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Life Estimation of First Stage High Pressure Gas Turbine Blades

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Based on very early occurring ruptures found in the first stage high pressure turbine blades of a turbo reactor in a local aviation company, this study has the aim to determine their safe life. The first stage blades are subjected to simultaneous action of gas pressure coming from the combustion chamber, centrifugal forces in the case of the rotor blades and to important temperatures transients, which progress in a very aggressive environment due to hot gases. These combined parameters cause a high state of stress involving several complex mechanisms of damage, such as: fatigue caused by mechanical stress fluctuations, thermo-mechanical fatigue caused by temperature variations and corrosion caused on the stressed elements. Life cycle determination asks for stress evaluation of blades regarding several variables which are approached deterministically in the study. Heat exchange between combustion gases and metal blades is considered. The total stress on two kinds of blades is calculated by the addition of the thermal effect and the mechanical loading. The stress cycle is then calculated for different steps of the engine function during the operation by considering the variation of the thermal and the mechanical properties of the system. Safe life determination is done by two different approaches: the safe life approach by the initiation model and the damage tolerance approach considering the defect growth mechanics and considering the pitting corrosion effect. The calculation is applied for stator and rotor blades of an aero engine high pressure turbine made of NI 738. Since these parts are high risk components from the point of view of potential failure consequences, the risk is assessed as well.
The results obtained are studied to determine the solution to the problem, and to propose a safe decision to be taken about the design or maintenance procedures.

Key words: **turbine, gas turbine, turbine blades, fatigue, creep, pitting corrosion, life cycle, reliability prediction.**

Introduction

HIGH pressure turbine blades operate in high stress environments. Very elevated and varying temperatures and dynamic conditions due to variable pressures and rotating blades cause the material to be exposed to complex mechanisms of damage. The damage mechanisms responsible for early ruptures of blades found in the studied motor, regarding the surface aspect of rupture, present a fatigue mechanism [1-4], and little brittle fractures on the surface also seem to be characteristic of creep. The presence of corrosion pits is obvious (Fig.1). The shortened operation time shows that fatigue is aggravated by the creep mechanism in the elevated temperature environment.

To calculate safe life, it is necessary to find the thermo mechanical stress state, with the varying thermal and mechanical properties, regarding temperatures. This state is defined in this work by using the Brayton cycle and the strength material rules. Adding the centrifugal effect to the pressure effect, mechanical varying stresses are found. The temperature variations during normal operation induce thermal deformation and consequently thermal varying stress in the blade material. The two stresses added constitute the thermo mechanical stress cycle affecting the HP turbine blades. This loading state is a non reversible cycle, with long time at a very high constant temperature that leads to creep. Lives are calculated to fatigue by the Manson - Coffin equation, and to creep by the Larson Miller model. Complex damage accumulation is evaluated by two accumulation models; the Miner linear model and the Chaboche model. The safe lives are found and compared. The case study is done for NI based superalloy. The safe life calculations with the standard coefficient of security are evaluated. The evaluation of the risk of failure according to its probability of occurrence and the consequence of failure gives serious information on the level of the risk of that engine structure.

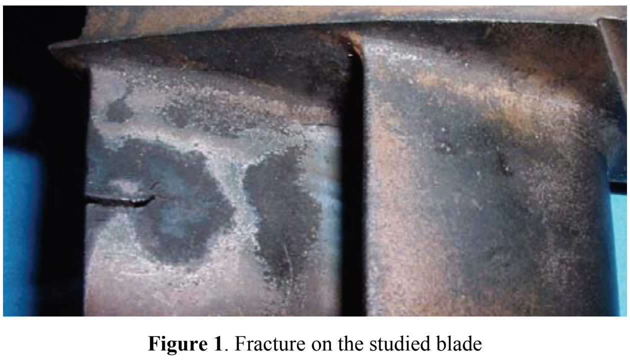

Figure 1. Fracture on the studied blade

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High pressure turbine blades

The life of aircraft high pressure turbine blades is a function of their service resistance to failure; this depends mostly on the operating environment and material properties. To define safe life of HP turbine blades, it is essential to understand their function and the resulting degradation phenomena.

The First Stage High Pressure turbine receives very hot gases from the combustion chamber at high pressure. Gas relaxation provokes shaft rotation at a very high speed.

The high pressure blade operates in a very complex environment – among very hot, dirty corrosive gases at significantly varying pressures. The rotor blade supports also the centrifugal force of the rotating shaft. During operation the temperature rises to the maximum, then stabilizes at a high temperature level until the stopping of the engine when it goes back down to the reference pressure. Severe mechanisms of deterioration take place in turbine blades; the most important ones are fatigue, creep, corrosion and thermal fatigue. The study tried to carry out the lives for the following mechanisms:

Fatigue:

Fatigue failure is caused by the application of variable loads to an element producing fluctuating stresses. These stresses lead to progressive material deterioration until failure occurs even if stresses are lower than the yield stress [1, 4, 5, 6].

Creep: The application of steady state stress at high temperatures can cause crack nucleation and growth [5, 6]**.**

Thermal Fatigue: The most severe stresses that turbine blades encounter are those induced by extreme temperature gradients and rapid thermal transients. These thermally induced stresses, combined with high mechanical load, give localised high transient strains producing thermo mechanical fatigue (TMF) cracking in the blade. As in the pure fatigue, the crack tends to be initiated on the component surface [5, 6].

Corrosion: Corrosion is an extremely complex damage mechanism, which depends strongly on material and environmental conditions. In the case of HP blades, material is exposed to erosive micro particles from combustion gases, breaking the protective oxide layer on the surface of the blade. Unprotected material is exposed to corrosive environment leading to nucleation of corrosion pits. The microscopic damage will grow as a consequence of the extreme service conditions of the HP blade [5, 6].

Load determination

To determine the stresses involved in the high pressure turbine, the stream of temperature and pressure must be computed.

Properties of the Brayton thermo dynamical cycle are [7] used to define the temperature cycle and the gas pressures. The chart defining the pressures and the temperatures in different steps of the gas evolution in the engine, as well as the thermal conductions to blade metal respecting cooling technologies [7, 8] is represented in Fig.2.

The rotor blades and the stator guide vanes are then considered respectively as a fixed free beam and a fixed - fixed beam. Moments and shear forces are defined by classical strength material rules, using pressure as a load defined by thermodynamic calculations [7], and material properties depending on temperatures.

Mechanical normal equivalent stress, regarding normal bending stress, centrifugal effect stress and shearing stress, is evaluated by the VonMises multiaxial criteria. The centrifugal effect is considered only for the rotor blades [7, 8, 9].

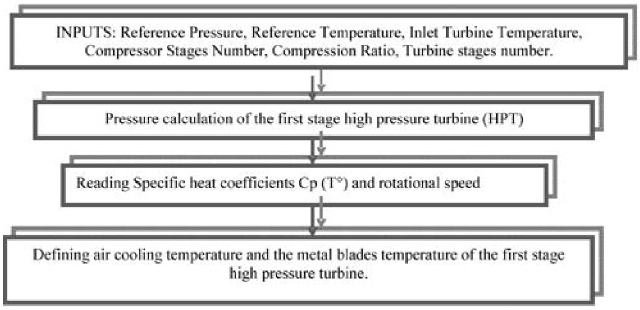

Figure 2. Flow Chart of the procedure calculating pressures and temperatures

Every functioning of the plane causes a temperature cycle in the engine. This leads to a thermal deformation, and consequently to the thermal stress cycle. The mechanical effects added to the thermal stress cycle give the total thermo mechanical stress cycle.

$$\sigma = \sigma_{mec} + \sigma_{th} \qquad (1)$$

The total stress is a non reversible cyclic stress with a mean value σ_m, and an alternative value σ_a.

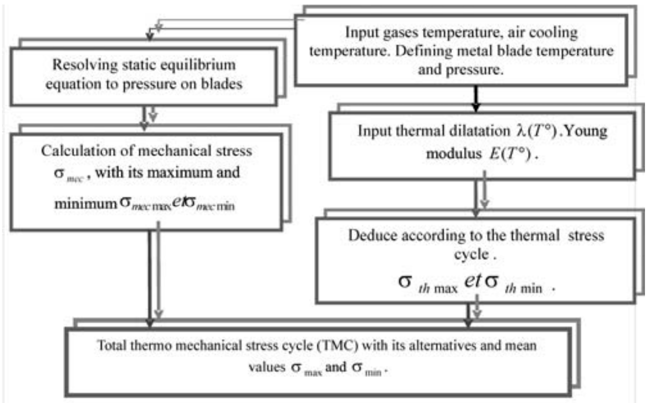

Figure 3. Determination of the thermo mechanical stress cycle for a functioning procedure.

Life determination

In order to find the safe life of the turbo reactor first stage high pressure turbine, it is necessary to adopt a model for the evaluation of the life. The life prediction, more generally the evaluation of the behaviour of a rupture mechanism, or several mechanisms at the same time, is a significant task in order to ensure the reliability of the system. The cycle is non reversible to deal with the mean stress; equivalent reversible stress is calculated with the Goodman rule [1-4].

Life prediction for fatigue (N_f):

N_f is defined in this work by the Manson Coffin formulation. Manson Coffin [1, 2, 4] established that the alternative strain is related to the number of stress cycles to rupture, by the equation:

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma_f}{E}(2N_f)^b + \varepsilon_f(2N_f)^c \qquad (2)$$

E is the elasticity modulus, σ_f material strength, and b and c for the majority of materials are equal to -0.12 and -

0.6, respectively [1, 4].

Life prediction for creep (*Nc*):

Creep for metal is described by the Larson Miller parameter (LMP). T is the temperature in *K* degrees, N_C is the time to creep rupture in hours. The LMP *C* coefficient is taken equal to 20 [7].

$$LMP = T\,\frac{Log(N_C) + 20}{1000} \tag{3}$$

Model for fatigue-creep interaction:

Many approaches have been developed these last years to predict the safe life of materials subjected to high temperatures.

To consider the interaction between fatigue and creep damage, several rules [1] of damage accumulation could be used [7]. This accumulation can be considered as linear or non linear. In this work two damage accumulation models are used: the linear Miner rule eq. (4) and the non linear Chaboche rule eq. (10).

$$\sum \frac{n_i}{N_i} + \sum \frac{t_j}{t_{Rj}} = D \tag{4}$$

It is considered that during a cycle, the damage of creep passes from the damage D_0 to D_1 and that the fatigue damage increases at the end of the cycle from D_1 to D_2. Two equations (6) and (7) give respectively the creep and fatigue interactive damage for one cycle:

$$\frac{1}{N_c} = (1 - D_0)^{K+1} - (1 - D_0)^{K+1} \tag{5}$$

$$\frac{1}{N_f} = \left(1 - (1 - D_1)^{\beta+1)}\right)^{1-\alpha} - \left(1 - (1 - D_2)^{\beta+1}\right)^{1-\alpha} \tag{6}$$

The coefficients *α*, *β* and *k* are the material data, defined experimentally [10]. In the two cases, the damage is complete when damage accumulation reaches the unity. Then the rupture becomes certain. We can then predict the number of cycles to failure when:

$$\sum D_i = 1 \tag{7}$$

Model of pitting corrosion fatigue life description

The total fatigue life may be represented by the summation of four phases [8, 9, 10]:

Corrosion pit nucleation stage: The first stage of the model is related to the electrochemical process which results in the nucleation of the corrosion pit. Our model assumes that this time *tin* is calculated for the initiation creep problem and divided by tree as found in empirical research [10, 11]. The model is derived from the Shi and Harlow's reliability approach [10, 12].

Pit growth stage: This stage concerns pit growth and constituent particles and involves the electrochemical process. In this model, the pit is assumed to grow at a constant volumetric rate according to the Faraday's law. This law gives *tin g*, time to pit growth which is a function of molecular weight, valence of the material, Faraday constant, activation energy, universal gas constant, absolute temperature and the pitting current. The pit is assumed to be a half prolate spheroid in an infinite plate.

Determination of transition 1: This third stage considers the transition from the pit growth to short fatigue crack growth where the effect of stress concentration factor is taken in account. The transition is assumed to occur when:

$$(\Delta K)_{pit} = (\Delta K)_{crack} \tag{8}$$

Applying the Faraday law to define the time of propagation and resolving the equation of transition to define the critical pit size gives the transition. In this work the transition size is fixed to constant 1.53e -3 µm.

Short crack growth: The fourth stage involves chemical and micro structural factors and their interactions, the Paris law with a specific parameter is used in this stage to define the *tccg,* which is a function of the crack transition to the classical long crack growth.

Determination of transition 2: In the analytical approach the transition size from short crack growth to long crack growth equals the rate s of growth of the short crack and long crack. Some research gives a range to this value from 5 to 5mm. In our application this value is 1mm.

Long crack growth: The classical Paris law is used in this stage [10], dependant on the material constant for long crack growth and the stress intensity factor; the stress concentration factor is taken for a crack in a hole for an infinite plate regarding conclusions of research [12] which identifies that corrosion pits take place in holes of blades. The time to long crack growth *tlcg* is calculated to the final critical size for long crack growth. This value defines the limit to be considered for repair.

Fracture: It considers a crack size unacceptable or in need to be repaired, lower than the crack size when real fracture occurs.

The time is evaluated for fatigue corrosion life as:

$$t_f = t_{in} + t_{ing} + t_{scg} + t_{lcg} \tag{9}$$

Applications

The model of life prediction described in this work was applied to find the safe life of first stage turbine blades for one engine of a national flight company. Blade geometrical data and engine properties are given in Tables 1 and 2. The coefficients versus temperatures as specific heat $Cp(T°)$, thermal dilatation $\lambda(T°)$ and Young modulus $E(T°)$, and are found in the literature for the IN 637 [7,].

Table 1. First Stage Blades High Pressure Data

	NGV	Rotor blade
Cord (w_p in m)	0.02986	0.02745
Length (L_P in m)	0.03216	0.03216
Ray (R in m)	0.2	0.2
Thickness (e_P in m)	0.0005	0.0005
Revolution (N in RPM)	0	13820
Density (ρ in Kg/m³)	8110	8110
Number of blades	43	64

Table. 2. Thermodynamical inputs

P(MPa)	$T(^\circ C)$	$T_{gaz}(^\circ C)$	$T_{cool}(^\circ C)$
1.296	932	932	323.65
	1071	1071	323.65
P'(MPa)	$T'(^\circ C)$	$T'_{gaz}(^\circ C)$	$T'_{cool}(^\circ C)$
0.685	758.87	880.45	237.945
	880.60	921.60	239.175

The results of the program made for this calculation, are given in Figures 4 and 5 as graphs. The prediction done is not very different for the two damage accumulation models. At long life values the Chaboche's model is more conservative (32000 hours of flights). But for short life values the Miner model for damage accumulation is more simple and enough accurate. The Chaboche's model needs to define the material coefficient.

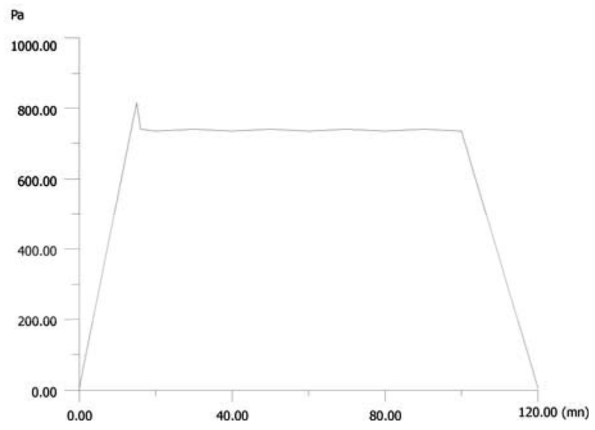

Figure 4. Stress cycle on rotor blade (stresses in *Pa* vs. time in minutes)

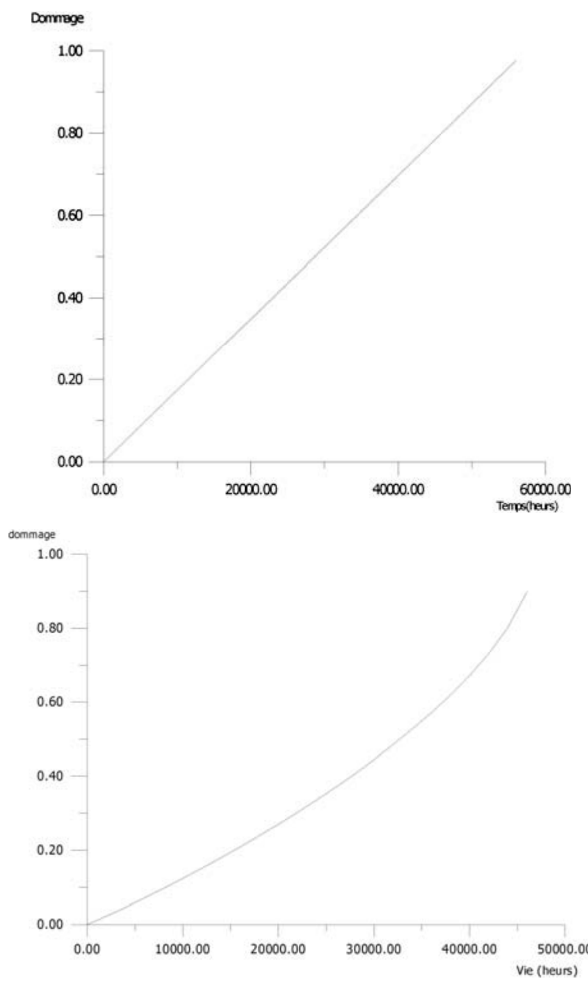

Figure 5. Damage curve versus time in minutes Miner accumulation for the rotor (a) Chaboche model for the rotor (b).

Table 3. Life predicted to crack initiation

	Linear model	Non linear model
Stator blades	56000 hours	46000 hours
Rotor Blades	32000 hours	30000 hours

Table 4. Life in years according to smaller and greater security coefficients

Time of utiliza-tion in years	Safe Life calculated by the model	Safety factor 4 (min)	Safety factor 20 (max)
stator	12.60	3.15	0.63
rotor	8.22	2.055	0.41

The life found through the damage tolerance model is above the safety norms using the coefficient of security of 4 on life for an element of a jet engine. At a damage crack length of 1 μm the life is 70000 cycles. Division by the deterministic safety factor gives the result of 17500 cycles, i.e. 9.5 years. But it shows that we are limit of slow crack growth (Fig.6).

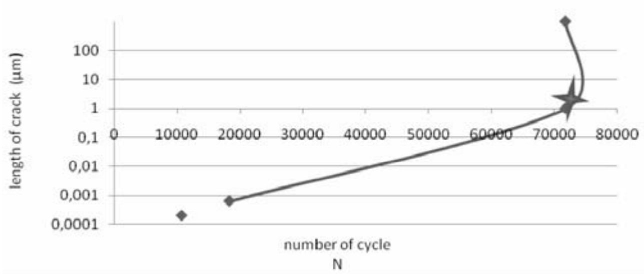

Figure 6. The length of the damage evolution during functioning according to the damage tolerance model

A simple risk and critical condition evaluation for the case study:

The engine studied is a civil aviation aircraft engine. According to all standards [9-11] it is necessary to minimize the occurrence of this kind of failure. The failure consequences could cause human injuries or the loss of the flight. We assume that a probability of this worst event is 0.01, i.e. one case in a hundred, and it is a rough estimation.

The life calculations have a very important level of uncertainties, because of the lack of data, the difficulties of the empirical investigation and the vagueness of the models. We will use a probabilistic approach to evaluate the probability of rupture. We consider the life to rupture to be normal distributed [14, 16].

The probability of rupture for the blade is taken for a mean of the maximum life found 28000 and a variance of 10000. The probability of rupture cycle necessary for a safe life (60000cycles) [15] is then the inverse cumulative distribution function of life: Pr is 0.9997 that must be lower than 10 exp (-6) [14-16].

$$\text{RISK} = \text{Probability Failure*Consequence of failure} \qquad (10)$$

For our case study risk is equal to 10^{-2} and might be for such a component in the average of 10^{-6} for safe life. The maintenance procedure for such element engine could be done through the procedures in [14-16].

Conclusions

This work calculated the safe life of an element of a turbo reactor. It allows studying thermal evolution in the jet engine and evaluates the thermo mechanical stress cycle. Life to creep, fatigue or their interaction could be determined. It can study the evolution of damage to creep and fatigue interaction. The two models are similar regarding life prediction. The Miner approach is simpler and the Chaboche model is more conservative and takes into account the variations near the damage initiation. They could be used to predict lives for design or to be integrated in maintenance services, for a reliable maintenance.

The blade engine presented in this study had a serious problem regarding one secure function; the probably of rupture of 0.9997 is very important. According to the damage tolerance approach for pitting- fatigue in this same studied blade, by the deterministic approach, the damage is 10exp(-7) m for 60000 cycles. This study contains several uncertainty sources. They could occur in the material parameters or in the functional parameters that might be considered according to the structural integrity standards. But for more objective conclusions, deeper investigations must be done, in the perspective of a more reliable design for the constructor and a more reliable maintenance strategy for the user company.

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Procena veka lopatica prvog stepena gasne turbine visokog pritiska

Posle pojave prevremenih lomova koji su se javili kod lopatica prvog stepena turbine visokog pritiska određenog tipa turboreaktora u jednoj lokalnoj avio kompaniji, usledila su istraživanja u cilju određivanja njihovog sigurnog (pouzdanog) radnog veka. Lopatice prvog stepena turbine visokog pritiska su tokom rada izložene simultanom delovanju pritiska gasa iz komore za sagorevanje, centrifugalne sile u slučaju rotorskih lopatica, velikim temperaturnim promenama, kao i agresivnoj radnoj sredini. Kombinacija svih navedenih činilaca izaziva vrlo kompleksno naponsko stanje lopatica kao i mogućnost pojave višestrukih mehanizama oštećivanja: zamora izazvanog fluktuacijama u mehaničkom naponu, termo-mehaničkog zamora usled temperaturnih promena i korozije naponski opterećenih delova. Da bi se odredio radni vek u navedenim uslovima, neophodno je proceniti napone kojima su lopatice izložene uzimajući u obzir nekoliko promenljivih koje se u radu tretiraju deterministički. Razmatran je prenos toplote između sagorelih gasova i metala lopatica turbine. Izračunat je ukupni napon na dve vrste lopatica imajući u vidu termičke efekte i mehaničko opterećenje. Naponski ciklus je zatim izračunat za različite faze rada turboreaktora uz varijacije termičkih i mehaničkih osobina. Procena sigurnog radnog veka je izvršena primenom dva pristupa: inicijalnog modela i pristupa sa tolerancijom oštećenja uzimajući u obzir mehanički rast oštećenja i rast korozionog pita. Proračun je primenjen i na statorske i na rotorske lopatice turbine visokog pritiska turboreaktora koje su izrađene od čelika NI 738. Nađeno je da su ovo komponente visokog rizika, tako da je procenjen i rizik sa aspekta potencijalnih posledica od loma.

Dobijeni rezultati su razmatrani u cilju rešavanja problema i donošenja sigurne odluke sa aspekta procedure održavanja ali i konstrukcijeskog rešenja.

Ključne reči: **turbina, gasna turbina, lopatice turbine, zamor materijala, puzanje, piting korozija, radni vek, prognoziranje pouzdanosti.**

Оценка срока службы лопасток первой степени газовой турбины височного давления

После появления преждевременных изломов появившихся у лопасток первой степени турбины высокого давления определённого типа турбореактора в одной местной авиакомпании, наступили исследования с целью определения их надёжного срока службы. Лопастки первой степени турбины высокого давления в течение работы подвергались одновременному действию давления газа из камеры сгорания, центрифугальной силы в случае лопасток ротора, большим перепадам температур, а в том числе и агрессивной рабочей среде. Комбинация всех приведённых факторов вызывает очень комплексное напряжённое состояние лопасток, а в том числе и возможность появления многократных механизмов повреждений: усталости вызваной флюктуациями в механическом напряжении, термомеханической усталости вследствие температурных перепадов и коррозии напряжёно нагруженных составных частей. Чтобы определить срок службы в данных условиях, необходимо оценить напряжения которым лопастки подвергаются, учитывая несколько переменных в работе принятых в роли определителя. Здесь рассматривана взаимная передача теплоты между сгорелыми газами и металом лопасток турбины. Здесь расчитано совокупное напряжение на два типа лопасток имея в виду термические эффекты и механическую нагрузку. Напряжённый цикл потом расчитан за различные фазы работы турбореактора, с переменами термических и механических свойств. Оценка надёжного срока службы сделана с применением двух подходов: первоначальной модели и подхода с допуском повреждений, учитывая механический рост повреждения и рост эффекта питинг коррозии. Расчёт применён и на лопастки статора и на лопастки ротора турбины высокого давления турбореактора, выработаные из стали НИ 738. Обнаружено, что эти комплектующие части высокого риска и в том числе и определён и риск с аспекта возможных последствий от излома. Полученные результаты рассматриваны с целью разрешения проблем и приношения надёжного решения с аспекта процедуры обслуживания, но и конструкционного решения.

Ключевые слова: турбина, газовая турбина, лопастки турбины, усталость материала, ползучесть, питинг коррозия срок службы, обсуждение надёжности.

Estimation des durées de vies des aubes du premier étage d'une turbine à gaz haute pression

A la suite des défaillances précoces enregistrées par les équipes d'entretien, des aubes du premier étage d'un turboréacteur de la compagnie aérienne nationale, cette étude concerne la détermination du temps de fonctions sures de ces éléments. Le premier étage de la turbine est soumis simultanément aux effets des gaz émanant de la chambre de combustion, à des variations de températures très élevées, à un environnement très agressif ainsi qu'aux effets de la force centrifuge dans le cas des aubes rotor. Ces paramètres combinés induisent des contraintes très élevées, conduisant à des mécanismes de dommages complexes, dont principalement la fatigue due aux effets des contraintes mécaniques fluctuantes, la fatigue thermo mécanique causée par la variation de température et la corrosion due à la réactivité du métal avec les éléments contenus dans les gaz de combustion. La détermination des durées de vies nécessite l'estimation des contraintes dans les aubes par rapport à beaucoup de variables qui sont considérées toutes déterministes dans cette étude. L'effet du transfert de chaleur autre le métal des aubes et les gaz chauds est pris en considération. Les contraintes totales sont calculées pour les deux types d'aubes en tenant compte de l'effet thermique et de l'effet mécanique. Le cycle de contrainte subi par les aubes est retrouvé pour les différentes étapes d'un vol de deux heurs, en tenant compte de la variation des propriétés physiques avec la température. Les durées de fonction sures sont retrouvées par deux approches différentes. La détermination des vies sures à partir du modèle de l'initiation du dommage et à la durée de vie totale jusqu'à la rupture totale grâce au modèle de tolérance du dommage tenant compte du mécanisme de la progression du défaut et de l'effet de la corrosion par piqûres. L'application est faite pour des aubes stator et rotor du turbo réacteur dont le matériau est le NI 738. Ces éléments sont des éléments potentiels pour la sûreté fonction du moteur. Le risque induit par une défaillance est évalué. Les résultats obtenus sont étudiés afin de prendre une décision soit à une nouvelle conception ou à des procédures de maintenance appropriées.

Mots clés: **turbine, turbine à gaz, aube, fatigue des matériaux, fluage, corrosion par piqûres, durée de vie, prédiction de sûreté.**