THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

96-GT-363

AN ANALYSIS OF AXIAL COMPRESSORS FOULING AND A CLEANING METHOD OF THEIR BLADING

A. P. Tarabrin
Doctor of Science, Polzunov Boiler and Turbine Research Institute, NPO CKTI,
St. Petersburg, Russia

V. A. Schurovsky
Doctor of Science, Gas Research Institute,
VNIIGAS, Moscow, Russia

A. I. Bodrov
Engineer, Institute of Machine Building,
PIMACH, St. Petersburg, Russia

J-P. Stalder
Turbotect Ltd., Baden, Switzerland

ABSTRACT

The paper describes the phenomenon of axial compressor fouling due to aerosols contained in the air. Key parameters having effect on the level of fouling are determined. A mathematical model of a progressive compressor fouling using the stage-by-stage calculation method is developed. Calculation results on the influence of fouling on the compressor performance are presented. A new index of sensitivity of axial compressors to fouling is suggested. The paper gives information about the Turbotect's deposit cleaning method of compressor blading and the results of its application on an operating industrial gas turbine. Regular on line and off line washings of compressor flow path make it possible to maintain a high level of engine efficiency and output.

NOMENCLATURE

A_t - cascade throat, m,
b - chord, m,
b/t - solidity,
C_0 - stream velocity at the distance upstream, m/s,
C_z - axial component of absolute velocity, m/s,
C_{1u} - circumferencial component of absolute velocity in the inlet, m/s,
C_{2u} - circumferencial component of absolute velocity in the outlet, m/s,
c_p - specific heat, J/(kg K),
c_m - relative maximum thickness of profile,
D_c - tip diameter, m,
D_h - hub diameter, m,
D_m - mean diameter, m,
d_p - particle diameter, m,
E - entrainment coefficient,
E_c - cascade entrainment coefficient,
G - mass flow, kg/s,
G_0- nominal mass flow, kg/s,
H - real head, J/kg,
H_{th} - theoretical head, J/kg,
h - distance between the stream centreline and the limit trajectory at the upstream infinity, m,
i - incidence angle, deg,
k_{Hr} - the work done factor,
k_{Hf} - the work done factor including fouling effect,
k_{Ne} - output coefficient of the technical condition,
L - characteristic size of the body, m,
l - blade length, m,
N_e - output, kW,
N_{e0} - nominal output, kW,
N_e^{T10} - operational output corrected to design conditions, kW,
n - rotation speed, rpm,
n_0 - nominal rotation speed, rpm,
P^*_{in} - total pressure in the inlet of the compressor, Pa,
P^*_{ex} - total pressure in the exit of the compressor, Pa,
R - cylinder radius, m,
Re_f - Reynolds number of flow,
r_1- mean radius in the inlet of the rotor, m,
r_2- mean radius in the outlet of the the rotor, m,
$\overline{r}_h$ - hub/tip ratio of the first stage,
Stk- Stokes number,
T^*_{in} - total temperature in the inlet of the compressor, K,
T^*_1 - total temperature in the inlet of the rotor, K,
T^*_2 - total temperature in the outlet of the rotor, K,
ΔT^*_{stg} - average total temperature rise per stage, K,
t - pitch of a cascade, m,
u - circumferential velocity, m/s,
w_1 - flow velocity in the inlet of the blade, m/s,
w_2 - flow velocity in the outlet of the blade, m/s,
x_{cmax} - position of maximum profile thickness, m,
x_f - position of maximum profile concavity, m,
β_b - stagger angle with the axis u, deg,
β_1 - flow angle with the axis u, deg,
β_2 - flow exit angle, deg,

Presented at the International Gas Turbine and Aeroengine Congress & Exhibition
Birmingham, UK — June 10-13, 1996
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until September 30, 1996

ξ_{in} - loss coefficient of inlet duct,
ξ_{ex} - loss coefficient of exhaust duct,
ρ - air density, kg/m^3,
ρ_p - particle density, kg/m^3,
π - pressure ratio,
η_{ad} - adiabatic efficiency,
δ - deviation, deg,
χ_1 - angle between the chord and the tangent to the profile centre line in the inlet of the blade, deg,
χ_2 - angle between the chord and the tangent to the profile centre line in the outlet of the blade, deg,
μ - coefficient of viscosity, Pa s,
$\Delta\overline{p_\Sigma}$ - coefficient of the total pressure loss,
$\Delta\overline{p_{f\Sigma}}$ -coefficient of the total pressure loss including fouling effect,
k_f - factor of fouling,
ω - wheel angular velocity, rad/s.

INTRODUCTION

The air used for the operation of gas turbines always contains a certain amount of aerosols - airborne solid and liquid particles. Their quantity greatly increases under unfavourable conditions: dust and sand storms, dirty industrial environments, etc. To reduce an adverse effect of fouling on the compressor blading and its performance, an effective filtration system can be installed in the compressor inlet. Theoretically, complete inlet air purification can be obtained, but there is no purpose to this in view of great expence and additional losses in output and efficiency. Therefore the main function of inlet filters is to protect the blading from erosion, and compressor fouling cannot be completely eliminated. It is noted (Diakunchak, 1991) that compressor blading fouling is caused mainly by dust particles of about 2 μm and less in diameter and their mass constitutes up to 80-90% of the total dust mass behind the filter.

The sources of fouling of the compressor inlet air are described in (Diakunchak, 1991), (Mikhaylov et al, 1978), (Schurovsky and Levikin, 1986), (Olhovsky, 1985). Fouling of the compressor blades is caused by the "soft" aerosols contained in the air: dirt, dust, pollen, insects, oil and water vapour, sea water salt, sticky industrial chemicals, unburnt hydrocarbons, soot particles, etc. Some of these contaminants are products of the operating process of the engine main components and auxilaries: oil leaks from compressor seals and bearings, salt and mechanical impurities from the water-evaporative cooling system, etc. Some of these particles falling on the blade surface adhere to it, stick to each other and produce a deposit. This leads to changes in aerodynamic form, profiles roughness and in angle of attack. According to (Olhovsky, 1985), the deposits do not considerably change the throat of a compressor profile cascade. The compressor fouling causes a decrease in its mass flow, efficiency, pressure ratio and surge margin. For example, as it was noted (Diakunchak, 1991), compressor fouling resulted in a drop of about 5% in the mass flow, 2.5% in the efficiency and 10% in the output. Further, and to enhance this, General Electric is quoting in (Hoeft, 1993) that an airflow reduced by 5% will reduce output by 13% and increase heat rate by 5.5%.

In connection with the development of gas turbine diagnostics it is necessary to develop reliable methods for predicting the behaviour of engine performance in the course of fouling.

Some mathematical simulations of fouling in multistage compressors have recently appeared (Aker and Saravanamuttoo, 1988), (Sedigh and Saravanamuttoo, 1990). Using the field operating data it was found that the first stages of a compressor are the most fouled ones and the number of fouled stages constitutes 40-50% of the total. Besides that, the degree of fouling diminishes from stage to stage. Based on these physical considerations, Aker and Saravanamuttoo (1988) suggested the linear progression of fouling in compressor stages (k_1:k_2), where k_1 is a percentage of the flow coefficient reduction from $n \times k_1$ for the first stage to k_1 for n^{th} stage; in the same way k_2 factor is applied to stage efficiency. Stage stacking technique (Howell and Calvert, 1978) was used in this mathematical model to calculate a compressor map through the performance of every stage. Attempts were made to lay down an index of fouling (Sedigh and Saravanamuttoo, 1990) in order to determine axial compressors sensitivity to fouling. In the authors opinion the parameter $N_e/(G\ c_p\ \Delta T_{stg})$ is the most suitable index for comparing different engines in order to determine the sensitivity of their compressor to fouling. It can be seen that sometimes this index is the same for engines with small and large output, that is, for the engines of small and large sizes. At the same time there is evidence (Sedigh and Saravanamuttoo, 1990) that the performance of a smaller engine shows more deterioration by fouling than that of a larger one.

In this connection it seems to be reasonable to consider principles of the motion of particles in the flow path of the compressor, the mechanism of the deposits formation and to work out on this basis a new generalized criterion (index) for the compressor sensitivity to fouling.

MECHANISM OF FOULING

The mechanism of entrainment of particles by a surface of a body situated in the stream of the air-aerosol mixture was described in (Fuks, 1955). The depositing of particles on the surface of a blade takes place under the action of inertia forces acting on the particles and forcing them to move across the curved stream lines (the particle trajectory deviates from the stream lines). Particles of dirt (dust) colliding with the blade can stick to the blade surface. On the other hand, when the particles move in the flow path of compressor the centrifugal inertia forces make them move to the periphery of the compressor passage.

The coefficient of entrainment or the separation factor E according to (Fuks, 1955), is determined as a ratio:

$$E = \frac{\text{number of particles colliding with the surface of the body}}{\text{number of particles which could fall on the body surface if the stream lines were not deviated by the body}}$$

As it appears from Fig. 1:

$$E = \frac{h}{L} \tag{1}$$

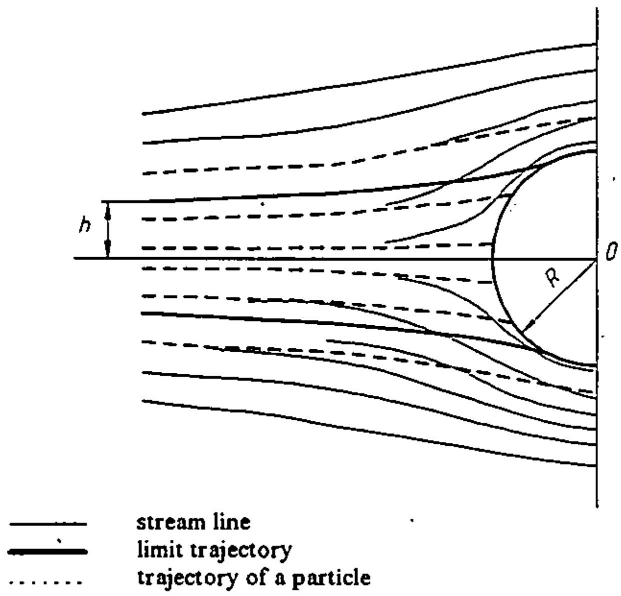

Figure 1
Inertial Deposition of Particles on Cylinder Surface

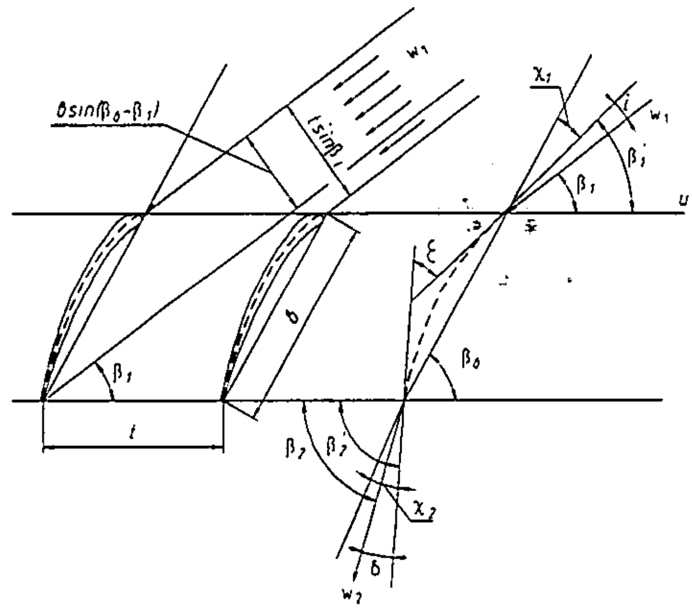

Figure 2
Cascade of Profiles of Axial Compressor

In case of cylinder L=R.

In the general case, E value can be presented as a function:

$$E = f\left(Stk, \frac{d_p}{2L}, Re_f\right) \tag{2}$$

where

$$Stk = \frac{\rho_p d_p^2 C_0}{18 \mu 2L}, \quad Re_f = \frac{\rho\, 2L C_0}{\mu},$$

In case of a cascade, Fig. 2,

$$L = b \sin(\beta_b - \beta_1) \tag{3}$$

can be taken for the characteristic size (Olhovsky, 1985).

Coefficient of entraiment for the cylinder with $R = b \sin(\beta_b - \beta_1)$ within the unbounded stream with R as the characteristic size will be the following:

$$E = \frac{h}{b \sin(\beta_b - \beta_1)} \tag{4}$$

Considering that in the cascade the flow path is bounded by two adjacent profiles with pitch t, one can take for the cascade entrainment coefficient:

$$E_c = \frac{\text{number of particles sticking to surface}}{\text{number of particles in the flow for one pitch of the cascade}} \tag{5}$$

Then

$$E_c = \frac{h}{t \sin\beta_1} = \frac{h}{b \sin(\beta_b - \beta_1)} \, \frac{b}{t} \, \frac{\sin(\beta_b - \beta_1)}{\sin\beta'_1} \tag{6}$$

or

$$E_c = E \, \frac{b}{t} \, \frac{\sin(\beta_b - \beta_1)}{\sin\beta_1} \tag{7}$$

It can be seen from Fig.2 that

$$\beta_1 = \beta_1' - i = \beta_b - \chi_1 - i \tag{8}$$

$$\beta_2 = \beta_2' - \delta = \beta_b + \chi_2 - \delta \tag{9}$$

From Eq.(8) and (9) for a profile with a mean (skeleton) line of circular arc

$$\frac{1}{2}(\beta_1 + \beta_2) = \beta_b - \frac{1}{2}(\delta + i) \tag{10}$$

Taking into account that for the cascade situated in the mean radius of a compressor stage and for the optimal regime of work $i \approx 0°$ and $\delta \leq 8°$-$10°$, we will roughly obtain

$$\beta_b \cong \frac{1}{2}(\beta_1 + \beta_2) \tag{11}$$

and

$$\beta_b - \beta_1 = \frac{\beta_2 - \beta_1}{2} = \frac{\Delta\beta}{2} \tag{12}$$

The entrainment coefficient E for the endless cylinder at large Reynolds numbers can be defined theoretically by calculating its potential flow. Based on the data obtained from the calculation of the potential flow around the cylinder by T. Langmuir, K. Bladget and H. Landahl (Fuks, 1955) within the full range of the Stokes numbers, characteristic for axial compressors, the dependence E = E (Stk) at large Reynolds numbers can be approximated with sufficient precision by the formula:

$$E = 0.08855\ Stk - 0.0055 \tag{13}$$

$Stk_{cr} = 0.0625$ corresponds to the critical Stokes number, with which E=0. If Stk > 11.2, we should take E=1.0.

For the cascade the Stokes number equals:

$$Stk = \frac{\rho_p\, d_p^2\, w_1}{18\, \mu\, 2\, b\, \sin(\beta_b - \beta_1)} \tag{14}$$

Considering Eq. (12) and (13), Eq. (7) can be written in the form:

$$E_c = (0.08855\ Stk - 0.0055)\, \frac{b}{t}\, \frac{\sin(\Delta\beta/2)}{\sin\beta_1} \tag{15}$$

As the head (work in a stage) of the compressor stage increases, the flow turning angle ($\Delta\beta$) and the solidity (b/t) of the cascade also increase. Therefore, the high-head stages are more sensitive to fouling than the low-head ones.

As it follows from Eq. (14) and (15), when the size of the chord decreases the Stokes number and the entrainment coefficient increase, that is, a model compressor is more sensitive to fouling than a full-scale one. Besides that, when the velocity w_1 increases the value of E_c also increases.

It is evident that Eq. (15) define the inertial deposition of the particles on the windward side of the blade facing the flow. The deposition of the particles is also present on the leeward side of the blade profile as a result of the whirls and turbulence. The results of the investigation of the deposit formation on the gas turbine compressors blading performed by Olhovsky (1985) have shown that the fouling affects the first 5-6 compressor stages. The blades fouling becomes less as the air passes from stage to stage. The blades fouling is present on both the concave and the convex sides, but the convex side of the rotor and stator blades, except the IGV, is more affected by the deposit formations, than a concave one.

As Fig. 2 and Eq. (15) display, the separation coefficient E_c increases with the increase of angle of attack. This also leads to the increasing number of the tear-off whirls on the convex side of the blade and, consequently, the deposition of the particles becomes more intensive on the both sides of the blade.

It should be observed that not all the small particles that have reached the blade, stick to it. A number of them rebound from the blade surface and are taken away by the flow. Significant factors that influence the adhesion of the particles are the humidity of the air, the presence of oily evaporations, etc.

After this, one can make some conclusions upon the sensitivity of the cascades and stages of axial compressors to fouling: on condition of keeping the aerodynamical and geometrical similarity, the compressor of a smaller size (a model) is more sensitive to fouling than a full-scale one;

- sensitivity to fouling of the axial compressor stage increases when the stage head grows;
- the degree of the particles deposition on the blades increases when the angle of attack grows.

Having determined the main factors influencing the sensitivity of the axial compressor to fouling, in the next section of the present paper we have made an attempt to find a generalized criterion of the sensitivity and to develop a mathematical model for evaluation of the change in the compressor performance in the course of fouling.

FOULING EFFECT ON THE PERFORMANCE OF AXIAL COMPRESSOR

As it was noted in the introduction, the methods of estimation of the fouling effect on the compressor performance were developed recently. In these methods mathematical models are used for the calculation of the behaviour of the compressor under fouling. The application of these methods allows one to get a clear idea about the dynamics of the change in the performance of a compressor and a gas turbine unit, as well as to establish a correlation between the change in such parameters as pressure at the exit of compressor, inlet turbine temperature, rotation speed of a gas generator, compressor mass flow and the decrease in output and efficiency caused by fouling. Some authors try to find a criterion for sensitivity of a compressor to fouling.

Following (Sedihg and Saravanamuttoo, 1990), we have developed the linear progressive model of compressor fouling. However, in contrast to (Sedihg and Saravanamuttoo, 1990), in present paper it is assumed that in a multistage compressor the number of the fouled stages equals 5-6, that is, the first five or six stages retain almost all dirt entering the compressor. This conclusion is based on the experimental data and the field observations of fouling. Quite different from our approach, the paper (Sedihg and Saravanamuttoo, 1990) assumes that up to 40-60% of the compressor stages are submitted to fouling. This difference between the approaches may cause difficulties in comparing the degree of fouling of the compressors with different number of stages. In the linear progressive model of fouling it is assumed that the process of fouling goes by intervals, each of them increases the number of fouled stages by one. The first interval corresponds to the fouling of the first stage when the decrease in its head and efficiency equals ΔH and $\Delta\eta_{ad}$, the second interval causes further decrease in the head and efficiency of the first stage equal to 2 ΔH and 2 $\Delta\eta_{ad}$ accordingly, and the decrease in the second stage characteristics by ΔH and $\Delta\eta_{ad}$ and so on to the sixth stage.

The method based on the cascade test results is used in the present paper for the stage-by-stage calculation of the compressor performance using the computer program. For calculating on the mean radius of each rotor and stator vane, the following values are given: b, β_1', β_2', $\overline{x}_{cmax} = x_{cmax}/b$, $\overline{x}_f = x_f/b$, $\overline{c}_{max} = c_{max}/b$, b/t, $\overline{A}_t = A_t/t$; D_c, D_b, ξ_{in}, ξ_{ex}, n, P_{in}^*, T_{in}^*. The mass flow is defined in the process of solving the equation from the maximum one to the surge limit. In the process of calculation the coefficient of mass flow, the parameters of the flow before and after the vane are determined for each stage.

Table 1
Index of Sensitivity to Fouling (ISF) for Different Compressors

Parameters	engine compressors				
	Centaur Solar	LM 2500 General Electric	GTE-150 LMZ model i=1:4.14	GTE-150 LMZ full scale	V 94.2 Siemens
engine output, kW	2850	20134	-	150000	150000
air mass flow, kg/s	17.2	65.80	35.47	630.00	500.0
pressure ratio	9:1	17.2:1	12.5:1	12.9:1	10.6:1
number of stages	11	16	14	15	16
hub/tip ratio of the first stage	~0.6	0.480	0.422	0.422	0.520
ΔT^*_{stg} per a stage, K	28.20	25.90	24.60	23.34	19.25
tip diameter of the first stage, m	0.4400	0.7356	0.5712	2.3650	2.1730
nominal rotation speed of a gas generator, rpm	15015	9160	12420	3000	3000
ISF	8.94	5.59	5.72	1.36	1.29

The real head of the stage is determined as:

$$H = c_p(T^*_2 - T^*_1) = k_{Hf}\,\omega\,(C_{2u}r_2 - C_{1u}r_1) \qquad (16)$$

where $k_{Hf} = k_H - \Delta k_{Hf}$

The summary coefficient of total pressure loss including fouling effect for the cascade is written in the form:

$$\Delta\bar{p}_{f\Sigma} = k_f\,\Delta\bar{p}_\Sigma \qquad (17)$$

One can see from Eq. (15) that the coefficient of entrainment of the particles for the rotor cascade of the axial compressor is mainly defined by the product:

$$Stk\,\frac{b}{t}\,\frac{\sin(\Delta\beta/2)}{\sin\beta_1} = \frac{\rho_p\,d_p^2\,w_1\sin\beta_1}{36\,\mu\,2\,b\,\sin^2\beta_1}\,\frac{b}{t} =$$

$$= \frac{\rho_p\,d_p^2\,C_z}{36\,\mu\,\dfrac{b}{l}\,\dfrac{D_c}{(D_m/l)+1}}\,\frac{1}{\sin^2\beta_1}\,\frac{b}{t} \qquad (18)$$

For the nominal regimes of the cascade flow there exists an empirical formula (Howell, A. R.):

$$\frac{b}{t} = \frac{1.5\,H_{th}}{1.55\,C_z\,u - H_{th}} \qquad (19)$$

where $H_{th} = C_z\,u\,(\mathrm{ctg}\beta_1 - \mathrm{ctg}\beta_2)$.

The axial speed component is defined from the equation of mass flow:

$$C_z = \frac{G}{\rho\,D_c^2\,(1 - \bar{r}_h^2)} \qquad (20)$$

Considering Eq. (15), (18), (19) and (20) the index of compressor sensitivity to fouling (ISF) can be presented as:

$$ISF = \frac{G\,c_p\,\Delta T^*_{stg}}{(1-\bar{r}_h^2)\,D_c^3}\,10^{-6} \qquad (21)$$

We suppose that this value will reflect qualitatively the sensitivity of the axial compressor to fouling under similar environmental conditions. The values of ISF for different compressors are given in Table 1.

As one can see from this table, the Centaur compressor has the largest ISF value in comparison with the other units shown in Table 1. Assuming the same operation mode and time, the same air quality and its degree of filtration, the Centaur compressor will have a greater reduction in inlet mass flow, pressure ratio and efficiency than the LM 2500 compressor or the LMZ GTE-150 model compressor. Therefore, the regimes of washing of these compressors must be quite different.

An account of the influence of fouling on the compressor performance is carried out using the factors Δk_{Hf} and k_f. From the physical point of view it is evident that these coefficients must depend on ISF and also on a concentration and fractional composition of aerosols. Comparing the sensitivity to fouling of different compressors operating in the same environment, we suppose that Δk_{Hf} and k_f values are linear functions of argument ISF.

$$\Delta k_{Hf} = m\ ISF \qquad (22)$$
$$k_f = 1 + n\ ISF \qquad (23)$$

where m and n are constants.

It is obvious that when ISF = 0 then $\Delta k_{Hf} = 0$ and $k_f = 1$. We suppose that values $\Delta k_{Hf} = 0.01$, $k_f = 1.1$ with ISF = 12.5; then $m = 8 \times 10^{-4}$, $n = 8 \times 10^{-3}$.

Eq. (22) and (23) and the chosen values of m and n are approximate, but they reflect qualitatively the data obtained by GASPROM of Russia concerning the changes in the performance of the gas turbines of different output operating at the same station, due to fouling. No special experiments on this problem have been carried out up to now. These coefficients will be defined more precisely in the process of accumulation of experimental data.

The developed mathematical model of the progressive fouling has been adapted to the calculation of performance of the model axial compressor GTE-150 LMZ.

Design parameters of the compressor under the initial condition P^*_{in}=101.340 kPa, T^*_{in}=288 K are G = 35.47 kg/s, n_0 = 12420 rpm, π =12.5, η_{ad}=0.86, Dc=0.5712 m, $\bar{r}_h$=0.422, ISF=5.72.

As it can be seen in this case Δk_{Hf}=0.5% and k_f=1.05. Calculation results of the compressor performance are shown on Fig. 3, where $\bar{n} = n / n_0$, $\bar{G} = G / G_0$.

In accordance with the assumed number of fouled stages (z_f=6), the calculation included six intervals, during the last interval the characteristic of the first stage was calculated with $\Delta k_{Hf}^{VI} = 6 \times 0.005 = 0.03$, $k_f^{VI} = 1.05^6 = 1.340$, but the sixth stage characteristic was calculated with Δk_{Hf}^{I} =0.005, k_f^{I} =1.05. As one can see from Fig. 3 when the first six stages are fouled (after the sixth interval of progressive fouling), this results in a 4.5% reduction in inlet mass flow, 4% reduction in pressure ratio and 2% reduction in compressor efficiency.

In connection with the development of axial compressor and turbine diagnostics, the method of estimation of engine performance deterioration under fouling and erosion damage gains more ground. The above described method can be used for this purpose. It allows one to establish not only a tendency but also a character and a degree of a change of the main parameters of a compressor and a gas turbine as a whole under fouling conditions; it helps the operators to take a decision about the cleaning (washing) of the compressor.

There are also some recommendations concerning the regularity of cleaning, based on the operational experience.

It was shown in (Schurovsky and Levikin, 1986) that the behaviour of compressor fouling is similar to the exponential law: after 1×10^3 - 2×10^3 operational hours the engine performance stabilisation was noted because there took place the stabilisation of a thickness and a form of deposits. An empirical formula was proposed to estimate the relative output change under fouling:

$$\Delta k_{Ne} = a\,(1 - e^{-b\tau}) \tag{24}$$

where a=0.07, b=0.005 1/hour, $k_{Ne}=N_e^{\tau_0}/N_{e0}$, τ - operational time of an engine before cleaning in hours.

Eq. (24) has been obtained on the base of the measurements of the actual output of the following gas turbines: GT-700-4, GT-700-5, GT-6-750, GTK-10, GPA-C-6.3, GTK-10I (MS 3000 GE) that have operated for 100 to 1000 hours without cleaning at the compressor stations. After the cleaning the output has been restored to the level of Δk_{Ne} =0.965-0.985 in dependence of the time period between cleanings of 300 or 100 hours.

The cleaning (washing) of compressor allows one to restore the lost output to some extent; in some cases the initial output can be restored almost completely. The degree of restoration depends on the period of cleaning: long intervals between cleanings are not effective. Based on the field data a formula was worked out in (Schurovsky and Levikin, 1986) to calculate the mean integrated coefficient k_N when cleanings (washings) took

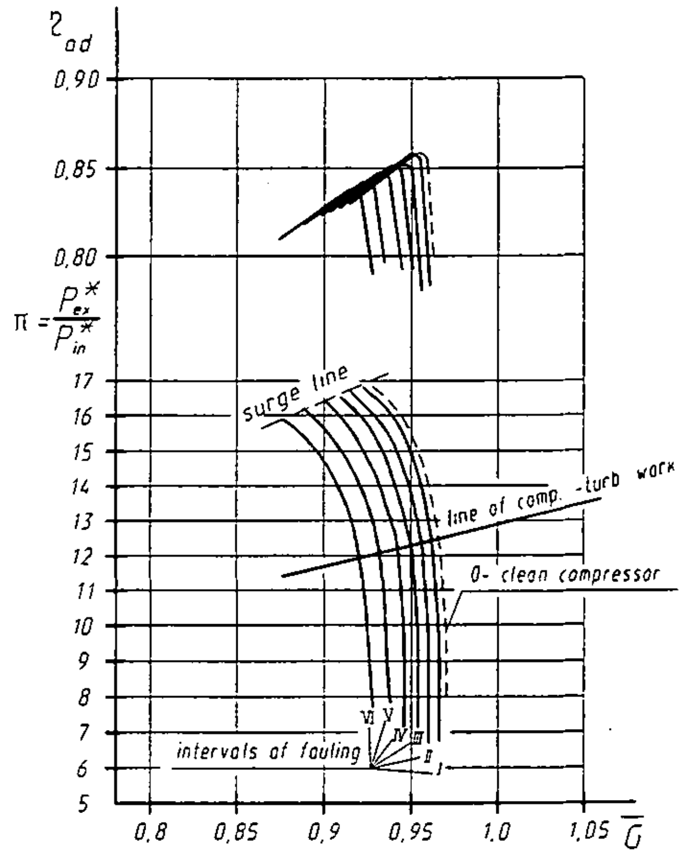

Figure 3
Design Performance of the Model Compressor GTE-150 LMZ under Fouling Conditions ($\bar{n}$=1.0)

place:

$$k_{Ne} = 1 - a\left(1 - \frac{1 - e^{-bT_0}}{b\,T_0}\right) \tag{25}$$

where T_0 is a time period between cleanings in hours, a and b are the same as in Eq. (24).

It was at the time recommended to make cleaning every 200-300 hours to maintain high values of the output coefficient of technical condition (k_{Ne}=0.97 when T_0=250 h).

Regular washings (cleanings) of compressor blading are the most effective way to resist fouling. They allow one to maintain output and efficiency at a high level during operation. Wet cleaning is the most effective means when it is carried out using the optimised regime of on line and off line compressor washing. The important practical results in applying compressor cleaning technique have been demonstrated by Turbotect, see (Nicholson, 1990), (Stalder and Oosten, 1994). This technique was tested on a large gas turbine at Utrecht, the Netherlands. A washing solution of 20% T-927, an organic solvent based cleaner and demineralized water was used. It does not cause pollution, it is safe in service for the operational personnel and it does not cause corrosion of engine elements (compressor, gas turbine, combustion chamber, regenerator). A compressor wet cleaning system comprises 30 on line and 7 off line injection nozzles. During experimental work the optimal arrangement of the nozzles in the intake of a compressor was found. The nozzles are arranged so as to inject a fine atomized spray into the air stream which will mix and will be carried uniformly into the compressor bellmouth. The period of on line and off line washing was determined with minimum consumption of the washing solution, electric power, labour intput.

SUMMARY

1. This paper describes the main factors producing an effect on the sensitivity of an axial compressor to fouling:
- given aerodynamical and geometrical similarity, a compressor of a smaller size (a model) is more sensitive to fouling than a full-scale one;
- sensitivity to fouling of the axial compressor stage increases when the stage head grows;
- the degree of the particle deposition on the blades increases when the angle of attack grows.

2. The mathematical model of progressive fouling of the compressor flow path has been developed using the stage-by-stage calculation method based on the test results of cascades in a wind tunnel. Calculation of the model compressor GTE-150 LMZ performance under fouling conditions has been carried out. Fouling up to the 6th stage of the compressor causes the reduction in the mass flow by 4.5%, pressure ratio - by 4% and efficiency - by 2%.

The index of compressor sensitivity to fouling is suggested in the form:

$$ISF = \frac{G\, c_p\, \Delta T^*_{stg}}{(1-\bar{r}_h^2)\, D_c^3}\, 10^{-6}$$

When two compressors with different geometrical dimensions and different performance are compared, a greater ISF corresponds to a greater sensitivity of a compressor to fouling.

3. For a new installation, it is recommended to select the most appropriate and effective air inlet filtration system for the best cleaning efficiency at the lowest additional losses of output and heat rate. Climatic conditions and plant environment are to be considered.

4. As fouling of the compressor cannot completely be avoided, thus, despite the use of a very effective air inlet filtration system and in order to improve the gas turbine unit operational safety and profitability as well as to reduce the cost of energy production, it is recommended, on the basis of analysis of published data, to adopt an optimised regime of on line and off line washing in the preventive turbine maintenance program. Appropriate nozzle design and arrangements are available.

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