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

97-GT-199

TRANSIENT PERFORMANCE ANALYSIS OF A TWO SPOOL TURBOFAN FOR LIFING AND HANDLING PURPOSES

Petros Kotsiopoulos

Hellenic Air Force Academy
Chair of Propulsion Systems
Dekelia Air Base, Attiki, GREECE

Pericles Pilidis

School of Mechanical Engineering
Cranfield University, Beds MK43 0AL, U.K.

Dimitrios Adamopoulos

Hellenic Air Force Research Center (KETA)
A. Glyfada, Attiki, GREECE

1 ABSTRACT

The handling of a gas turbine and its life usage are closely related. As the handling capability required from an engine increases so does the life consumption. Faster transients and response require higher temperature levels, this reduces component lives. The phenomenon investigated here is thermal fatigue and the main focus is in the hot section of the engine, specifically the turbine blades.

To make a quantitative assessment of the reduction of life due to the requirement of a faster transient response a simulation code of a two spool turbofan engine has been developed. It has a routine that estimates the High Pressure Turbine blade metal temperature. This temperature has been used to estimate the reduction of life loss when fast shaft accelerations are required.

The main conclusion of this work is that, as expected, fast transients consume life faster. The present investigations suggest a quantitative level of thermal life reduction of approximately twenty five to thirty percent when comparing a fast acceleration of two to three seconds with a slower one of eight to ten seconds.

2 NOMENCLATURE

α	Linear expansion coefficient
$\Delta\varepsilon$	Strain Range
Δt	Time interval duration
Bi	Biot number
$c_{p\,met}$	Blade metal specific heat
D	$\log_e(100/100\text{-}RA)$
dT_{bl}	Metal temperature increment
E	Young's modulus
θ	Power lever Angle
h	Heat Transfer Coefficient
I	The ranking of a particular time interval
n_{cool}	Blade Cooling Efficiency
N_f	Cycles to failure
Nu	Nusselt Number
ρ_{met}	Blade metal density
RA	Percentage reduction in area at fracture
Re	Reynolds Number
S_{bl}	Blade surface
ss	Steady State
t	Time
T_{bl}	Blade metal temperature

Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
Orlando, Florida — June 2–June 5, 1997

T_{c2}	Temperature of cool. air leaving the blade
T_{c2}	Temperature of cool. air entering the blade
V_{bl}	Blade metal volume
W_c	Cooling mass flow rate

List Of Subscripts

bl	blade
c	coolant
c1	Coolant at the passage entry
c2	Coolant at the passage exit
f	failure
g	gas
init	Initial
inter	At end of time interval
met	metal
ult	ultimate

3 INTRODUCTION

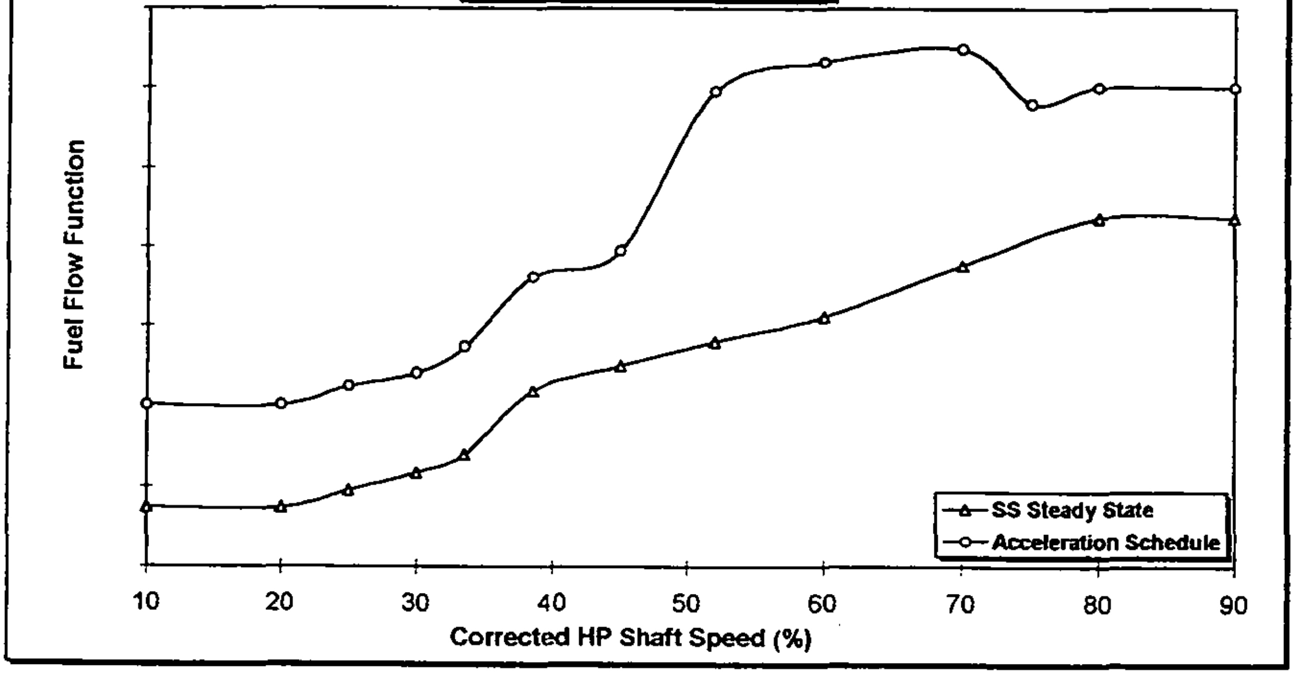
The two main mechanisms responsible for the aging of gas turbine components are fatigue and creep. Creep life consumption is associated with stressing components under high temperature for prolonged periods of time while fatigue is linked with cyclic stressing. This is due either to thermal gradients within the component's body, called thermal fatigue, or to changing centrifugal stresses of the engine components. The latter is further subdivided into Low Cycle Fatigue (LCF) and High Cycle Fatigue (HCF) depending on the frequency of the loads.

In the model described here, thermal fatigue has been calculated, during an acceleration, as a function of the difference between the lowest and highest temperatures of a mission. In the present case the mission is assumed to consist of a cycle from start-up to idle plus an acceleration to full power. Therefore the lowest cycle metal temperature is ambient before the engine has been started and the maximum metal temperature is the highest encountered in the transient under investigation. Because of gas temperature overshoots, the maximum transient metal temperature is usually greater than the maximum steady state blade metal temperature. This overshoot of the metal temperature is used to estimate the reduction in life of the blade.

Many transients have been examined, some fast others slow. The relative life loss due to thermal fatigue has been calculated for all these, using a slow acceleration from idle to full power as reference. This acceleration is demanded by moving the power lever from its initial to its final setting in 8 seconds, i.e., its power lever transient times is eight seconds. The first part of the cycle, from startup to idle, is assumed to be the same for all transients, fast or slow so it will have the same effect in all cases studied.

In practice, a real mission will consist of many such cycles of varying amplitudes in power setting. The method described here could be applied to any set of throttle movements, however the aim of the present work is to assess the effect of rapid transients on thermal fatigue life. The mission or cycle selected is adequate for this purpose.

Figure 1. Fuel Scheduling



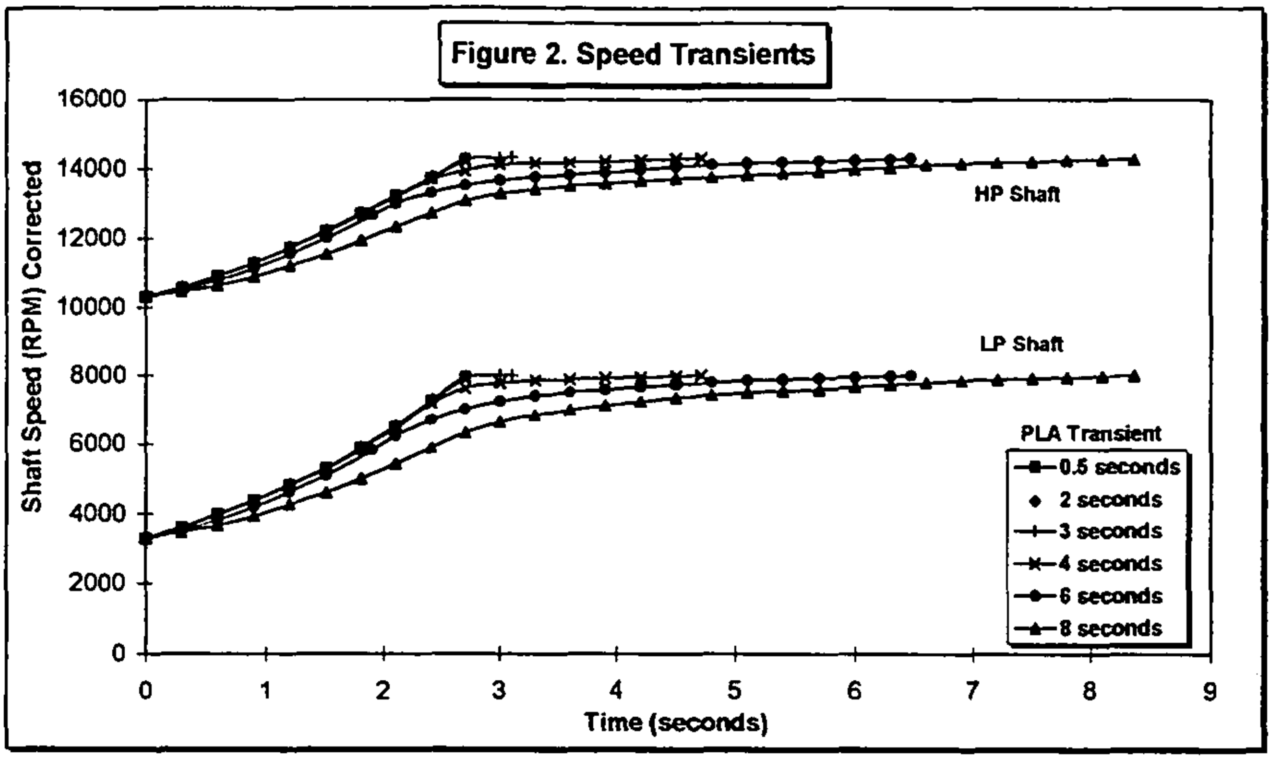


Figure 2. Speed Transients

4 THE PROGRAM 'TRANS'

The calculation has been performed with the use of a code specially developed for this task, it is called 'TRANS'. The program 'TRANS' [Reference 1] consists of a transient code based on the Continuity of Mass Flow (CMF) method and is very similar in its philosophy to those described in Reference 4. The code is written in FORTRAN 77. It has been developed to suit a two spool medium bypass turbofan engine in the inventory of tha Hellenic Air Force. The code is able to model variable geometry compressor configurations and variable convergent-divergent (con-di) nozzles. Tha programma was calibrated in the steady state with known engine data.

Fuel schedules were developed by the authors based on the predicted steady state performance of the engine and the requirements to achieve the acceleration rates that would ensure the simulated engine would respect surge limits and notional temperature limits. An example is given in Figure 1, where two curves are shown. The lower one, labelled 'steady state' shows, for a given corrected speed the fuel flow needed to keep the engine operating in the steady state. This curve arises naturally due to mass and energy conservation within the engine.

The fuel flow function is usually a non-dimensional form of fuel flow that allows the schedule to be valid over wide operating ranges. The higher curve, labelled 'acceleration' shows the maximum fuel flow allowable to accelerate the engine. This maximum fuel flow is determined by safety considerations, such as surge avoidance and temperature limitations. The schedule is programmed in the control system, whether hydromechanical or digital, so that engine variables would determine the fuel flow in conjunction with the power lever angle.

When an acceleration is demanded, fuel flow would be the lower of two values. One value would be the fuel flow demanded by the power lever angle. The second value would be determined by the engina variables (in the case of Figura 1 would be corrected speed) and the fuel schedule.

If the power setting change demanded is small, then it is safe to introduce the fuel demanded by the power lever angle. Otherwise a lower fuel flow, as limited by the fuel schedule, is permitted. Thus engina integrity considerations are respected.

In the case of a slow acceleration, where the power lever is moved slowly, the power lever transient is assumed to take place at a constant rate. That is:

$$\frac{d\theta}{dt} = \text{const}$$

As the shaft speed transient time is divided into time intervals so is the power lever transient. At the end of each interval the power lever angle setting is:

$$dPLA = \frac{d\theta}{dt}\Delta t \text{ and}$$

$$PLA_{inter} = PLA_{init} + I * dPLA$$

During an acceleration the power lever angle is calculated at the end of each time interval. The allowable fuel flow rate is then either the one read from the schedule or that defined by the power lever angle value for steady state. Of the two values, the lower one is always selected, as described above. For fast transients, therefore, the fuel flow is dependent on the schedule, while for slow transients it depends on the rate of change of the power lever angle.

The resulting speed changes for all power lever transients are illustrated in Figure 2 and the gas temperature changes are shown in Figure 3. It can be seen that there are some irregularities in some of the curves. This is because the schedules have not been fully optimised. Further optimisation could be carried out, however because the analysis described here is comparative the effects of the irregularities should have no major effect on the results.

5 BLADE METAL TEMPERATURE CALCULATION

The computer model calculates gas temperatures. From these it is necessary to estimate blade metal temperatures.

In these first investigations the main aim is to establish the magnitude of the increased life consumption of fast transients, rather than to calculate it exactly. Therefore many simplifications have been introduced.

As a first approximation it is assumed that the blade employs single pass convection cooling only. It is assumed to act in a way such that the overall effect on the blade is the same as that of the more complex cooling techniques employed in practice. This approach is acceptable for a first analytical look at the problem. More elaborate turbine blade cooling models need to be developed in the future to obtain a more accurate assessment.

It is assumed that the average heat transfer coefficient is related to the Nusselt number ($\overline{\mathrm{Nu}}$) by the following expression [Reference 2]:

$$\overline{\mathrm{Nu}} = 0.235\mathrm{Re}^{0.64}$$

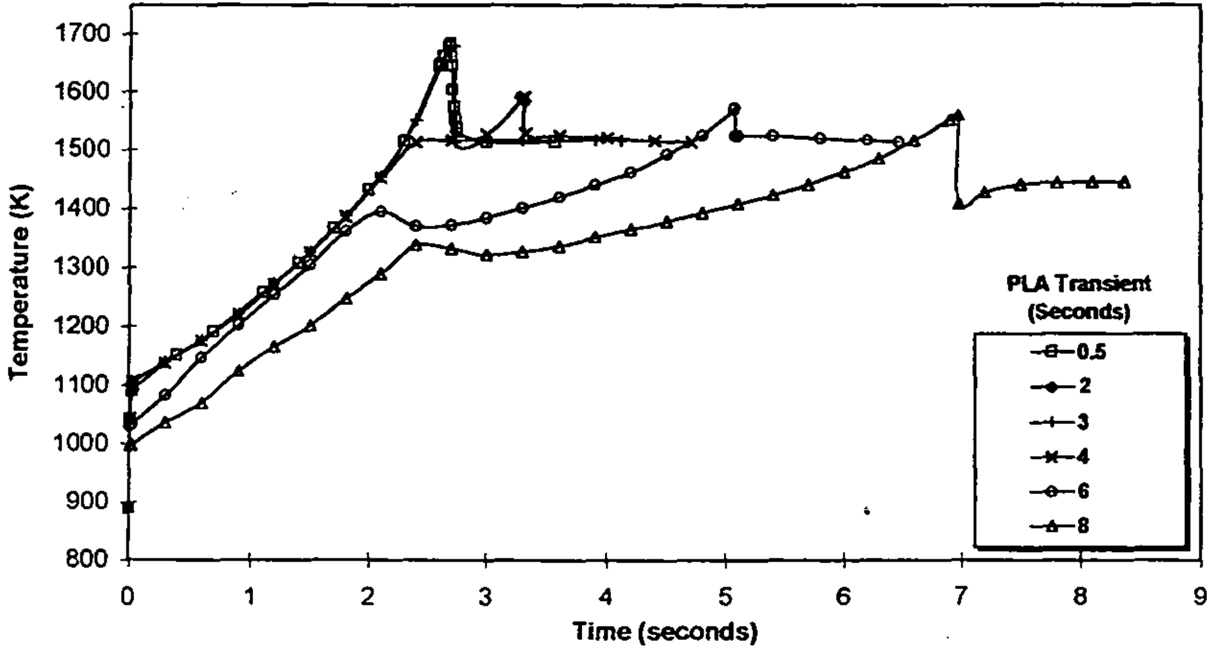
Rearranging the above correlation, an expression for the gas side heat transfer coefficient (h) can be obtained. The heat transfer coefficient is calculated at each time interval along with the temperature of the cooling air leaving the blade.

$$n_{cool} = \frac{T_{c2} - T_{c1}}{T_{bl} - T_{c1}}$$

The cooling efficiency needed evaluation. For two power settings, ground idle and military power, the blade metal temperature was known from test bed data using a pyrometer. At these two power settings, the coolant outlet temperature T_{c2}, can be calculated with the known blade metal temperature and an energy balance. The cooling efficiency, for these two power settings, can then be obtained with the use of the above equation. The value of cooling efficiency for all other power settings was assumed to vary linearly with non-dimensional shaft speed. Typical values calculated in the present analysis were around 10 to 50 percent.

Conduction time across the blade wall, from the gas to the cold side, has been ignored. The Biot number (Bi) was found to be less than 0.1 so the error is estimated to be less than 5%.

Figure 3. Turbine Rotor Inlet Temperature



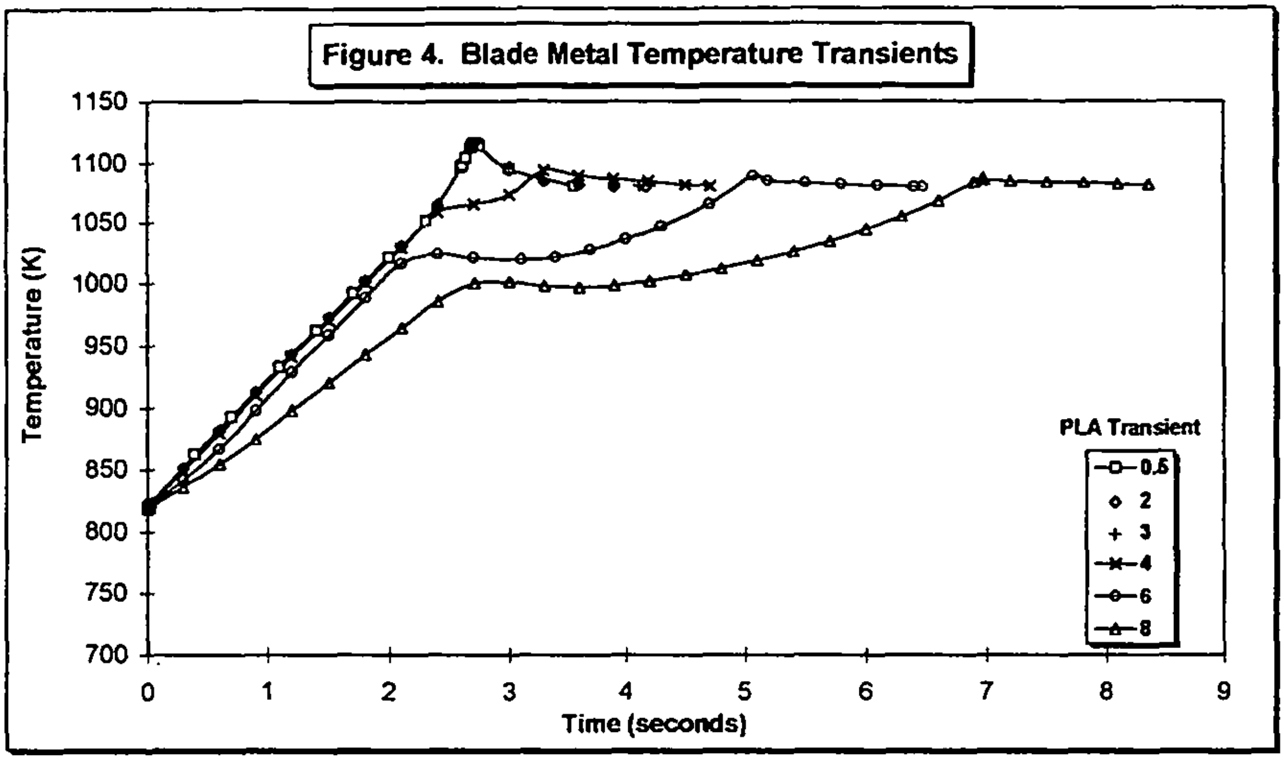


Figure 4. Blade Metal Temperature Transients

At each time interval, the blade metal temperature is calculated using the energy balance equation that can be expressed as:

$$h\, S_{bl} (T_g - T_{bl})\, \Delta t = A + B$$

where

$$A = \rho_{met} V_{bl} C_{pmet}\, dT_{bl}$$

and

$$B = W_c\, c_{pc} (T_{c2} - T_{c1})\, \Delta t$$

The metal temperatures resulting from the gas temperatures shown in figures 3 are in Figure 4.

6 LIFING CALCULATION

Having obtained the blade metal temperatures (Figure 4) as indicated above, the cyclic life, N_f, of the blade was calculated. This was based on thermal fatigue considerations. The life of the blade was calculated for each acceleration under consideration. Manson's expression [Reference 3] has been employed for this:

$$\Delta\varepsilon = \left(\frac{3.5}{E}\right)\sigma_{ult} N_f^{-0.12} + D^{0.6} N_f^{-0.6}$$

The strain range is calculated using the expression for linear expansion for a certain point on the metal:

$$\Delta\varepsilon = \alpha(T_{max} - T_{min})$$

The above expressions are equated and solved for N_f using an iterative technique. The strain range has been calculated from the above equation where the thermal cycle is defined by the minimum and maximum temperatures of the blade metal during the transient.

The minimum cycle temperature is assumed to be equal to ambient temperature. This is because the starting point of the cycle (which is different to that of the transient) is with the engine at rest, unstarted. Therefore all its components are at approximately ambient temperature. The maximum temperature is calculated for various types of acceleration, fast or slow. That temperature is usually greater than the steady state maximum temperature because of overshoots that may take place during the transient.

These lives, or cycles (N_f) to failure, are then expressed in percentage terms using the 8 second power lever angle acceleration as a reference. The life loss, being the difference between unity and the life for a given transient. The reader is reminded that the cycle ranges from engine off to full power, however for all transients the first part of the cycle, startup to idle power, is the same.

7 RESULTS AND DISCUSSION

Six accelerations have been simulated at two different ambient conditions. To distinguish between transients, the power lever transient time has been chosen as an identifier or label. The user can control the power lever transient time. Power lever transient times and speed or thrust transient times do not coincide, the latter being longer.

The power lever transient times are 0.5, 2, 3, 4, 6 and 8 seconds. The ambient conditions examined are ISA SLS and ISA +15K SLS. The results for these conditions are presented in Figures 5 to 7.

In the present work it was assumed that two speed limits were introduced in the control system, one for corrected speed and one for absolute rotational speed. These represent aerodynamic and structural limits respectively. In a standard day, the corrected speed limit is encountered first, However, in hot days, as ambient temperature rises, absolute shaft speed rises until the absolute speed limit is encountered. Thus higher shaft speeds and gas temperatures would be expected at higher ambient temperatures.

In figure 3 it can be seen that there is a gas temperature overshoot at the end of all speed transients. This results in a metal temperature overshoot which can be seen in Figure 4, where fast transients exhibit larger overshoots than slow transients.

The difference between maximum and minimum metal temperatures in the cycle is shown in Figure 5. There it can be seen that, in both cases, of standard and high ambient temperature, this temperature difference is similar for power lever transient times of up to 2 seconds. Therefore these fast accelerations consume the same amount of thermal fatigue life.

The cyclic life of the blade, due to thermal fatigue, is directly dependent on the difference between minimum and maximum metal temperatures, they define the strain range. The peak of the overshoot is the maximum metal temperature and defines the higher limit of the strain range. The magnitude of these overshoots above the steady state value is shown in Figure 6.

In these fast accelerations the fuel flow is determined by the acceleration schedule as explained in section 3. The power lever changes angle very quickly and it demands a fuel flow that is too high, therefore the control system does not allow it and meters the fuel according to the schedule programmed in it to safeguard the integrity of the engine. Thus all the speed transients are completed within approximately 2.5 seconds and are nearly identical.

As the power lever transient time increases the fuel flow rate demanded by the position of the power lever falls below the limits of the schedule and the control system allows it to flow into the engine because it is safe to do so. As a result the maximum blade metal temperatures are lower for the longer, or slower, transients (Figure 5).

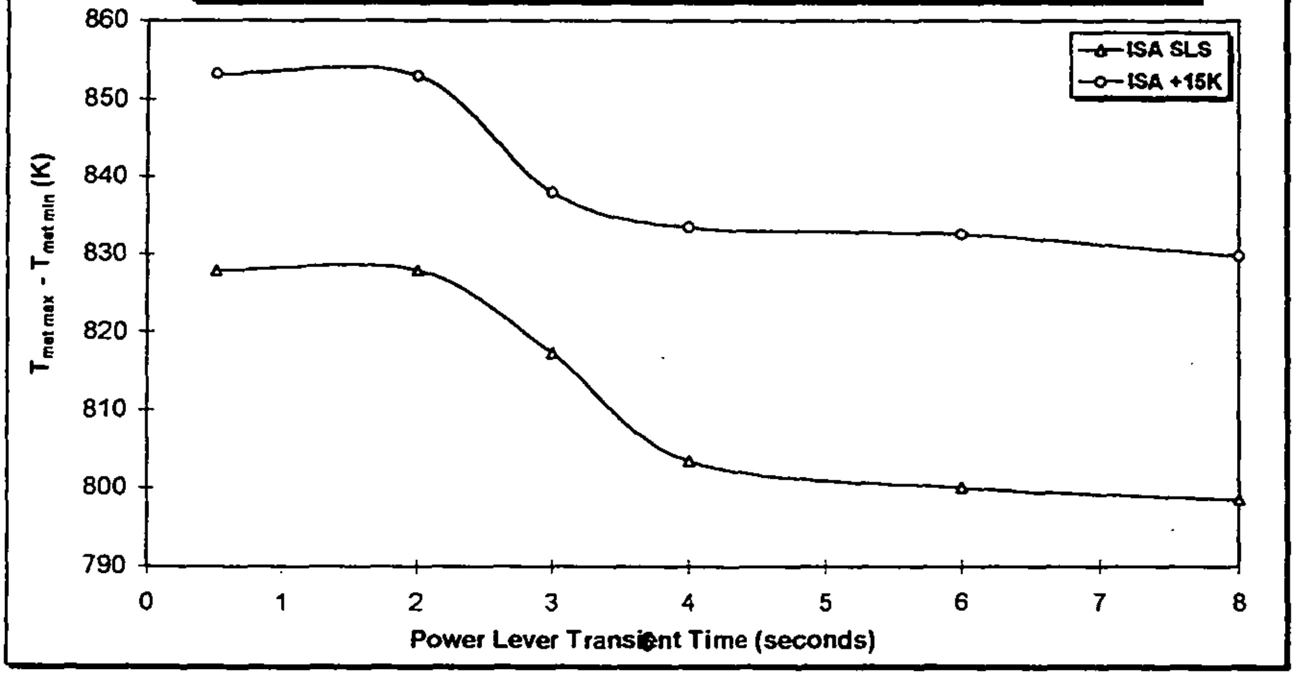
When the power lever transient time is more than 4 seconds the fuel flow rate is always limited by the power lever steady state fuel demand value and very small differences in the metal temperature range are observed, i.e. the metal temperature overshoots above the steady state value are small.

These metal temperature profiles allow an estimate of the changes of blade life. These are presented in Figure 7, where it can be observed that fast transients shorten blade life. The relative thermal fatigue life is calculated with the 8 second acceleration as a datum. The life loss is directly related to the difference between the maximum metal temperature value for an acceleration and that for the 8 second acceleration. The minimum cycle metal temperature is always the same, equal to ambient.

The trend of the life loss curves is similar to that of the temperature overshoot in Figure 6. A constant temperature overshoot up to the 2 second acceleration is observed. The same can be said for the acceleration where the power lever is moved in 4 seconds or more. There is of course a smaller overshoot in the latter cases.

Comparing the metal temperature difference in the standard and high ambient temperature cases, in Figures 5 and 6, it can be seen that the hot day case exhibits a smaller temperature overshoot in most of the accelerations. This is because the maximum steady state turbine temperature is higher in a hot day, as permitted by the steady state schedule while the maximum cycle temperature is limited by integrity considerations and it is allowed to rise only slightly with higher ambient temperatures.

Figure 5. Thermal Cycle Temperature Difference vs. PL Transient Time



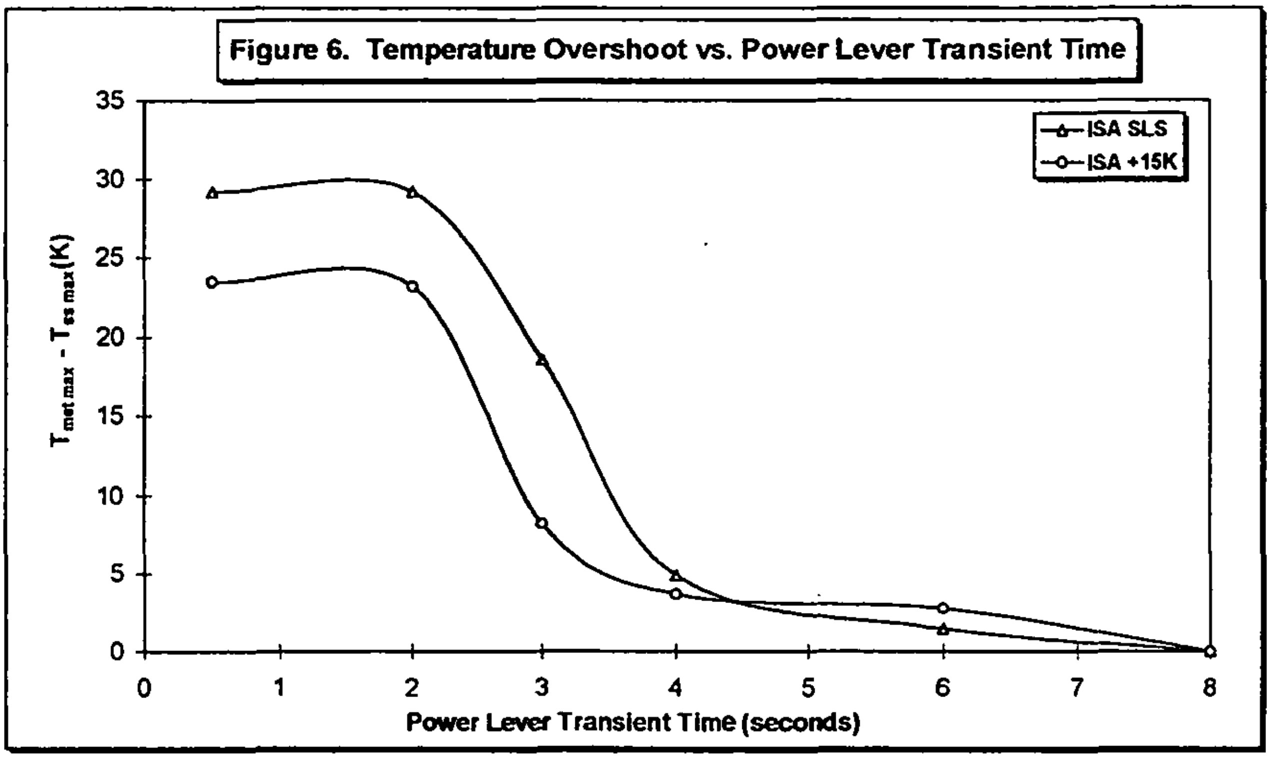


Figure 6. Temperature Overshoot vs. Power Lever Transient Time

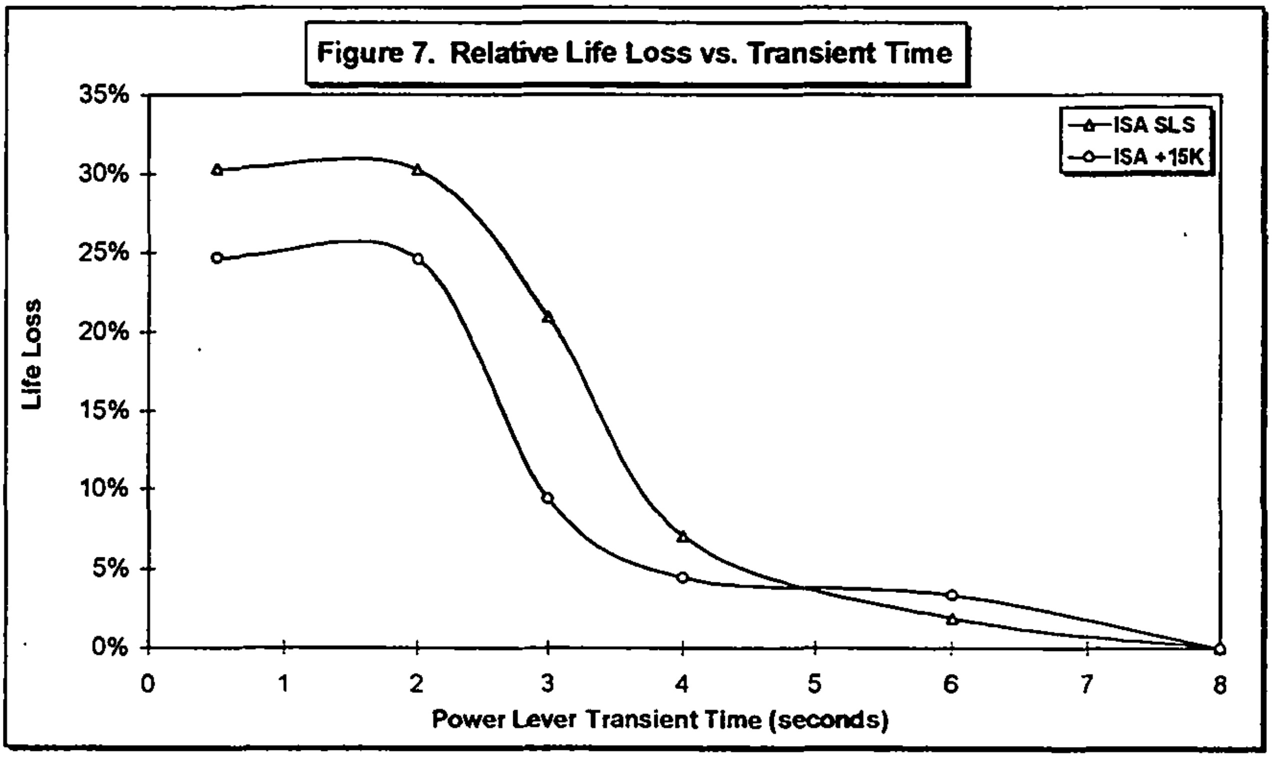


Figure 7. Relative Life Loss vs. Transient Time

So the difference between maximum transient metal temperature and maximum steady state temperature is now smaller. This explains the greater relative life loss for the standard day shown in Figure 7.

When the power lever angle transient lasts more than five seconds an inversion of the trend of life lost is observed which can be attributed to calculation inaccuracies.

The calculated relative life loss when requiring a fast response and the power lever movement is completed in less than 2 seconds approaches 30% for standard ambient temperature and 25% for high ambient temperature. The reader is reminded that this loss of life is attributed to thermal fatigue only and that the datum acceleration is that where the power lever movement is completed in 8 seconds. The 3 second acceleration exhibits a significantly smaller thermal fatigue life reduction (~ 20% for standard day and ~ 8% for a hot day). As the power lever transient times increase the relative life loss is reduced and for a 4 second power lever transient it is approximately 5%. For the 6 second one it is of the order of 4%.

From the above results it can be seen that, for the standard day case, for each degree of turbine temperature overshoot there is a reduction of thermal fatigue life of approximately one percent in this engine. For the hot day case the life loss appears to be slightly greater than this. The overshoots are smaller but the temperatures are higher, so the loss of life is similar. The conclusion could be drawn that the life penalty that has to be paid per degree K of metal temperature overshoot is approximately one percent.

8 CONCLUSIONS AND FUTURE RESEARCH

An investigation into the effects of accelerating a gas turbine at different rates has been carried out. The vehicle for the present work has been a transient engine model. It has thermal fatigue estimation algorithms that allow a comparative life loss assessment. As expected, the faster the acceleration, the faster the life consumed, from the of thermal fatigue point of view.

Specifically at the ISA SLS case a 30 percent reduction of thermal fatigue life has been estimated. The life loss seems to be constant for power lever transients of up to 2 seconds. Beyond that, there is a sudden reduction in relative life consumption up to transients of approximately four seconds, where the life loss is estimated to be approximately five percent higher than that of the datum case of the power lever transient of eight seconds. Beyond this point, the transient approaches a state where it can considered to be a sequence of steady state points and the relative life loss is very small.

Similar trends are observed in the case where ambient temperature is ISA + 15 K. The relative life loss, in this case, is lower. It is approximately 25 percent for the fast accelerations while it appears to be the same as that of the standard day temperature for slow accelerations.

Only thermal fatigue effects have been accounted for. Low Cycle Fatigue will not play an important role if the maximum rotational speed reached is assumed to be the same for all the accelerations at the same ambient temperature. This will not be the case for the High Pressure shaft if its rotational speed is not taken into account by the acceleration schedule. HP shafts may overshoot in speed during the transient due to their lower rotational Inertia.

Creep is a major failure mechanism that must be considered, it is the intention of the authors to do so. This phenomenon is expected to influence the results very significantly. The observed trend for lower life loss at higher ambient temperatures will probably be inverted because the maximum cycle temperature is approximately 25 K higher for all the accelerations in a hot day. It is generally accepted that a 15-20 K increase in metal temperature halves the creep life of a hot section component.

The main result of this work is that, as expected, engine component life is shortened by fast transients. In the present paper the focus has been on thermal fatigue, where a life loss of one third to one quarter has been predicted if a fast transient is demanded.

The results obtained here are very interesting. The authors will develop the model described here so it can account for creep and low cycle fatigue. This will enable comparative life assessments in many different missions.

9 ACKNOWLEDGMENT

The authors are grateful to their colleagues in the Hellenic Air Force and Cranfield for their support and encouragement throughout this work.

10 REFERENCES

1	D. Adamopoulos	Handling and Transient Operation of a Military Turbofan, MSc Thesis, Cranfield University, 1996
2	Bailey, F. J. & Tumer, A. B.	Transpiration Cooled Turbines AGARD-CP-73-71
3	Manson, S.S.	Thermal Stress and Low Cycle Fatigue, Mc Graw-Hill Book Company, 1966
4	Pilidis, P. & Maccallum, N. R. L.	A General Programme for the Prediction of the Transient Performance of Gas Turbines. Paper 85-GT-208