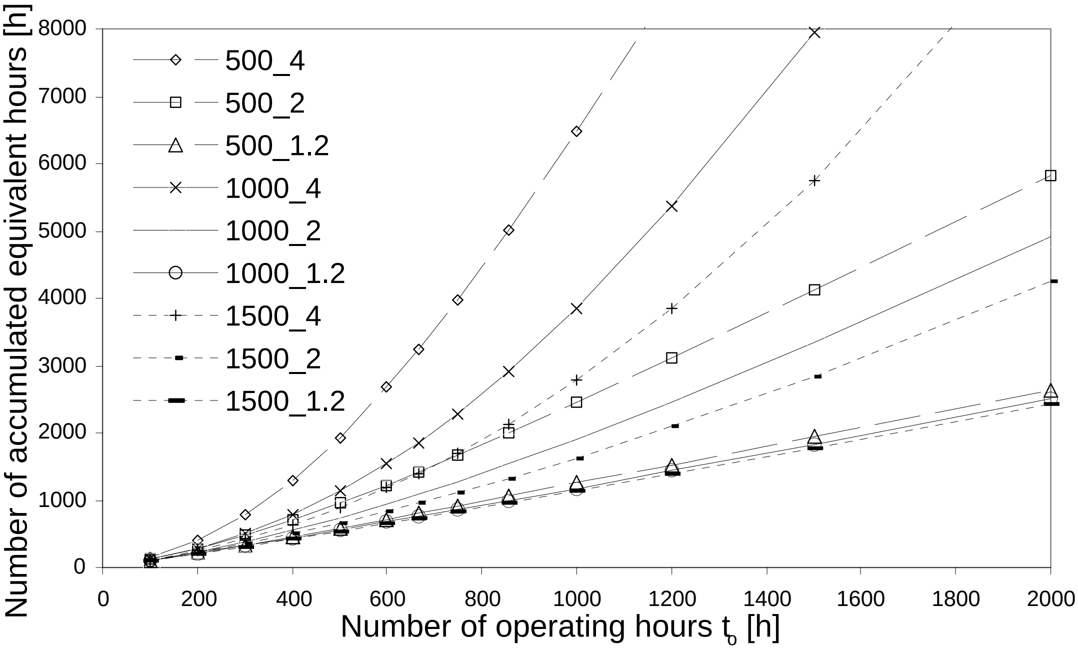

Figure 4. Accumulated equivalent hours for various degradation rates and blade materials

and in the subsequent *figures* should be read so that the first number (500, 1000 or 1500) indicates the number of hours after which mass flow has decreased by 4%, and the number after the underscore (4, 2 or 1.2) indicates how many equivalent hours are consumed at a blade temperature increase of 10 K.

The first obvious result that can be seen in *Figure 4* is that with a compressor that has a rapid fouling rate, the number of accumulated equivalent hours is much more important than for a compressor in an engine located at a place where fouling is slower. It can also be seen that in a gas turbine with good blade materials, there is less difference in accumulated equivalent hours among different fouling rates. In other words, a modern gas turbine with single-crystal blades should be less sensitive to where it is placed than an older gas turbine with polycrystalline blades.

4. Basic Economic Considerations

The benefits of compressor washing in terms of recovered electricity generation capacity were described by Stalder (1998). In the present work, the focus on the economic impact of compressor washing is instead put on how it affects blade lifetime and gas turbine life cycle cost. Through the choice to study a power plant with a contract to deliver a fixed electric power, the income from sold electricity will be identical in all cases in this study, and, hence, can be neglected in any studies of varying cost. Instead three other economic variables are employed: blade cost, fuel cost and the cost of washing the compressor. The input data for the economic studies are given in TABLE IV.

TABLE IV. ECONOMIC INPUT

Economic lifetime, n	20 years*
Rate of return, r	8%*
Operation time, t_o	6000 h/year*
Specific fuel cost, fc	0.10 SEK/kWh*
Exergetic fuel cost, c_f	0.097 SEK/kWh
Blade cost, k_b, _1.2	300 SEK/eqh
Blade cost, k_b, _2	250 SEK/eqh
Blade cost, k_b, _4	200 SEK/eqh
Cost for one compr. wash, K_w	100000 SEK

Figures in TABLE IV that are marked with * are set according to data given in a report by the Swedish Electric Power Research and Development Association (Elforsk 2000).

Regardless of whether a first-law approach or a second-law approach is applied to an economic study of a power plant, the total fuel cost will be the same for the power plant owner, i.e.

$$fc \cdot LHV = c_f \cdot e_f \tag{7}$$

For a set of blades for one stage in a modern gas turbine, a typical total cost, including manufacturing and replacement would be 10 MSEK, which, when divided with the design lifetime of 40000 equivalent hours, results in a cost per equivalent hour of 250 SEK. The more advanced blading (index _1.2) is assumed to have a higher hourly cost and, consequently, the simpler blading (index _4) to have a lower cost.

5. Thermoeconomic Aspects

Thermoeconomics can be applied to study the internal cost flows of a power plant, or more precisely in the present case, to determine the *differences* in the internal cost flows of the plant. The only difference in internal cost is the difference associated with blade life time and compressor wash, and a minor difference in fuel cost, which means that the control volume model in *Figure 1* is sufficient to determine the change in cost for the product, i.e. the produced electric power.

Letting c denote the average cost per unit of exergy and $\dot{C}$ the cost rate we can write (Bejan et al. 1996):

$$\dot{C} = c\dot{E} = c\dot{m}e \tag{8}$$

In this particular case, only electric power generation is considered, meaning that the value of the exhaust gas stream is zero, which is also the case for the cooling water leaving the bottoming cycle condenser. Also, the air entering the compressor and the water entering the condenser are considered not to be associated with any cost. This means that the cost balance over the entire plant can be written as:

$$\Delta\dot{C}_W = \Delta\dot{Z}_k + \Delta\dot{C}_f \tag{9}$$

where $\dot{Z}_k$, following the nomenclature of Bejan et al. (1996), is the cost rate due to capital investment and operation and maintenance costs. The use of Δ means that the difference between an arbitrary time and at the new and clean condition is considered. Note that even if an economic value had been assigned to the streams leaving the plant, this value could, in this particular case, have been neglected due to the very small changes in exergy of the streams leaving the plant (refer to *Figure 2*).

Since the difference in investment cost is zero, regardless of when the compressor is washed, the interpretation of equation 9 is that the difference in cost of produced electric power with respect to the new and clean conditions is equal to the difference in blade cost, washing cost and fuel cost, which means that it is possible to write:

$$\Delta\dot{C}_w = k_b\left(n_{eqh}(t) - 1\right) + \frac{K_w}{t_w} + c_f \cdot e_f\left(\dot{m}_{f,d}(t) - \dot{m}_{f,cl}\right) \tag{10}$$

for the difference in cost rate from the new and clean condition till the moment when the compressor is washed. It must be pointed out at this moment that, since the electric power produced is unchanged and consequently also the income of sold electric power, equation 10 actually gives the *net* cost of compressor fouling per unit of time.

Integrating over one year gives that:

$$\int_0^{6000} \Delta\dot{C}_w dt = K_a \tag{11}$$

where:

$$K_a = n_w\Big(k_b \int_o^{tw} \left(n_{eqh}(t) - 1\right)dt + K_w + c_f \cdot e_f \int_o^{tw} \left(\dot{m}_{f,d}(t) - \dot{m}_{f,cl}\right)dt \Big) \tag{12}$$

where n_w is the number of compressor washes executed during one year.

6. The Cost of Compressor Fouling

The annual operation time in TABLE IV, 6000 hours, was divided into different time intervals from 100 to 2000 hours to study the impact of varying number of compressor washes over one year. Washing intervals longer than 2000 hours were not assumed to be realistic and, hence, not included in the calculations. Equation 12 was employed to study the cost for compressor fouling over one year. The results for equation 12, but without the cost of the washing, are shown in *Figure 5*. The change in fuel consumption in the studied case was found to be so low that it had no significant impact on the cost in this particular case.

Blade lifetime and blade cost are (obviously) closely connected, as can be seen in *Figure 5*. The blades with little resistance to temperature increase (index _4) become very expensive as washing intervals increase, although their specific cost was lower than the other blade materials. It can also be seen that for very heat resistant materials, the compressor fouling rate is of little importance for the cost of compressor fouling. In *Figure 5*, it appears that short time intervals between the compressor washes is the most beneficial since this means that blade lifetime is preserved. However, there is a cost for preserving this blade lifetime, that is, the cost of the washing itself, included as K_w in equation 12. This cost consists of several parts: the power that is lost (if washing occurs during hours when power is normally delivered according to contract) must be replaced, perhaps through a purchase on the spot market. Furthermore, there is a cost for detergents and a labor cost for the washing. This cost can be most varying among different plants and much depends on the cost of replacing missing electric power generation capacity during the washing. It was set to SEK 100000 in this work just to illustrate that it has an impact. The results can be seen in *Figure 6*.

The cases with very heat resistant blading show their lowest cost at the minimum number of compressor washes (3/year), but for all other cases there is an obvious minimum where the cost of the blading has decreased with more frequent washes, and the accumulated cost of the washes is not yet too big. For each case studied, the minimum cost is listed in TABLE V.

Through the use of basic capital investment theory (Northcott 1998), the accumulated cost of an expense over a long time can be calculated. The sum of present values of a series of future costs is calculated through

$$\Sigma PV = K_a \frac{(1+r)^n - 1}{r(1+r)^n} \qquad (13)$$

where r is the rate of return and n is the number of years. Depending on the fouling rate, it can be seen in TABLE V that for the probably most realistic case of blading (index _2), the present value of the cost of compressor fouling over 20 years, including its impact on materials deterioration, lies between 12 and 21 MSEK, which is a small amount compared to the approximate investment cost of 250 MSEK for such a power plant, but definitely not negligible. In particular, this cost is not negligible when considering that it is the *minimum* cost, and that careless choice of compressor washing intervals can increase this cost.

7. On-Line Gas-Turbine Monitoring

In order to monitor a power plant and plan for maintenance and operation in an optimum way, it is desirable to have some kind of on-line condition monitoring system (CMS), such as one that is based on a thermodynamic model or on an artificial neural network (Assadi 2000).

The impact from compressor fouling on how the first-stage gas turbine blading accumulates equivalent hours, which calls for washing, was demonstrated above. Although the assumptions made initially for fouling rate and materials' behavior were very simple, the calculations become somewhat complicated. Hence, it is a very complicated task to create thermodynamic

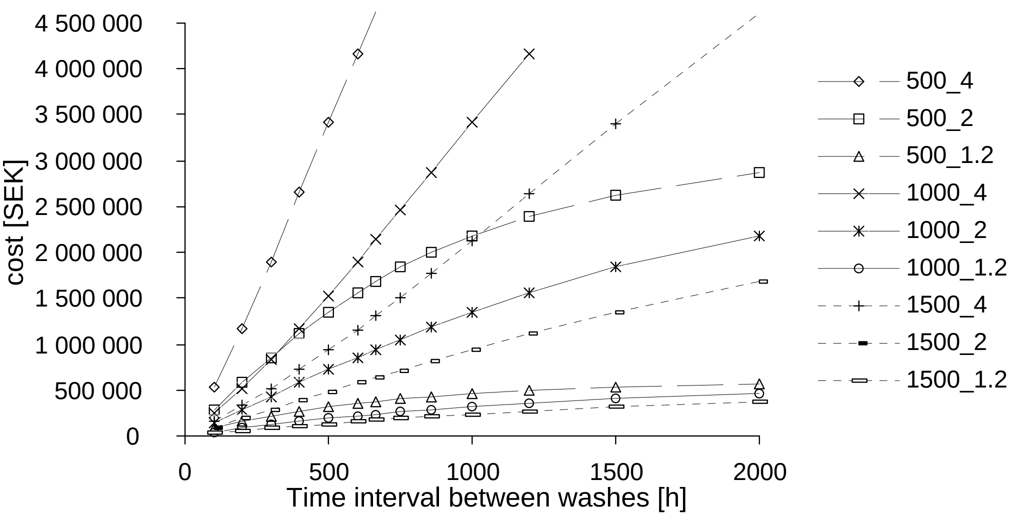

Figure 5. The annual cost of compressor washing as a function of washing intervals.

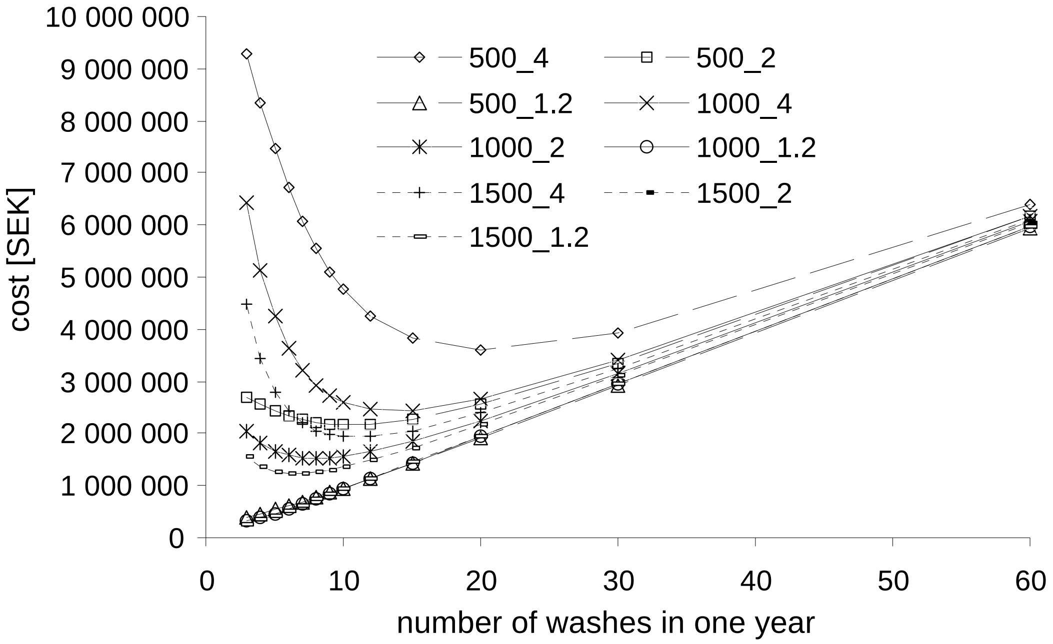

Figure 6. Annual cost of compressor fouling, including cost of washing as a function of number of washes.

TABLE V. DATA FOR POINTS OF MINIMUM ANNUAL COST

Case	# washes	Time interval [h]	# eqh/year	# blade replacements in 20 years	annual cost (MSEK)	PV, 20 years' cost (MSEK)
500_4	20	300	15460	7	3.60	35.4
500_2	10	600	12251	6	2.16	21.2
500_1.2	3	2000	7895	3	0.40	3.93
1000_4	15	400	11805	5	2.44	24.0
1000_2	8	750	10214	5	1.53	15.0
1000_1.2	3	2000	7535	3	0.31	3.05
1500_4	10	600	11789	5	1.94	19.0
1500_2	7	857	9262	4	1.24	12.1
1500_1.2	3	2000	7256	3	0.26	2.60

models that make on-line calculations of performance behavior from measurements for an entire power plant, including all kinds of temporary and permanent degradation. Examples of this can be found in Razak and Carlyle (2000) and Gayraud and Singh (1999).

In this context it should be recalled that degradation phenomena are rather complex. For instance, the compressor fouling varies from one engine to another. Also, the fouling can vary over time in one engine where the constant B in equation 1 varies due to varying salt concentration in the air, increasing traffic, new industries or cleaning of the emissions from existing industries close to the gas turbine. Furthermore, temperature profiles are not necessarily uniform around the stage, due to the non-uniform hot gas temperature field after the combustion chamber and due to manufacturing tolerances that can affect the cooling inside the blade. Monitoring of the blade surface temperatures with infrared pyrometers can help in discovering this.

Hence, an alternative to thermodynamic modelling should be to employ artificial neural networks (ANN) for on-line monitoring. In order to implement preventive or condition-based maintenance techniques, it is essential that early warning of developing faults be provided so that the appropriate decisions may be taken and correct actions planned in advance. For this reason, various condition monitoring techniques have been developed (Volponi et al. 2000, Bergman and Woud 1993), which generally involve power plant health monitoring using case based reasoning, ANN, performance physical modelling analysis and many more.

Existing/proposed monitoring techniques involve mathematical and/or causal models, which simulate the healthy and faulty plant/component in order to identify or verify a particular condition (Kao and Moskwa 1995). These models have been applied with various degrees of success. Often they are slow to execute and hard to implement.

In recent years there has been an increased interest in the application of ANN for engineering problems. An ANN is formed by connection of artificial neurons which are very simple processing units It is not programmed, as are traditional mathematical models or algorithms. Instead ANNs have the ability to identify and learn the relationships between the inputs and the outputs of e.g. a non-linear multi-dimensional system, and therefore, they have become very popular for solving problems that are difficult to solve by traditional means.

The learning process is called training, which requires that some knowledge about the system to be modelled is available, meaning that some initial data acquisition must be made in the case of a power plant. (Here it must be noted that the accuracy of the ANNs is only as good as the accuracy of the data they are trained with.) After the neural network is trained, it is able to recall this knowledge, in order to make predictions of the outcomes when new inputs (not used during training) are presented. This property is called generalization (Arriagada, 2002). The overall features of ANNs indicate that they should be most useful for on-line monitoring of power plants to detect e.g. unhealthy materials temperatures and help in the planning of washing and other maintenance measures.

Furthermore, ANN combined with models from financial analysis, as shown above, should be suitable for optimizing the operation of a power plant from an economic point of view – making it possible to extrapolate and make performance predictions from the current status of the power plant, if the system is combined also with a database with blade material's behavior at varying loads and temperatures, and how the material degrades. This opinion is supported by the fact that a goal of the ongoing EPRI blade life management program is to link a database with thermal mechanical fatigue cases to on-line tracking signals/controls (Stambler 2000). Also for long-term studies, which are of use for studies of various design concepts, and their economic viability or for the assessment of the future expected profitability of an existing power plant, experiences and output data from a decision support system based on neural networks ought to be helpful.

8. Conclusions

An attempt was made to quantify the in practice well known but, in theory little studied, correlation between compressor fouling and turbine blade replacement cost and intervals. It was shown that for advanced blade materials (such as single crystal), the lifetime is less sensitive to the fouling rate than for more common materials (such as polycrystalline blades). It was also shown how there could be an economically optimum number of compressor washes over a year, where a trade-off is made between the cost of the washing and the cost of the consumed blade life.

The ANN based condition monitoring and sensor validation hardware/software is an on-line system, which can be transferred to an adaptive platform. With this improvement, new engine/module conditions could be added to the trained data sets for a retraining process embracing new operational situations.

Acknowledgements

The work of Kristin Jordal and Mohsen Assadi was financed by the Swedish National Energy Administration, as a part of the project "Methods for Analysis and Optimisation of Thermal Power Plants".

Nomenclature

A,B,C,D	constants
$\dot{C}$	cost rate [SEK/h]
c	exergetic cost rate [SEK/h]
$\dot{E}$	exergy [MW]
e	specific exergy [kJ/kg]
fc	fuel cost (SEK/kWh)
K	cost (SEK)
k_b	specific blade cost (SEK/eqh)
LHV	lower heating value (kJ/kg)
$\dot{m}$	mass flow (kg/s)
n	number
P	electric power [MW]
PV	present value (SEK)
T	temperature (K)
t	time (h)
t_w	time between two compressor washes (h)
$\dot{Z}$	non-exergetic cost rate [SEK/h]
η	efficiency

Subscripts

a	annual
air	air
b	blade
cl	clean
cv	cooling water
D	destroyed
d	degraded
e	exit

eqh	equivalent hour
exh	exhaust
f	fuel
i	inlet
o	operating
th	thermal

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