Optimal gas turbine inlet temperature for cyclic operation

R Z Aminov¹, A B Moskalenko¹ and A I Kozhevnikov²

¹ Saratov Scientific Center of Russian Academy of Sciences, 410054, Russia, Saratov, Politechnicheskaya street, 77
² Turbulence and Vortex Dynamics Group, Department of Applied Physics, Eindhoven University of Technology, P.O. Box 513, 5600 MB Eindhoven, The Netherlands

e-mail: a.kozhevnikov@tue.nl, oepran@inbox.ru

Abstract. Historically, the development of gas turbine technologies is intrinsically linked with the increase in the initial temperature of the working fluid, which provides an increase in the thermal efficiency and thermodynamic efficiency of the gas turbine cycle. The increase in the gas turbine initial temperature leads to the unit costs reduction and the unit capacity is increased. However, higher turbine inlet parameters increase significantly thermal stresses in the metal, especially during a start-stop operation. In this study we compared the efficiency of three gas turbines with different values of the turbine inlet temperature during covering different parts of the daily electric load curve. The sum of fuel and depreciation costs is chosen as an optimization criterion. Service life parameters were determined for the most thermal stressed gas turbine element, namely the first stage gas turbine blades. To implement the low-cycle fatigue analysis, the numerical simulations were implemented using ANSYS®. The modelling was divided in two stages. Firstly, the fluid dynamics behaviour was analysed around the blade cascade with the aim to determine the thermal state during the whole start-stop cycle. In the second stage of calculation, the mechanical loads were added caused by centrifugal force and the fluid flow forces from the working fluid side. To determine depreciation costs, the modified equivalent operating hours principle was used. Obtained results showed that for cyclic operation mode with a high number of start-ups and shutdowns, the costs associated with service life reduction exceed markedly the fuel economy, making turbines with lower turbine inlet temperature more profitable.

1. Introduction

In recent decades, the development of energy systems has been accompanied by an inherently increasing non-uniformity between the consumption and generation of electricity. Despite the development of different energy storage technologies, the most appropriate types of plant for providing electrical energy needs are currently conventional types running on fossil fuel. Among all such power plants, those based on gas turbines are most suitable for cyclic operation. Although gas turbines are highly flexible, frequent start-ups, shutdowns, and rapid load changes significantly reduce their service lives, consequently increasing operating and maintenance (O&M) costs.

Throughout the developmental history of gas turbine technology, increasing efficiency has been gained mainly by raising the initial temperature of the working fluid. However, the capital costs increase along with thermal efficiency as a result of the use of more expensive materials and advanced cooling for high-temperature components. Also, a higher-temperature working fluid places much greater thermal stresses onto metal during start-up and shutdown operations. These contrasting factors lead to an optimization problem regarding turbine inlet temperature for use on different parts of the
daily electric load curve. The present study attempted to define the most preferred types of gas
turbines for different operational modes, taking into account the additional costs associated with
changes in the operating life of highly stressed elements. Some of the first research from a general
point of view devoted to the accounting of wear and tear on power plants’ equipment and the
associated additional costs are [1], [2]. Later studies analysed how cyclic operation increases O&M
costs, related to steam turbines in [3], [4] and related to steam turbine and combined cycle power-plant
equipment in [5]. All these studies evaluated the additional O&M costs based on statistical data
obtained from plant owners, an approach that is applicable only when the considered equipment has
been operated for a long time. In the case of new or even newly designed power plants, performing a
strength analysis of the most thermally and mechanically stressed elements for the given unit is more
suitable.

Currently, the maintenance strategy of power equipment is widely optimized using the concept of
equivalent operating hours (EOH). This principle suggests that each start-up, shutdown, or rapid
change in load be assigned a certain number of operating hours at nominal load equivalent to these
stresses. in [6] EOH was used to assess the reduction in life of a combined cycle power plant’s
equipment. The authors of this article have previously used the EOH concept to determine the lifetime
potential of gas turbines [7].

2. Methodological preliminaries

Total operating costs for a gas turbine unit comprise the fuel costs and the costs associated with
changes in its operating life (depreciation costs):

\[ C = C_{fuel} + C_{life} \]  

(1)

Fuel costs can be determined according to the following expression:

\[ C_{fuel} = \left( \sum_{i=1}^{l} B_{\text{start},i} + \sum_{j=1}^{J} B(N) \tau_j \right) \cdot P_{fuel} \]  

(2)

where \( B(N) \) is the fuel consumption as a function of the load; \( B_{\text{start}} \) the fuel consumption necessary
to start-up a gas turbine; \( P_{fuel} \) is the price per unit of the used fuel; \( I \) is the number of start-ups from
different thermal states of a turbine; \( J \) is the number of considered power levels.

As mentioned above, the EOH principle is currently used to determine the optimal frequency for
gas turbine maintenance [8]. However, to solve the problem concerning the optimal parameters of the
working fluid, consideration can be limited only to those components that depend on the operational
mode. Accordingly, the modified equation can be rewritten in the following form:

\[ T_{eqv} = \sum_{i=1}^{l} a_i n_i + \sum_{j=1}^{J} b_j \tau_j \]  

(3)

where \( a_i \) is the coefficient for the \( i \)-th start-up type; \( n_i \) is the number of start-ups and load changes of
the \( i \)-th type; \( I \) is the total number of start-ups and load changes; \( b_j \) is the coefficient for the \( j \)-th
operational mode; \( \tau_j \) is the operating time at the \( j \)-th operational mode; \( J \) is the total number of
operational modes in the considered period. In equation (3), the first term accounts for the depreciation
of the gas turbine as a result of intermittent operations, and the second term defines the operating life
reduction during operation at a constant level of power output.

To determine the coefficient \( a_i \) in equation 3 it is necessary to perform the calculation of the most
critical elements on fatigue strength. The aim of this calculation is to determine the thermo-mechanical
stresses, followed by the determining the number of cycles to failure of the considered elements. In general, the formula for the coefficient \(a_i\) is as follows:

\[
a_i = \frac{T_{\text{life}}}{N_{f,i}}
\]

where \(T_{\text{life}}\) is the lifetime of the element; and \(N_{f,i}\) is the number of \(i\)-th types cycles to failure.

Finding the coefficient \(b_j\) assumes the calculation of turbine elements on the long-term strength.

The coefficient can be found by the following formula:

\[
b_j = \frac{T_{\text{life}}}{\tau_{f,j}}
\]

where \(\tau_{f,j}\) is the time to failure for the \(j\)-th operation mode, which can be defined by the long-term strength equation [9].

3. Results and analysis

As an example of the choosing the optimal turbine inlet temperature (TIT) three gas turbines of the same power range level but with different initial temperatures were considered (table 1) [10, 11].

| Turbine                        | Base Load (kW) | Efficiency (%) | Budget Price (thousand S) | Exhaust Temp. (°C) | Inlet Temp. (°C) | Pressure Ratio | Fuel consumption at 100% load (for LHV=36.62 MJ/m³) (m³/hour) | Start fuel consumption (m³) |
|--------------------------------|----------------|----------------|---------------------------|--------------------|-----------------|----------------|---------------------------------------------------------------|-----------------------------|
| Alstom (50 Hz) GT13E2          | 184500         | 37.8           | 41396                     | 225                | 505             | 1100           | 16.9                                                         | 47983                       | 11996                       |
| Siemens Energy (60 Hz) SGT6-5000F | 208000         | 38.1           | 43307                     | 208                | 578             | 1260           | 17.2                                                         | 53669                       | 13417                       |
| Mitsubishi Heavy Industries (50 Hz) M701F4 | 324300         | 39.9           | 63683                     | 196                | 592             | 1350           | 18                                                          | 77506                       | 19377                       |

Turbine blades for different TIT values are not the same. Turbine blades of Siemens Energy (60 Hz) SGT6-5000F have the same geometry as blades of Alstom (50 Hz) GT13E2 but it was assumed that they are covered by a thermal barrier coating. Turbine blades of Mitsubishi Heavy Industries (50 Hz) M701F4 have also a thermal barrier coating with a higher number of orifices for the air exit and blade cooling.

The gas temperature after the nozzle blades is determined by the following formula:

\[
T = T_3 - \varphi^3 \frac{(1-\varphi)H_1}{C_p} + \Delta T_{\text{gas}} - \Delta T_{\text{mixing}}
\]

(6)
where $T_3$ is the gas temperature of the combustion chamber; $\varphi$ is the velocity coefficient of the first stage nozzle blades; $\rho$ is the degree of reactivity; $H_1$ is the enthalpy drop in the first stage; $\Delta T_{\text{gas}}$ is the gas temperature increase due to the flow deceleration in the nozzles; $\Delta T_{\text{mixing}}$ is the gas temperature decrease due to the mixing with the cooling air.

Accounting of these components leads to the gas temperature increase by 15–16 K or 1–1.5%.

This study only considered startup and shutdown operations when calculating the depreciation costs, assuming for simplicity of calculation that gas turbines always operate at 100% load (that is, without considering partial-load operation). The following operational modes covering the daily electric load curve were chosen [13, 14]:

- base load: operation for 24 hours;
- part peak load: operation for 19 hours, followed by stopping for 5 hours with one startup;
- peak load: operation during peaks for 3 and 4 hours with two startups.

To determine the coefficient in equation 3 the first-stage turbine blades were considered to be the most thermally stressed elements. In figure 1 the startup graph is shown according to [15]. Temperatures of the working fluid before turbines Alstom GT13E2 and Siemens SGT6-5000F are taken from [16, 17]. The turbine Mitsubishi Heavy Industries M701F4 is the type F turbine and according to [18] the turbine inlet temperature is 1350 °C. Gas turbines Alstom GT13E2 and Siemens SGT6-5000F have the startup time 28 and 30 minutes correspondingly. According to the data sheet of Mitsubishi Heavy Industries every turbine from type D to type J has the startup time of 30 minutes. The turbine inlet temperatures of the turbine Mitsubishi Heavy Industries M701F4 is significantly higher than for turbines Alstom GT13E2 and Siemens SGT6-5000F, while the startup times are equal [19]. From this one can assume that the depreciation of the turbine Mitsubishi Heavy Industries M701F4 during the startup is higher. Calculation of low cycle fatigue was carried out using the commercial software package ANSYS Workbench. The simulation was implemented in two stages. The first involves the calculation of fluid dynamics in Fluent code, which models the thermal state of the gas turbine blades across the whole startup cycle. The second stage was implemented in the Mechanical application, adding to the modeled thermal loads from the first stage the mechanical loads from centrifugal and hydrodynamic forces from the working fluid. Modern gas turbines use different cooling techniques and thermal barrier coatings to prevent overheating. Dealing with these variations is out of the scope of this research; discussion can be found in [20–23]. The authors decided to model the effects of thermal barrier coatings by reducing the surface temperature of a blade by 100°C compared to the temperature of the surrounding gases near the wall [24].

In the beginning of a startup, just after the ignition in the combustion chamber we can observe the most rapid gas turbine temperature increase (figure 2). The most rapid and amplitude gas turbine temperature change occurs in the outer surface of the blade leads to the highest temperature gradient and as a result to the highest thermal stresses in this place.

Figure 3 shows the dependence of the maximum blade temperature on the time during startup.

In figure 4 there is the graph of the temperature difference between points on the outer surface of the blade in the near-root cross section on the inlet and outlet edges.

Figure 5 shows the stresses in the gas turbine blade within the first 250 seconds.
Figure 1. General start-up graph of a gas turbine.

Figure 2. Turbine inlet temperature during the startup; 1 – Alstom (50 Hz) GT13E2; 2 – Siemens Energy (60 Hz) SGT6-5000F; 3 – Mitsubishi Heavy Industries (50 Hz) M701F4.
Figure 3. Maximum blade temperature during startup; 1 – Alstom (50 Hz) GT13E2; 2 – Siemens Energy (60 Hz) SGT6-5000F; 3 – Mitsubishi Heavy Industries (50 Hz) M701F4.

Figure 4. The temperature difference between points on the outer surface of the blade in the near-root cross section on the inlet and outlet edges; 1 – Alstom (50 Hz) GT13E2; 2 – Siemens Energy (60 Hz) SGT6-5000F; 3 – Mitsubishi Heavy Industries (50 Hz) M701F4.
Figure 5. Stresses in the gas turbine blade; 1 – Alstom (50 Hz) GT13E2; 2 – Siemens Energy (60 Hz) SGT6-5000F; 3 – Mitsubishi Heavy Industries (50 Hz) M701F4.

The number of cycles to failure $N_f$ was determined using the low-cycle fatigue curve for the alloy IN-738 (figure 6) [25]. Results of calculations on the low-cycle fatigue are presented in the table 2.

Figure 6. Low-cycle fatigue curve for the alloy IN-738.
Table 2. Calculation results of the life reduction for 1 start up.

| Turbine                          | Temperature (blade/gas) (°C) | Von Mises stress (MPa) | Total equivalent strain | Nf  | EOH/1 start |
|----------------------------------|------------------------------|------------------------|-------------------------|-----|-------------|
| Alstom (50 Hz) GT13E2            | 883/1024                     | 607                    | 3.4484·10⁻³            | 4228| 5.9         |
| Siemens Energy (60 Hz) SGT6-5000F | 908/1154                     | 613                    | 3.4977·10⁻³            | 3902| 6.4         |
| Mitsubishi Heavy Industries (50 Hz) M701F4 | 928/1237                  | 837                    | 4.4383·10⁻³            | 1014| 25          |

Figure 7 shows the stress field for the Alstom (50 Hz) GT13E2 turbine blade.

As can be seen from table 2 the increase of TIT significantly affects on the life reduction during cyclic operation. Heating of the blades of the first stages above 1200 °C is not allowed in accordance with the data of [26]. As can be seen from table 2 and figure 8 that the maximum temperature of the blade in the high-temperature turbine itself does not exceed 928 °C.
Figure 8. Field of temperature distribution in metal at rated load a – Alstom (50 Hz) GT13E2; b – Siemens Energy (60 Hz) SGT6-5000F; c – Mitsubishi Heavy Industries (50 Hz) M701F4.

Results of calculations of the fuel consumption and life reduction are presented in table 3 and 4, respectively.
Table 3. The fuel consumption for a day (thousand m³).

| Turbine                        | Base-load | Part-peak-load | Peak-load           |
|--------------------------------|-----------|----------------|---------------------|
| Alstom (50 Hz) GT13E2          | 1151.6    | 911677+11996=923.7 | 335881+2×11996=359.9 |
| Siemens Energy (60 Hz) SGT6-5000F | 1288.1    | 1019711+13417=1033.1 | 375683+2×13417=402.5 |
| Mitsubishi Heavy Industries (50 Hz) M701F4 | 1860.1    | 1472614+19377=1492  | 581296+2×19377=581.3 |

Table 4. The life reduction for a day (hours).

| Turbine                        | Base-load | Part-peak-load | Peak-load           |
|--------------------------------|-----------|----------------|---------------------|
| Alstom (50 Hz) GT13E2          | 24        | 19+5.9=24.9    | 7+2×5.9=18.8        |
| Siemens Energy (60 Hz) SGT6-5000F | 24        | 19+6.4=25.4    | 7+2×6.4=19.8        |
| Mitsubishi Heavy Industries (50 Hz) M701F4 | 24        | 19+24.7=43.7  | 7+2×24.7=56.4       |

The cost of repairs can be determined using the data [27]. To determine depreciation costs according to equation (3) the cost replace and the major overhauls were taken into account and costs of repair events with less periodicity were included into costs of abovementioned ones. Accordingly, the costs of life reduction of gas turbines per 1 EOH are shown in table 5 [28, 29].

Table 5. The costs of 1 EOH for the considered gas turbines ($).

| Turbine                        |             |
|--------------------------------|-------------|
| Alstom (60 Hz) GT13E2          | 952.1       |
| Siemens Energy (50 Hz) SGT6-5000F | 996.0      |
| Mitsubishi Heavy Industries (50 Hz) M701F4 | 1464.7 |

The values of fuel costs, depreciation costs and the sum of them calculated according to equations (1) – (3) presented in tables 6 - 8.

Table 6. Daily fuel costs for the fuel price 61.4 $/1000 m³ (absolute values, (thousand $) / specific values ($/MWh)).

| Turbine                        | Base-load | Part-peak-load | Peak-load |
|--------------------------------|-----------|----------------|-----------|
| Alstom (50 Hz) GT13E2          | 70.74/16  | 56.74/16.14    | 22.11/17.14 |
| Siemens Energy (60 Hz) SGT6-5000F | 79.12/15.86 | 63.46/16       | 24.73/17  |
| Mitsubishi Heavy Industries (50 Hz) M701F4 | 114.27/14.71 | 91.65/14.86    | 35.71/15.71 |
### Table 7. Daily depreciation costs (absolute values, (thousand $) / specific values ($/MWh)).

| Turbine                        | Base-load | Part-peak-load | Peak-load |
|--------------------------------|-----------|----------------|-----------|
| Alstom (50 Hz) GT13E2          | 22.85/5.14| 23.72/6.71     | 17.92/13.86|
| Siemens Energy (60 Hz) SGT6-5000F | 23.90/4.86| 25.31/6.43     | 19.74/13.57|
| Mitsubishi Heavy Industries (50 Hz) M701F4 | 35.15/4.57| 63.94/10.43    | 82.48/36.29|

### Table 8. Daily total costs (thousand $).

| Turbine                        | Base-load | Part-peak-load | Peak-load |
|--------------------------------|-----------|----------------|-----------|
| Alstom (50 Hz) GT13E2          | 93.59     | 80.46          | 40.03     |
| Siemens Energy (60 Hz) SGT6-5000F | 103.03    | 88.77          | 44.46     |
| Mitsubishi Heavy Industries (50 Hz) M701F4 | 149.42    | 155.59         | 118.19    |

To account that the considered turbines have different power output table 9 shows specific total costs per 1 MWh generated electricity.

### Table 9. Specific total costs ($/MWh).

| Turbine                        | Base-load | Part-peak-load | Peak-load |
|--------------------------------|-----------|----------------|-----------|
| Alstom (50 Hz) GT13E2          | 21.14     | 23             | 31        |
| Siemens Energy (60 Hz) SGT6-5000F | 20.57     | 22.43          | 30.57     |
| Mitsubishi Heavy Industries (50 Hz) M701F4 | 19.14     | 25.29          | 52        |

As can be seen from tables 6, 7 and 9 show that the depreciation, fuel and specific total costs for the Siemens Energy (60 Hz) SGT6-5000F and Alstom (50 Hz) GT13E2 turbines are practically the same. The decrease in blade life is negligible due to the use of a thermal barrier coating and because of the increase in the start-up time by 5 minutes (from 25 to 30 minutes). The difference in the resource of these turbines is extremely small with a large difference in power, which leads to this result.

The fuel costs of the Mitsubishi Heavy Industries (50 Hz) M701F4 turbine are lower than those of the previous ones in any mode, but the resource and total costs are higher in the part-peak and peak loads. This indicates a significant excess of resource costs over fuel and low efficiency of using high-temperature gas turbines in the part-peak and peak loads.

### 4. Conclusions
1) The increase of the initial temperature increases the thermodynamic efficiency of the Brayton cycle, increases thermal efficiency and reduces the fuel consumption. On the other hand, the operation of higher temperatures requires the use of more expensive materials and the incorporation of more complex technical solutions to prevent the overheating. All of these greatly increase the capital costs and O&M expenditures of power plants. Moreover, the cyclic operation results in more intensive thermal stresses for gas turbines with higher working fluid parameters, which increases the costs associated with the life reduction. As can be seen from tables 6-8 the depreciation costs have a large
share in the structure of the total operation costs for gas turbines. And this share is higher for more uneven electric load curves.

2) Performed calculations shows that for the base load operation it is more suitable to use gas turbines with higher TIT that provides high fuel efficiency. For peak operation with high number of startups and shut downs the costs associated with life reduction markedly exceed the economy in fuel and turbines with lower TIT become more profitable.

References
[1] Grubb M 1989 The inclusion of dynamic factors in statistical power system cost model Part 1: Assessment of startup and banking costs IEEE Transactions on Power Systems 4–2 p 419–425 DOI: 10.1109/59.193811
[2] Le D Khai, Jackups R Ron, Feinstein Jack, Thompson H Henry, Wolf H Matt, Stein C Edward, Gorski A Daniel and Griffith S Jim 1990 Operational aspects of generation cycling IEEE Transactions on Power Systems 5–4 p 1194–1203 DOI: 10.1109/59.99370
[3] Mirandola A, Stoppato A and Casto E Lo 2010 Evaluation of the operation strategy of a steam power plant on the residual life of its devices Energy 35–2 p 1024–1032 DOI: 10.1016/j.energy.2009.06.024
[4] Stoppato A, Mirandola A, Meneghetti G and Casto E Lo 2012 On the operation strategy of steam power plants working at variable load: Technical and economic issues Energy 371 p 228–236 DOI: 10.1016/j.energy.2011.11.042
[5] Keatley P, Shibli A and Hewitt N J 2013 Estimating power plant start costs in cyclic operation Applie Energy 111 p 550–557 DOI: 10.1016/j.apenergy.2013.05.033
[6] Radin YU A and Kontorovich T S 2012 Ispol'zovaniye printsipa ekvivalentnoy narabotki dlya otsenki nadezhnosti oborudovaniya PGU Elektricheskiye stantsii 1 p 16–18
[7] Aminov R Z and Kozhevnikov A I 2014 Estimating the influence of intermittent operation on the change of life of a gas turbine Electric Power Systems Research 107 p 153–157 DOI: 10.1016/j.epsr.2013.10.003
[8] Gas turbines - Procurement - Part 9: Reliability, Availability, Maintainability and Safety ISO 3977-9:1999
[9] Maslenkov S B 1983 Zharoprochnye stali i splavy (Moscow: Metallurgiya) p 192
[10] 2012 Gas Turbine World Handbook vol 29 (Fairfield: Pequot Publishing Inc) p 148
[11] Moran M J, Shapiro H N, Boettner D D and Bailey M B 2011 Fundamentals of engineering thermodynamics (Hoboken: John Wiley & Sons, Inc)
[12] Syrkin P E and Sheherbakov V V 2010 Osnovy prikladnoy gazovoy dinamiki: kompleks uchebno-metodicheskikh materialov ed E E Rams (Nizhniy Novgorod: NGTU) p 124
[13] 2015 European network of transmission system operators for electricity URL: www.entsoe.eu
[14] 2013 Sistemnyy operator Yedinoy energeticheskoy sistemy URL: www.so-ups.ru
[15] Takahashi T, Watanabe K and Takahashi T 2001 Transient Analyses of Conjugate Heat Transfer of a First Stage Rotor Blade in Start-Up and Shut-Down ASME Turbo Expo 2001: Power for Land, Sea, and Air. American Society of Mechanical Engineers V003T01A049-V003T01A049
[16] Klohr M, Schmidtke J, Tschirren S and Rihak P 1995 Initial Operating Experience and Test Results of ABB’s Gas Turbine GT13E2 ASME International Gas Turbine and Aeroengine Congress and Exposition. American Society of Mechanical Engineers V003T06A044-V003T06A044
[17] 2015 Reliable and powerful – economical, safe-investment packages SGT6-PAC 5000F / SCC6-PAC 5000F (Erlangen: Siemens AG) p 18
[18] Ai T, Masada J and Ito E 2014 Development of the High Efficiency and Flexible Gas Turbine M701F5 by Applying “J” Class Gas Turbine Technologies Mitsubishi Heavy Industries Technical Review 51(1) p 1–9
[19] Peng J Q, Zhang H T and Li Y F 2015 Review of blade materials for IG
turbojet Procedia Engineering 130 p 668–675

[20] Vaßen R, Jarligo M O, Steinke T, Mack D E and Stöver D 2010 Overview on advanced thermal
barrier coatings Surface & Coatings Technology 205 p 938–942
DOI: 10.1016/j.surfcoat.2010.08.151

[21] Asghari S and Salimi M 2010 Finite element simulation of thermal barrier coating performance
under thermal cycling Surface & Coatings Technology 205 p 2042–2050 DOI:
10.1016/j.surfcoat.2010.08.099

[22] Nowak G and Wróblewski W 2009 Cooling system optimisation of turbine guide vane Applied
Thermal Engineering 29 p 567–572 DOI: 10.1016/j.applthermaleng.2008.03.015

[23] Research on heavy-duty gas turbine vane high efficiency cooling performance considering
coolant phase transfer / Jiang Y, Zheng Q, Dong P, Zhang H and Yu F 2014 Applied
Thermal Engineering 73 p 1177–1193 DOI 10.1016/j.applthermaleng.2014.09.023

[24] Padture NP, Gell M, Jordan E H Thermal Barrier Coatings for Gas-Turbine Engine Applications
2002 Science 296 p 280–284
DOI: 10.1126/science.1068609

[25] Radonovich D C 2007 Methods of extrapolating low cycle fatigue data to high stress amplitudes
(Orlando: College of Engineering and Computer Science at the University of Central
Florida)

[26] 2004 Instruktsiya po prodleniyu sroka sluzhby metalla osnovnykh elementov turbin i
kompressorov energeticheskikh gazoturbinnykh ustanovok CO 153-34.17.448-2003
Ministerstvo energetiki rossiyskoy federatsii (Moscow: TSPTI ORGRES) p 46

[27] Klimenko A V and Zorin V M 1999 Teploenergetika i teplootekhnika handbook (Moscow:
publishing house MEI) p 528

[28] Aminov R Z and Kozhevnikov A I 2017 Optimization of the operating conditions of gas-turbine
power stations considering the effect of equipment deterioration Thermal Engineering 64 p
715–22 DOI: 10.1134/S0040601517100019

[29] Kumar N, Besuner P, Lefton S, Agan D and Hilleman D 2012 Power plant cycling costs
National Renewable Energy Laboratory, NREL/SR-5500-55433 p 83