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ANALIZA VIJEKA TRAJANJA DO ZAMORA OŠTEĆENE LOPATICE ROTORA PARNE TURBINE

FATIGUE LIFE ANALYSIS OF THE DAMAGED STEAM TURBINE BLADE

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Sažetak: U radu su prikazani rezultati analize vijeka trajanja do zamora oštećene lopatice zadnjeg stupnja rotora parne turbine. Razmatrana su oštećenja lopatice u obliku bočnih kružnih isjeka lista lopatice na trima različitim pozicijama uzduž lista. Za svaku vrstu oštećenja dodatno je provedena analiza vibracija lopatice pod utjecajem centrifugalne sile pri čemu su rezultati prikazani u vidu formi vibriranja i Campbellovih dijagrama.

Ključne riječi: – procjena vijeka trajanja do zamora
– proračun vibracija
– Campbellov dijagram

Abstract: The paper presents the results of fatigue life analysis of the steam turbine last stage damaged blade. Damages to the shape of circular side cut of the airfoil at the three different positions along airfoil are considered. For each type of blade damage, an additional vibration analysis with the influence of centrifugal force is conducted and the results are presented in the form of mode shapes and Campbell diagrams.

Keywords: – fatigue life estimation
– vibration calculation
– Campbell diagram

1. UVOD

Jedan je od najvažnijih utjecaja na sve rotacijske strojeve, pa tako i na lopaticu rotora parne turbine, utjecaj centrifugalne sile [1, 2]. Centrifugalna sila uzrokuje efekt ukrućivanja, tj. povećanja vlastite frekvencije s brzinom vrtnje. Stoga postoji opasnost da se u području nominalne brzine vrtnje neka od vlastitih frekvencija pobudi nekim od harmonika (višekratnika brzine vrtnje). Utjecaj centrifugalne sile na vlastite frekvencije najpreglednije se prikazuje Campbellovim dijagramom [2-4], koji prikazuje cijelo područje od mirovanja stroja do nominalne brzine vrtnje. Glavni izvor vibracija turbinskih lopatica proizlazi iz nejednolike raspodjele opterećenja, koja je uzrokovana tokom pare. Dodatni razlog neuravnotežene raspodjele sila fluida po površini lopatice tokom jednog okreta proizlazi iz nekontinuiranog pristupa pare lopaticama. Para se distribuira na jednom ili više mjesta po obodu rotora, čiji se broj podudara s brojem harmonika potencijalne uzbude.

Visokociklični zamor materijala lopatica u više je izvješća prepoznat kao glavni uzrok puknuća lopatice. Danas postoji mnoštvo metoda i alata za procjenu vijeka trajanja do zamora. U ovom je radu korišten program *MSC.Fatigue* [5]. Ulazni podaci u obliku prisilnih

1. INTRODUCTION

One of the major influences on rotating machinery as well as on the steam turbine blade is the influence of centrifugal force [1, 2]. It causes the stiffening effect, i.e. increasing of the natural frequencies with running speed. Therefore there is a danger of interference of the blade natural frequencies with the excitation harmonics (multiple of the running speed). Centrifugal force influence can be easily presented in the Campbell diagram [2-4], which displays a whole range of speeds from standstill to the rated speed. The major source of the turbine blade vibration presents uneven load distribution of the steam flow. An additional reason for unbalanced fluid force distribution at the blade surface during one revolution comes from the discontinuous steam approach.

Steam is distributed from one or more places along the rotor circumference, which is a number that presents possible multiples of running speed excitation. High-cycle fatigue cracking of the blade material was reported in many case studies as a major factor of blade failure. Nowadays many different methods and tools exist for assessment of fatigue life. In this work, the program *MSC.Fatigue* is used. Input data in the aspect of forced vibration are prepared in the program *NX Nastran 6*.

vibracija pripremljeni su u programu *NX Nastran 6*. Razmatrana su oštećenja lopatice u obliku bočnih kružnih isjeka lista lopatice na trima različitim pozicijama uzduž lista. Za svaku vrstu oštećenja dodatno je provedena analiza vibracija lopatice pod utjecajem centrifugalne sile, pri čemu su rezultati prikazani putem formi vibriranja i Campellovim dijagramima.

2. VIBRACIJE LOPATICE, CAMPBELLOW DIJAGRAM

Analiza utjecaja centrifugalne sile na vlastite vrijednosti prikazana je na primjeru lopatice rotora zadnjeg stupnja parne turbine [3].

U obzir su uzeti sljedeći opći podaci o parnoj turbini:

- Turbina pogoni generator spojen preko reduktora
- Snaga turbine: $P_{\text{turb}} = 12 \text{ MW}$
- Stanje pare na ulazu: $\vartheta_{\text{ulaz}} = 515^\circ \text{ C}$
 $p_{\text{ulaz}} = 81 \text{ bar}$
- Broj stupnjeva turbine: $N_{\text{stupnj}} = 14$
- Brzina vrtnje: $n = 10810 \text{ min}^{-1}$,

Podaci o zadnjem stupnju turbine:

- Broj lopatica u vijencu: $N_{\text{lop}} = 50$
- Materijal: X22CrMoV12-1
- Temperatura pare: $\vartheta = 100^\circ \text{ C}$
- Polumjer korijena lista lopatice: $r = 200 \text{ mm}$

Damages in the shape of the circular side cut of the airfoil at the three different positions along the airfoil are considered. For each type of blade damage, additional vibration analysis with the influence of centrifugal force is conducted and the results are presented in the form of mode shapes and Campbell diagrams.

2. BLADE VIBRATION, CAMPBELL DIAGRAM

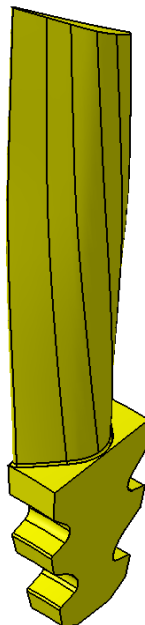
The analysis of centrifugal force influence on the eigenvalues is performed on the last-stage steam turbine blade [3]:

The following data of the steam turbine are taken into consideration:

- Steam turbine generator with gear head
- Power of turbine: $P_{\text{turb}} = 12 \text{ MW}$
- Inlet temperature: $\vartheta_{\text{ulaz}} = 515^\circ \text{ C}$
- Inlet pressure: $p_{\text{ulaz}} = 81 \text{ bar}$
- Number of stages turbine: $N_{\text{stupnj}} = 14$
- Revolutions speed: $n = 10810 \text{ min}^{-1}$,

Technical data of last stage steam turbine:

- Number of blades in the stage: $N_{\text{lop}} = 50$
- Material: X22CrMoV12-1
- Inlet temperature on last stage: $\vartheta = 100^\circ \text{ C}$
- Radius of root blades: $r = 200 \text{ mm}$



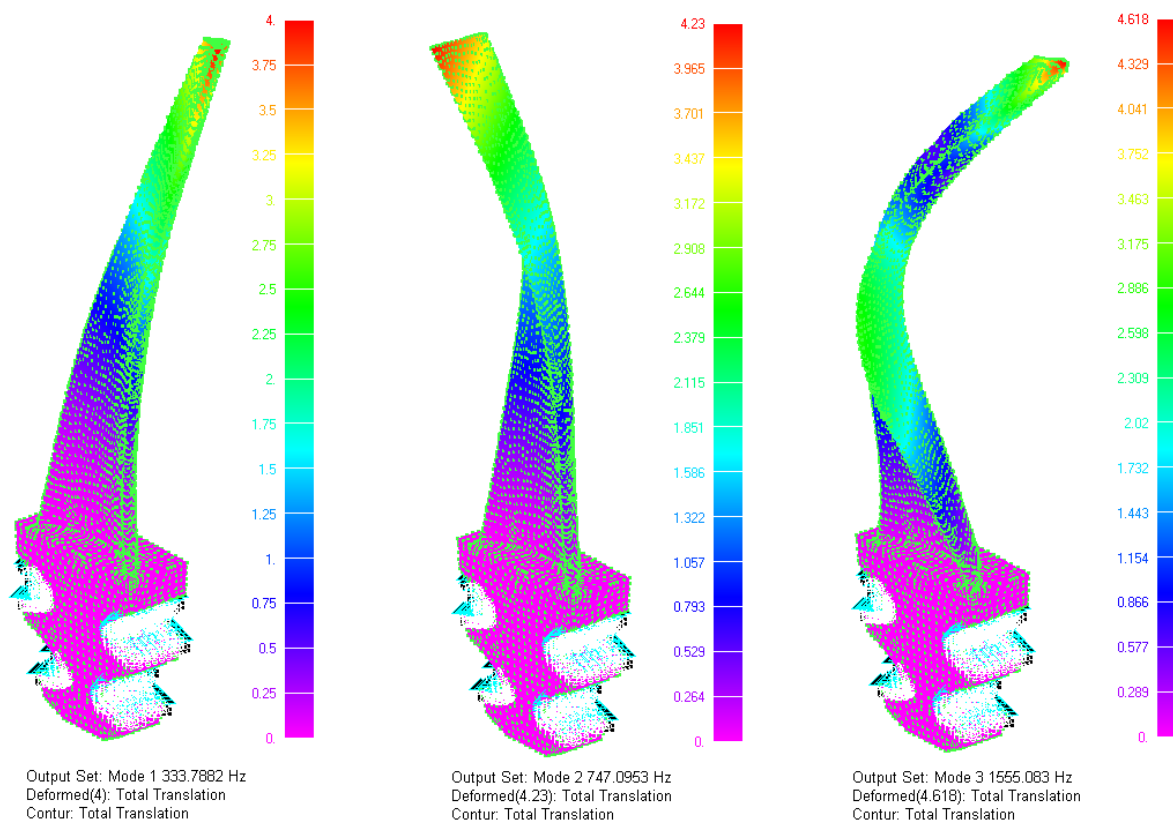
Slika 1. Lopatica rotora zadnjeg stupnja parne turbine
Figure 1. Last stage steam turbine blade

Proračun vlastitih frekvencija izvršen je metodom konačnih elemenata u programskom paketu *NX Nastran 6*. Prve tri forme vibriranja prikazane su na slici 2 dok su pripadne vlastite frekvencije u ovisnosti o brzini vrtnje navedene u tablici 1.

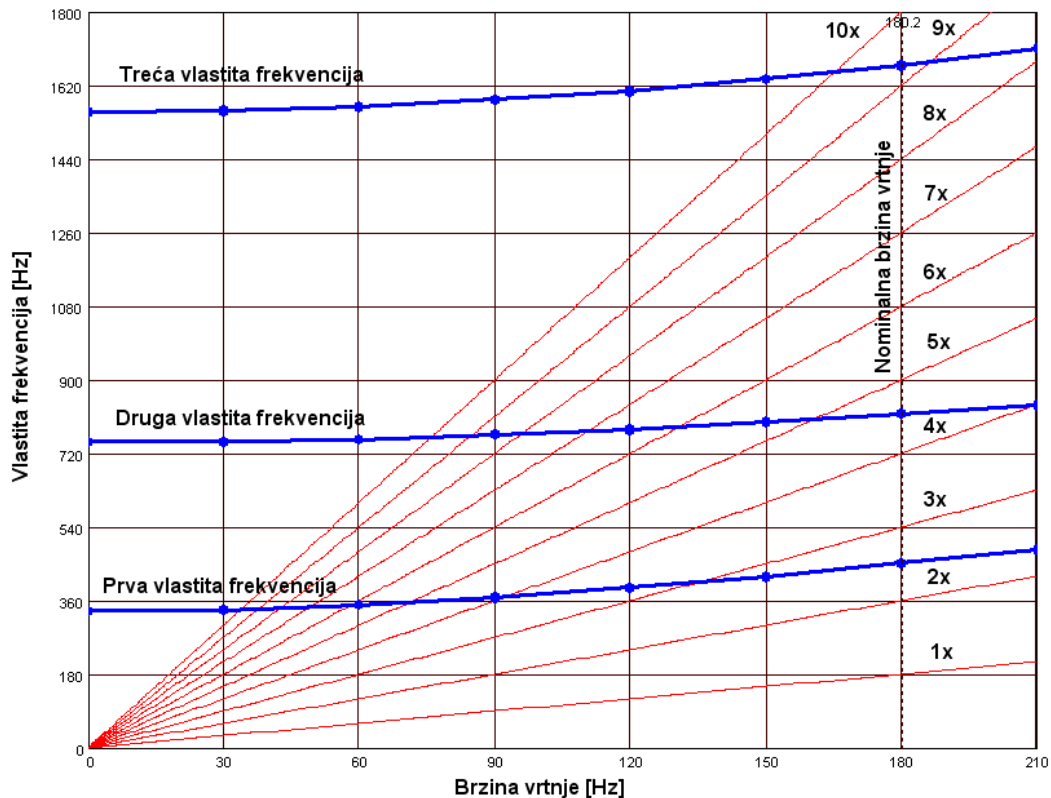
The calculation of natural frequencies is performed by the finite element method with the software *NX Nastran 6*. The first three modes of blade vibration are shown in Figure 2. The first three natural frequencies with respect to the rotor speed are shown on Table 1.

Tablica 1. Prve tri vlastite frekvencije u ovisnosti o brzini vrtnje rotora
 Table 1. First three natural frequencies with respect to the rotor speed

Vlastita frekvencija / Natural frequency [Hz]	Brzina vrtnje / Rotating speed n_i [Hz]						
	0	30	60	90	120	150	180,2
f_1	333,79	337,62	348,86	366,77	390,38	418,63	450,80
f_2	747,10	749,16	755,30	765,35	779,04	796,07	816,20
f_3	1555,08	1558,37	1568,20	1584,47	1607,04	1635,69	1670,40



Slika 2. Prve tri forme vibriranja lopatice
 Figure 2. First three modes of blade vibration



Slika 3. Campbellov dijagram
Figure 3. Campbell diagram

Na osnovi podataka iz tablice 1 kreiran je Campbellov dijagram, slika 3. Pri analizi vibracija lopatica parnih turbina uobičajeno je pratiti utjecaj prvih 10 harmonika [2]. Dobro je konstruirana ona lopatica kod koje je osigurana separacija vrijednosti vlastitih frekvencija od uzbudnih frekvencija, što je važno za pouzdanost lopatice tijekom eksploatacije. Vidljivo je da krivulja prve vlastite frekvencije siječe redom sve harmonike osim prvog i drugog (1x i 2x trenutna brzina vrtnje). Tako će, prilikom startanja ili zaustavljanja turbine, harmonici koji sijeku krivulju prve vlastite frekvencije biti u prilici pobuđivati prvu formu vibriranja. Druga vlastita frekvencija siječe harmonike od petog do desetog, a treća vlastita frekvencija siječe deveti i deseti harmonik uzbude. Treba napomenuti da je iz dijagrama vidljivo da u području nominalne brzine vrtnje niti jedna vlastita frekvencija ne siječe pravce harmonika uzbude, iz čega slijedi da se lopatica turbine pri nominalnoj brzini vrtnje neće naći u području rezonancije. Pri startanju i zaustavljanju turbine treba voditi računa da se one brzine vrtnje, pri kojima vlastite frekvencije sijeku harmonike, prođu što brže da ne bi došlo do oštećenja lopatica turbine zbog zamora.

The Campbell diagram (Figure 3) is drawn based on the natural frequencies from Table 1. In an analysis of steam turbine blades vibrations, usually only the first 10 harmonics of vibrations are monitored [2]. Proper blade design, which ensures separation of the natural frequency from the exciting frequency, is important for its reliability during operation. It can be seen that the curve of the first natural frequency intersects all harmonic lines except for the first two (1x and 2x rotational speed). Hence, at each turbine startup or shutdown, the exciting harmonics that have interference with the blade's first natural frequency will have an opportunity to excite the first mode shape. The curve of the second natural frequency intersects all harmonics from fifth to tenth while the curve of the third natural frequency intersects only the ninth and tenth harmonic. It has to be mentioned that at the rated running speed, neither natural frequency causes interference with the harmonic lines. During run-up and run-down, special attention should be taken to pass the resonance frequency as quickly as possible, to protect the blades from the possible fatigue damage.

3. VIBRACIJE OŠTEĆENE LOPATICE, CAMPBelloV DIJAGRAM

Za potrebe analize prikazane u ovom radu pretpostavljena su oštećenja na trima različitim mjestima na listu lopatice. Za svako oštećenje bit će provedena posebna analiza. Treba naglasiti da su pretpostavljena oštećenja nerealno velika, te da je cilj analize razmotriti ekstremne situacije kakve se u praksi u pravilu ne toleriraju.

3.1. Oštećenje lista pri korijenu lopatice

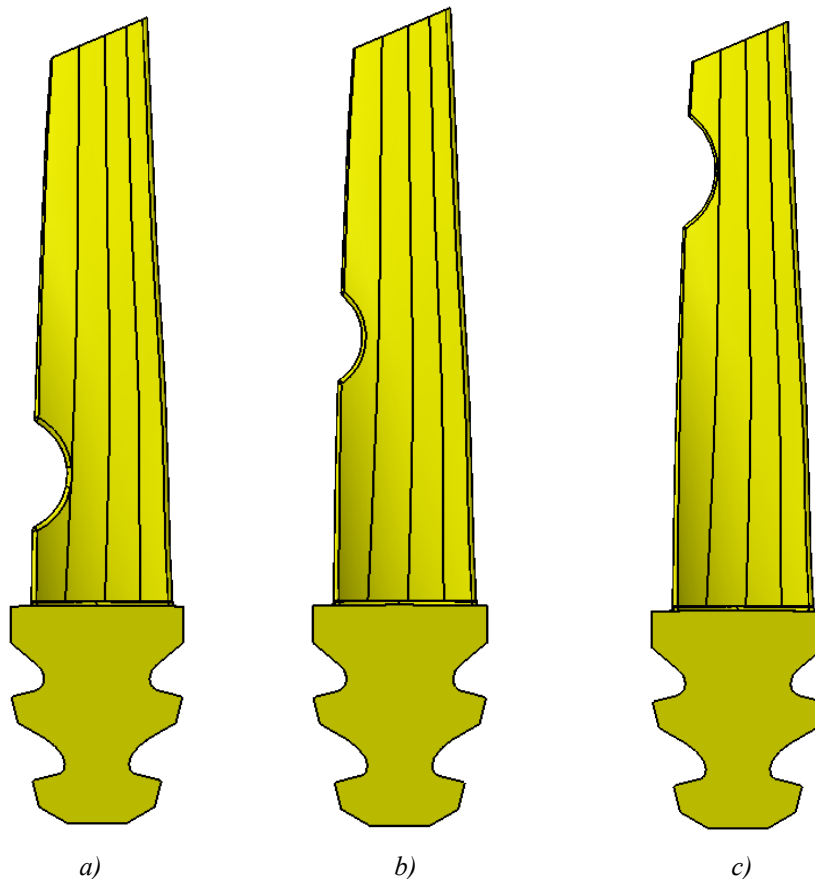
Za prvo oštećenje, u obliku kružnog isjeka, pretpostavljeno je da se nalazi na visini od jedne četvrtine lista (pri korijenu) i da je maksimalne dubine, manje od jedne četvrtine širine profila lista lopatice, slika 4.a).

3. VIBRATION OF THE DAMAGED BLADE, CAMPBELL DIAGRAM

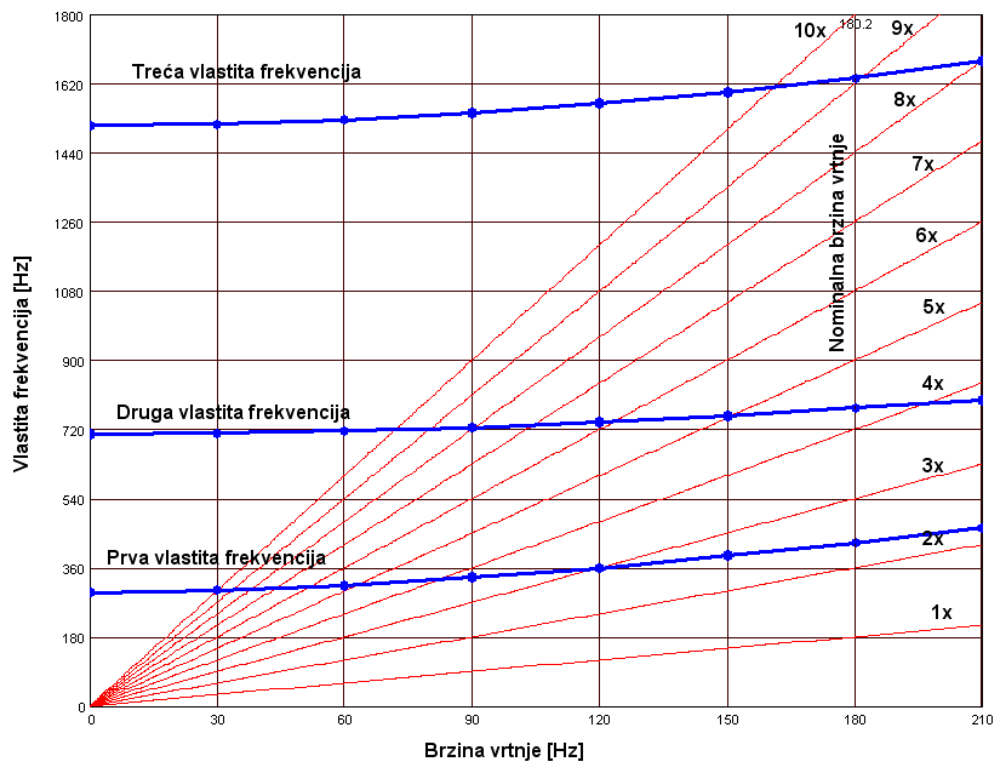
For the sake of the analysis represented in this paper, damage at three different positions along blade airfoil are assumed. For each damage, a separate analysis was conducted. It should be mentioned that the assumed damages are unrealistically big, and the goal of the analysis was to consider the extreme situations that won't normally be tolerable in actual operation.

3.1. Damage of the blade airfoil near blade root

The first damage is assumed as a round cut at the one quarter of the of the airfoil height (near the root) and with a maximum depth smaller than one quarter of the profile width, Figure 4.a).



Slika 4. Oštećen list lopatice, a) pri $\frac{1}{4}$ visine lista, b) pri $\frac{1}{2}$ visine lista, c) pri $\frac{2}{3}$ visine lista
 Figure 4. Damaged blade airfoil, a) at $\frac{1}{4}$ of the airfoil height, b) at $\frac{1}{2}$ of the airfoil height, c) at $\frac{2}{3}$ of the airfoil height,



Slika 5. Campbellov dijagram, oštećenje pri $\frac{1}{4}$ visine lista
 Figure 5. Campbell diagram, damage at the $\frac{1}{4}$ of the airfoil height

Slika 5. prikazuje Campbellov dijagram za oštećenje pri $\frac{1}{4}$ visine lista. Vidljivo je da krivulja prve vlastite frekvencije siječe redom sve harmonike od trećega do desetoga. Tako će, prilikom startanja ili zaustavljanja turbine, harmonici koji sijeku krivulju prve vlastite frekvencije biti u prilici pobuđivati prvu formu vibriranja. Druga vlastita frekvencija siječe harmonike od četvrtog do desetog, a treća vlastita frekvencija siječe deveti i deseti harmonik. Tu se može uočiti da se u području nominalne brzine vrtnje treća vlastita frekvencija može pobuditi devetim harmonikom. Tu spoznaju svakako treba uzeti u obzir ako bi se prilikom eksploatacije pojavilo slično oštećenje lopatice.

3.2. Oštećenje lopatice na sredini lista

Kod drugog oštećenja pretpostavljeno je da se nalazi na visini od jedne polovine visine lista i da je maksimalne dubine, ne veće od jedne četvrtine širine lista lopatice, slika 4.b). Na slici 6. prikazan je Campbellov dijagram za lopaticu s oštećenjem na sredini visine lista. Vidljivo je da krivulja prve vlastite frekvencije siječe redom sve

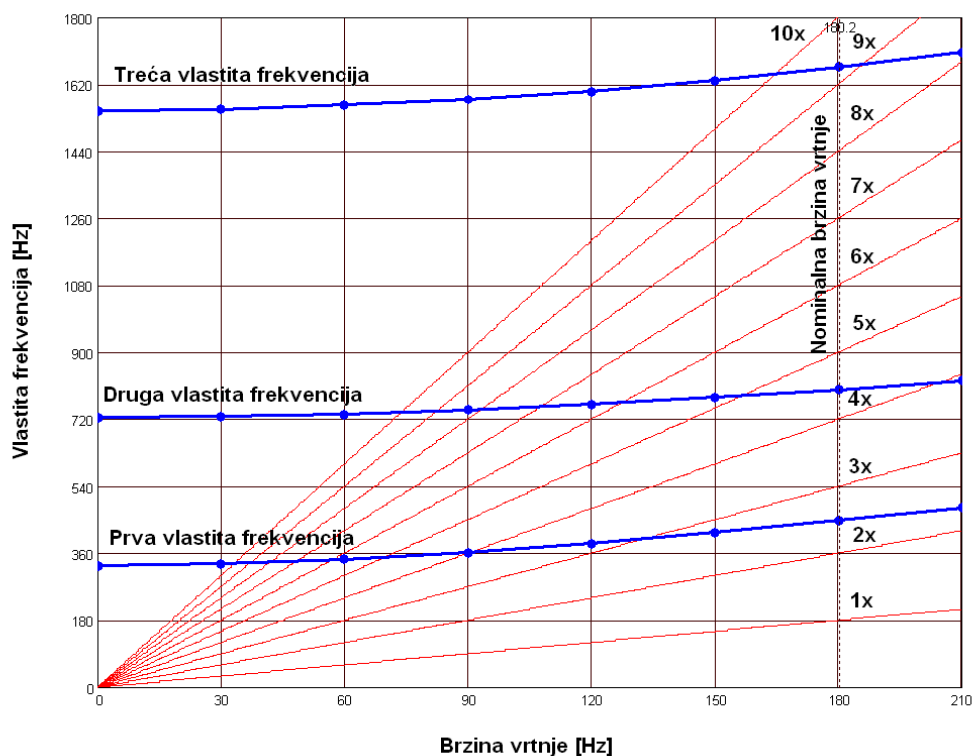
Figure 5 shows a Campbell diagram for damage at $\frac{1}{4}$ of the airfoil height. It can be seen that the curve of the first natural frequency has interferences with all harmonics of the running speed from third to tenth. Hence, at each turbine startup or shutdown, the exciting harmonics that have interference with the blade's first natural frequency curve will have an opportunity to excite the first mode shape. The curve of the second natural frequency intersects all harmonics from fifth to tenth while the curve of the third natural frequency intersects only the ninth and tenth harmonic. With this kind of damage, the interference of the third natural frequency with the ninth harmonic is notable. This information should be considered if some similar blade damage will appear during turbine service.

3.2. Damage at the middle of the blade airfoil

The second damage is assumed at the airfoil mid height and with a maximum depth smaller than one quarter of the profile width, Figure 4.b). Figure 6 shows a Campbell diagram for a blade with damage at mid-height. It can be seen that the curve of first natural frequency has interferences with all harmonics of the running speed

harmonike od trećega do desetoga. To upućuje na činjenicu da će, prilikom startanja ili zaustavljanja turbine, harmonici koji sijeku krivulju prve vlastite frekvencije biti u prilici pobuđivati prvu formu vibriranja. Druga vlastita frekvencija siječe harmonike od petog do desetog, a treća vlastita frekvencije siječe deveti i deseti harmonik. Iz dijagrama je također vidljivo da u području nominalne brzine vrtnje niti jedna vlastita frekvencija ne siječe pravce harmonika, iz čega slijedi da se lopatica turbine pri nominalnoj brzini vrtnje neće pobuditi ni s jednim harmonikom.

from the third to the tenth speeds. Hence, at each turbine startup or shutdown, the exciting harmonics that have interference with the blade's first natural frequency curve will have an opportunity to excite the first mode shape. The curve of the second natural frequency intersects all harmonics from the fifth to the tenth, while the curve of the third natural frequency intersects only the ninth and the tenth harmonics. It has to be pointed out that at the rated running speed, neither natural frequency has interference with the harmonic lines.



Slika 6. Campbellov dijagram, oštećenje pri $\frac{1}{2}$ visine lista
Figure 6. Campbell diagram, damage at $\frac{1}{2}$ of the airfoil height

3.3. Oštećenje lopatace pri vrhu lista

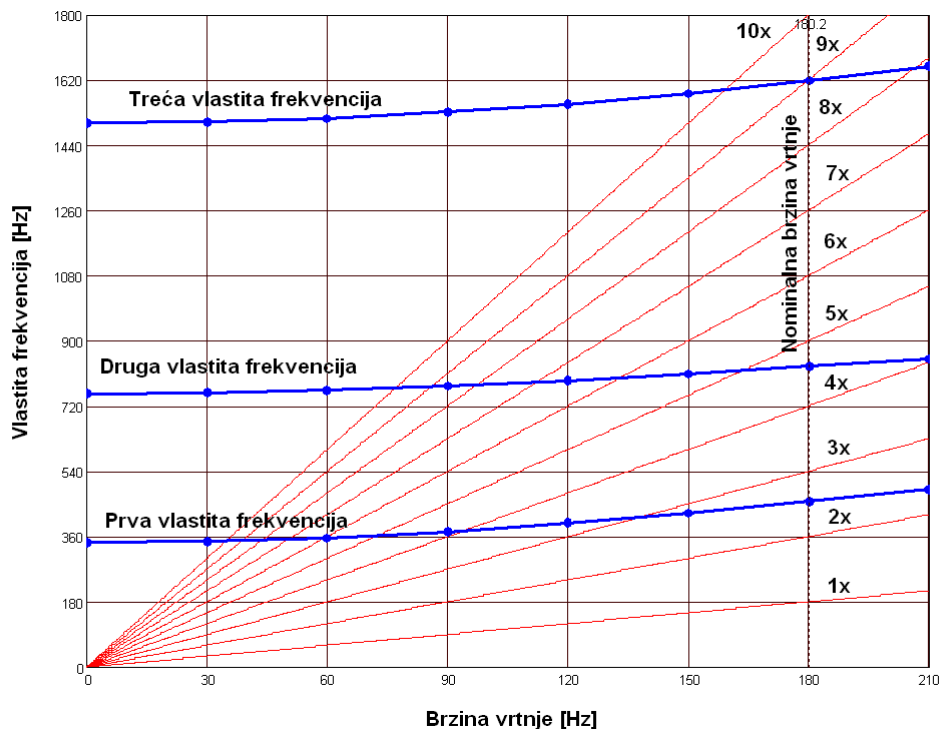
Treće oštećenje definirano je na visini od dvije trećine visine lista i maksimalne dubine ne veće od jedne četvrtine širine lista lopatice, slika 4c.

Slika 7. prikazuje Campbellov dijagram za opisani slučaj oštećenja lopatice. Kao i kod oštećenja pri korijenu, može se uočiti da se u području nominalne brzine vrtnje treća vlastita frekvencija može pobuditi devetim harmonikom, te na tu pojavu treba svratiti pozornost.

3.3. Damage of the blade airfoil near the blade tip

The third damage is assumed near the airfoil tip (at the two thirds of height) and with a maximum depth smaller than one quarter of the profile width, Figure 4.c).

Figure 7 shows a Campbell diagram for a described case of blade damage. As in the case of damage at the blade root, interference of the third natural frequency with a ninth harmonic can be expected.



Slika 7. Campbellov dijagram, oštećenje pri $\frac{1}{2}$ visine lista
 Figure 7. Campbell diagram, damage at $\frac{1}{2}$ of the airfoil height

4. ŽIVOTNI VIJEK DO ZAMORA LOPATICE

Postupak određivanja životnog vijeka do zamora lopatice obuhvaćao je dva dijela; izračun raspodjele naprezanja na osnovi izračuna odziva harmonijskih vibracija u programu *NX Nastran 6* te unos navedene raspodjele naprezanja u program *MSC.Fatigue* i izračun životnog vijeka do zamora lopatice na osnovi definirane funkcije promjene polja naprezanja.

Program *NX Nastran* nudi više rješavača za izračun raspodjele naprezanja, između ostalih i one za statičku analizu i frekvencijski harmonijski odziv. Budući da je predmet analize lopatica parne turbine, koja je podložna raznim vibracijskim uzбудama, u radu je korišten rješavač za direktni frekvencijsko-harmonijski odziv. Premda Campbellovi dijagrami za sva 4 analizirana slučaja pokazuju da se prva vlastita frekvencija nalazi točno između uzbude od 2x i 3x brzina vrtnje, razmatran je slučaj izravne pobude prve vlastite frekvencije od strane jednog od navedenih harmonika. Radi jednostavnosti izuzete su volumne sile koje odražavaju efekt ukrućivanja lopatice zbog generiranja centrifugalne sile te stacionarni pritisak pare na lopaticu. Dinamička uzbuda lopatice uzeta je u vidu koncentrirane sile u iznosu od 7 % u odnosu na stacionaran pritisak pare

4. FATIGUE LIFE OF THE BLADE

The procedure of blade fatigue life estimation was divided in two parts, calculation of stress field according to frequency harmonic response calculation in the software *NX Nastran 6*, and import of the obtained stress field into the software *MSC.Fatigue* and the calculation of fatigue life according of defined stress variation function.

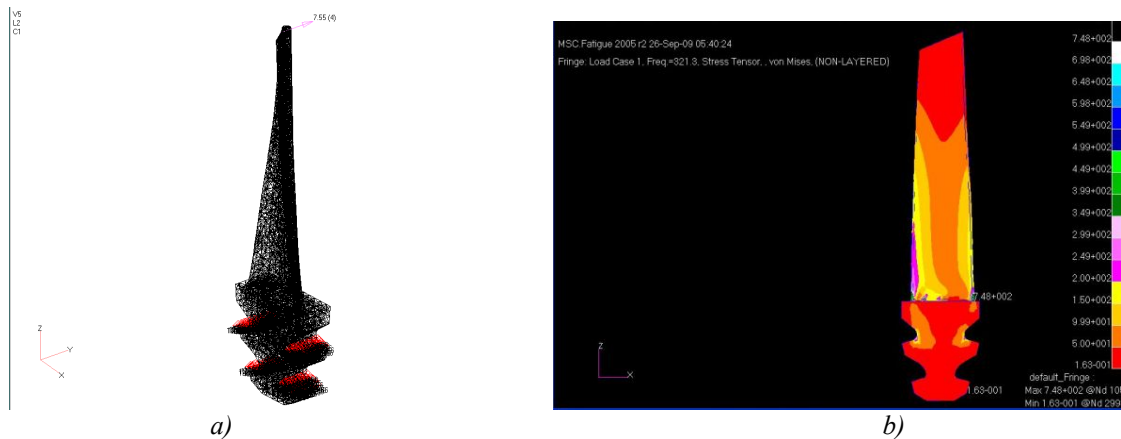
The software *NX Nastran* offers many different solutions for stress field calculation among others, Static and Frequency Harmonic Response. As the subject of analysis was the steam turbine blade which was susceptible to various vibrational excitations, in this work the direct frequency harmonic response solver was used. Although Campbell diagrams for all 4 analyzed cases show that the first natural frequency lies exactly in between the excitation of 2x and 3x running speed, it was considered to be the case when direct excitation from one of the two mentioned harmonics will happen. For the sake of simplicity, bodily forces which are responsible for the stiffening effect due to centrifugal force generation as well as stationary steam pressure on the blade are not taken into consideration. Dynamic excitation of the blade is taken in the form of nodal force in the amount of 7 % of stationary steam pressure on the blade. The excitation

na lopaticu. Za zadane režime rada turbine procijenjena je amplituda uzbudne sile, reducirane na vrh lopaticu u tangencijalnom smjeru u iznosu od 7,55 N. Slika 8.a prikazuje definiciju uzbudne sile dok je na slici 8.b prikazana odgovarajuća raspodjela naprezanja za neoštećenu lopaticu pri rezonantnoj uzbuli uz omjer prigušenja 0,01.

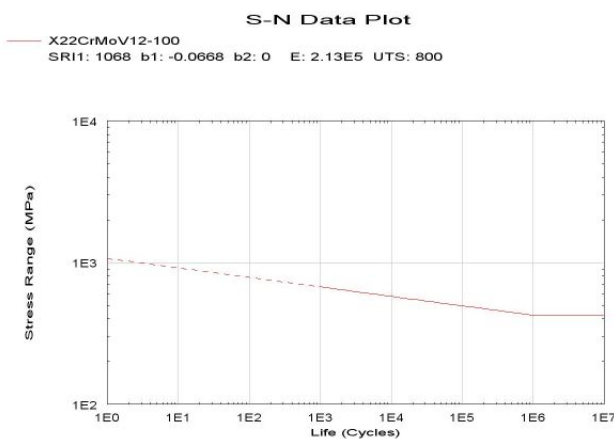
Dobivenu raspodjelu naprezanja zatim je trebalo unijeti u program *MSC.Fatigue* gdje je za potrebe S-N analize bilo dodatno potrebno definirati S-N (Wöhlerovu) krivulju korištenog materijala [6], slika 9. Uvjeti opterećenja definiraju se pomoću dodatne funkcije (slika 10) koja ima smisao varijacije postojećeg polja (raspodjele) naprezanja. Program zatim određuje broj ciklusa koji lopatica izdrži do puknuća. Rezultat manji od 10^3 ciklusa smatra se nevažecim s obzirom na navedenu teoriju. Za sva 4 analizirana slučaja (neoštećena lopatica, lopatica s oštećenjem do korijena, na sredini i pri vrhu

force was estimated, for a given turbine operating condition normalized to blade tip position in tangential direction, to be 7,55 N in magnitude. Figure 8.a) shows excitation force definition whereas Figure 8.b) presents the respective stress field for an undamaged blade at the resonant excitation with a damping ratio of 0,01.

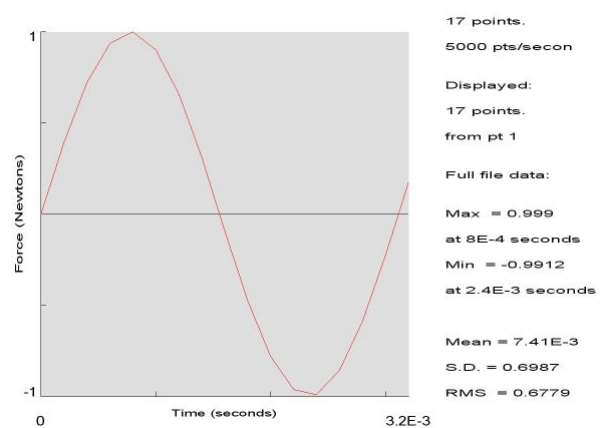
The resulting stress field now has to be imported into the software *MSC.Fatigue* where, for the purpose of simple S-N analysis, it has to additionally define the S-N (Wöhler) curve of the applied material [6], Figure 9. The loading conditions have to be defined by means of an additional function (Figure 10) which has a meaning of variation of the given stress field. The software then calculates the number of cycles that blade survives before failure. An outcome smaller than 10^3 cycles is considered not valid regarding the mentioned theory. For all analyzed cases (undamaged blade, blade airfoil damaged near blade root, at the middle and near



Slika 8. Neoštećena lopatica: a) Definicija uzbudne sile, b) Raspodjela Von Mises naprezanja
 Figure 8. Undamaged blade: a) Excitation force definition, b) Von Mises stress distribution



Slika 9. S-N (Wöhlerova) krivulja materijala lopaticice
 Figure 9. S-N (Wöhler) data plot of blade material



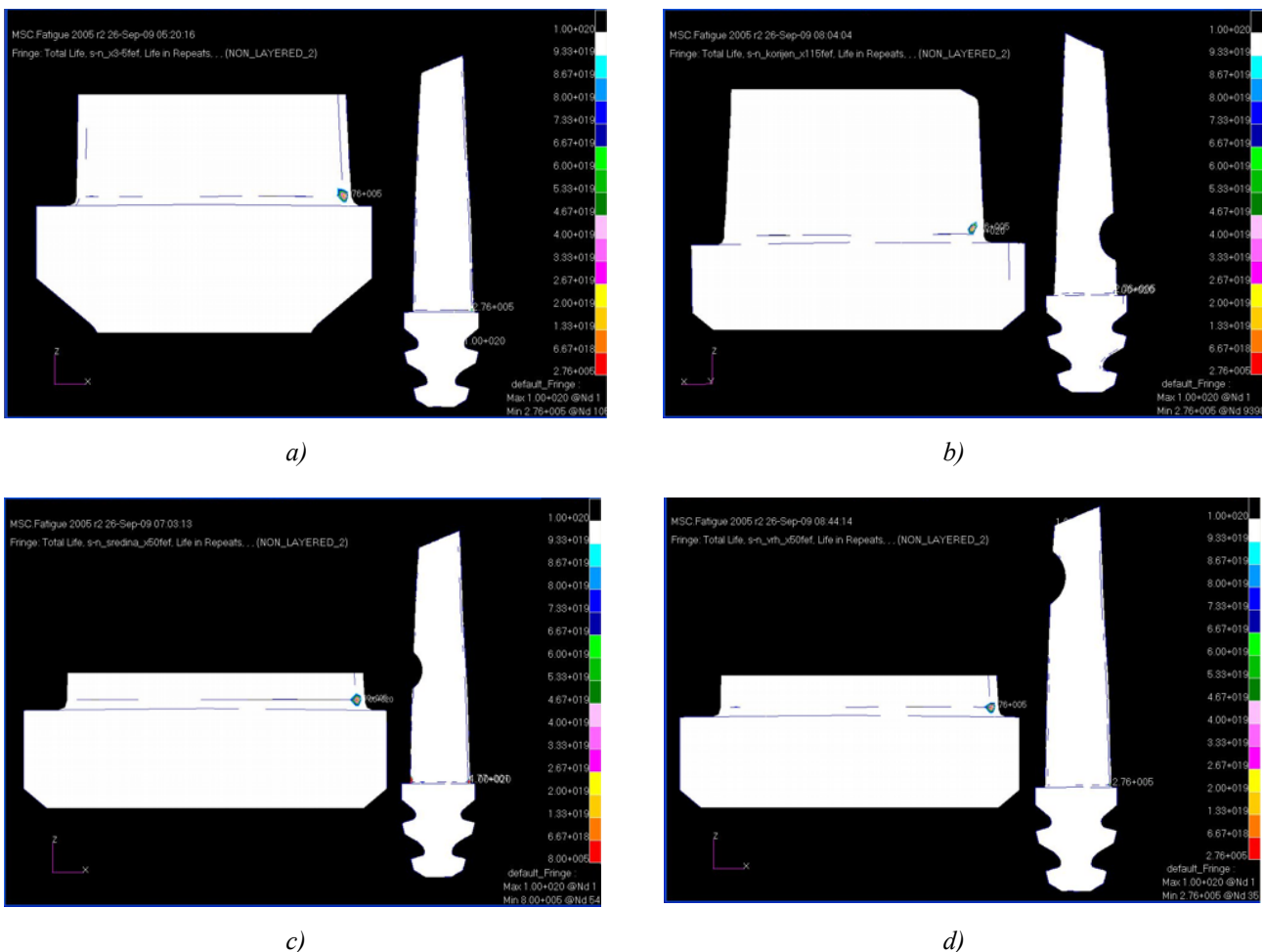
Slika 10. Funkcija promjene opterećenja u *MSC.Fatigue*
 Figure 10. Loading function in *MSC.Fatigue*

lista) rezultati pokazuju da neće doći do puknuća zbog primijenjene uzbudne sile. Ako bismo unesena naprezanja uvećali za određeni broj, došlo bi do puknuća lopatice i to u pravilu u zoni spoja lista i korijena. Slika 11 prikazuje kritična mjesta na kojima bi najprije došlo do puknuća kako neoštećene tako i oštećenih lopatica.

Rezultati izvršene analize upućuju na to da je utjecaj dinamičkog opterećenja, kako je to ranije opisano, na životni vijek lopatice zanemariv i životni vijek je praktički neograničen. Vijek trajanja do zamora lopatice ovisi o povijesti radnih uvjeta, utjecaju radne okoline te stacionarnim i vibracijskim naprezanjima. Stoga je točna procjena životnog vijeka u realnim uvjetima teško izvediva.

the blade tip), results show that failure won't happen due to applied excitation force. If someone multiplies the loading function by some number, the blade failure will occur ordinarily in the area of the junction of the airfoil and the root. Figure 11 presents the critical positions for blade failure as well as for the undamaged and damaged blades.

The results of the performed analysis indicate that the influence of dynamic loading on life, as explained earlier, can be neglected so that the life is practically unlimited. The fatigue life of a blade depends on the operational history, the effect of the environment and the steady and vibrational stresses. Therefore, the accurate prediction of lifespan in a real operational mode is difficult to determine.



Slika 11. Proračunati životni vijek u ciklusima / pri povećanju nominalnog opterećenja n puta za, a) neoštećenu lopaticu; $2,76 \cdot 10^5$ ciklusa / 3,5 puta, b) oštećenu pri $1/4$ visine lista; $2,76 \cdot 10^5$ ciklusa / 115 puta, c) oštećenu pri $1/2$ visine lista; $8 \cdot 10^5$ ciklusa / 27,5 puta, d) oštećenu pri $2/3$ visine lista; $2,76 \cdot 10^5$ ciklusa / 45 puta

Figure 11. Calculated life in cycles / at overload in index n times rated load for, a) undamaged blade; $2,76 \cdot 10^5$ cycles / 3,5 x, b) blade damaged at the $1/4$ of the airfoil height; $2,76 \cdot 10^5$ cycles / 115 x, c) blade damaged at the $1/2$ of the airfoil height; $8 \cdot 10^5$ cycles / 27,5 x, d) blade damaged at the $2/3$ of the airfoil height; $2,76 \cdot 10^5$ cycles / 45 x

5. ZAKLJUČAK

Razmatranjem Campbellova dijagrama za neoštećenu lopaticu može se zaključiti da će imati miran rad, zato što je malo vjerojatno pobuđivanje prvih triju vlastitih formi vibriranja.

Razmatrajući Campbellove dijagrame za nerealno velika oštećenja, pokazalo se da nema većih poremećaja u radu. Tek je moguće pobuđivanje treće vlastite frekvencije s devetom harmonikom čiji je utjecaj zanemariv.

Rezultati izvršene analize neoštećene lopatice upućuju na to da je utjecaj dinamičkog opterećenja, kako je to ranije opisano, na skraćivanje životnog vijeka zanemariv te da je životni vijek praktički neograničen. Čak i kod lopatice s velikim oštećenjem lista, dinamička sila ne bi uzrokovala nikakvo znatnije smanjenje životnog vijeka. Vijek trajanja do zamora lopatice općenito ovisi o povijesti radnih uvjeta, utjecaju radne okoline te stacionarnim i vibracijskim naprezanjima. Stoga je točna procjena životnog vijeka u realnim uvjetima teško izvediva.

LITERATURA REFERENCES

- [1] Elčić, Z.: *Parne turbine*, ABB, Karlovac, 1995.
- [2] Rieger N. F.: *Rotordynamics 2; Problem in turbomachinery*, CISM - Udine, Springer – Verlag, Wien, 1988.
- [3] Cazin, D.: *Analiza oštećenja lopatice zadnjeg stupnja rotora parne turbine*, Diplomski rad, Tehnički fakultet, Rijeka, 2009

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5. CONCLUSION

From the Campbell diagram for an undamaged blade, it can be concluded that it should have calm operation because it is unlikely that excitation of the first three normal modes will happen.

Analyzing Campbell diagrams for unusually heavy damage, it became apparent that there aren't any greater disturbances in operation. Just a possible excitation of the third natural frequency by a ninth harmonic that is negligible.

Results of the performed analysis of the undamaged blade show that dynamic loading, as described in the paper, has a negligible influence on the reduction of the fatigue life so that life is practically unlimited. Even in a case of blades with great damage of the airfoil, a dynamic load wouldn't cause any significant reduction of the fatigue life. The fatigue life generally depends on the operational history, the effect of the environment and the steady and vibrational stresses. Therefore, the accurate prediction of life in a real operational mode is difficult to determine.

- [4] Genta, G.: *Vibration of structures and machines, Practical aspects*, Springer-Verlag, New York, 1999
- [5] *MSC.Fatigue User's Guide*, MSC.Software Corporation, 2005
- [6] Nogde M., Hack M.: *An online algorithm for temperature influenced fatigue life estimation: stress-life approach*, International Journal of Fatigue, Vol. 26 (2004), p. 163-171

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