EAN 2017
55th conference on experimental stress analysis

Experimental verification of the mechanical properties of the material of the rotor shaft of the turbine damage

M. Pástor [1,a], F. Trebuňa [1], F. Šimčák [1], M. Buršák [2]

[1] Technical University of Košice, Faculty of Mechanical Engineering, Department of Applied Mechanics and Mechanical Engineering, Letná 9, Košice, 04200, Slovakia

[2] Technical University of Košice, Faculty of Mettalurgy, Institute of Materials, Letná 9, Košice, 04200, Slovakia

[a] miroslav.pastor@tuke.sk, [b]frantisek.trebuna@tuke.sk, [c]frantisek.simcak@tuke.sk, [d]marian.bursak@tuke.sk

Keywords: mechanical properties, fatigue test, lifespan.

Abstract. Analysis of lifespan of machines and equipment is connected with good knowledge of mechanical properties of used materials. In case of failure is necessary, beside others, to verify, if the material used in structure fulfill all demanded mechanical as well as fatigue properties. In the paper is described a procedure for experimental verification of mechanical properties of material of steam turbine rotor shaft after its damage. The results of measurements document that shaft material correspond to declared quality and mechanical as well as fatigue properties correspond to state of heating processing of given steel.

Introduction

The heating turbines work in cycle, where working medium changes its phase. The most spread is R-C cycle, where the working medium is water. In that case, in turbine expands water steam. The steam turbines are always parts of bigger technological unit which has a specific purpose (production of electricity, heat, work on the shaft, steam for industrial purposes and so on). From the purpose and properties of technological unit results basic parameters for design of turbine, e.g. power, rotations per minute, or special demands like power characteristics of turbine and so on [1,2].

The structural design of steam turbine includes problems of strength, stiffness and shape, oil systems, foundation of turbine, clutches, gear boxes. Very important are questions connected with purpose, using and service of the turbine. The steam turbines are equipped by hydraulic and electric equipment that serve for regulation, measurement and diagnostics of turbine. Design of turbine consists of many classical machine parts (bearings, oil system, clutches, valves, bolted joints, etc.) that can be computed by standard methods, they exhibit high level of unification and they are bought by turbine producers on the market.

One of critical part of steam turbine is a rotor shaft. From the design point of view, the rotor can be made with assembled rotors, forged as one piece, made as welded disc, welded drum, or combined [3]. The rotor with assembled wheels has simple structure from the point of design as well as technology. Its dimensions are obviously limited by shaft length and sometimes such a rotor cannot be used. The strength of turbine shaft is checked for torque which is the biggest one in location of connection between turbine and generator or powered machine. In the location of joint between turbine and electric generator, the stress in shaft is checked under conditions of short circuit of generator, when the torque increases in comparison to common operation several times [4,5]. Failure analysis of steam turbines is performed in terms of microstructure and mechanical properties of the material, fatigue tests, fractography, and so on [6,7,8].

Description of turbogenerator damage

The condensation steam turbogenerator before damage is shown in Fig. 1. Analyzed turbine rotor is made of alloy steel, forged as one piece (Fig. 2). The blades processed from anti-corrosive steel are assembled to the discs by special joints.

Fig. 1 Condensation steam turbogenerator [3].

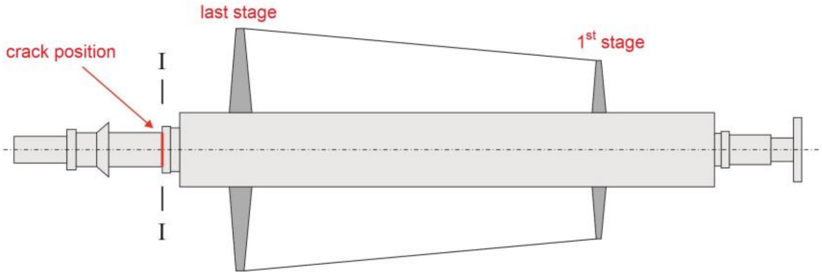


Fig. 2 Shaft of turbine rotor.

On the side of steam inlet on the end of rotor, the shaft consists of mechanical equipment for regulation of rotations. On the outer side of rotor, near mechanical equipment for rotation regulation are positioned cams of starting equipment for axial shifting of rotor. In case of excessive rotor shifting due to stochastic loading, or due to excessive wearing of axial bearing, these cams act as starting equipment and they activate stopping of the equipment. The rotor is positioned and led in two radial bearings and one axial bearing with pressure lubrication in oil bath. The turbine rotor is connected with the generator by reduction transmission box.

Cross-section of rotor in location of damage is loaded during operation of steam turbine by bending moment due to weight of rotor as well as by torque. The torque in the location of damage was caused by torsion moment of steam turbine. For the nominal rotations 8200 rpm, guaranteed power on generator (on terminals) 6 MW and efficiency 0.8, is the power of steam turbine after considering power lost on the rotation reductor equal cca to 7,5 MW. For nominal rotations 8200 rpm is the torque 8,74 kNm. Maximum stress due to torsion under nominal power in location of notch is 37,4 MPa. It results that such values of stress cannot lead to the failure state. State of equipment – steam turbogenerator after damage is given in Fig. 3.

Fig. 3 View to steam turbogenerator after failure.

Mechanical energy was transmitted from turbine to the reduction gear by shaft connected with flange of turbine rotor shaft by bolts (Fig. 4).

a)  b) 

Fig. 4 a) Shaft connecting turbine rotor and reduction gear, b) view to the side of the shaft consists of a pipe.

The right side in Fig. 4a represents broken part of turbine rotor shaft from the location damaged by fatigue fracture, while the outer diameter of full shaft is 120 mm in the location of fracture. The left side shows a shaft that consists of a pipe (Fig. 4b) on the end of which is a flange of clutch. View to the second part of clutch (from the side of gear) under aluminium cover is shown in Fig. 5.

Fig. 5 Part of clutch from the side of reduction gear.

On the basis of photographs taken during inspection (Fig. 6) as well as views to damaged part of shaft with fractured surface (Fig. 7), plastic deformation near fracture location (Fig. 8), detail view to fracture surface (Fig. 9), but also shape of fracture surface can be stated that the failure of shaft is connected with fatigue process (concluded also by bright conchoidal fracture).

Fig. 6 View to broken part of turbine rotor shaft.

Fig. 7 View to fracture surface on the shaft on the side of turbine.

Fig. 8 Shaft with fracture surface.

Fig. 9 Detail view to fracture surface – with marked area of macrocrack.

Creation of circular part of fractured surface concludes that it is resulting from combination of bending and torsion where the initial crack was caused not only by high level of equivalent stresses resulting from combination of bending and torsion, but also by stress concentrator due to defective structure of material resulting from smithing, Fig. 9. Such defective part together with sharp angle of relatively non-traditional shape (Fig. 10) localizes the area of first failure of turbine rotor shaft. This area was selected for further analysis of shaft material.

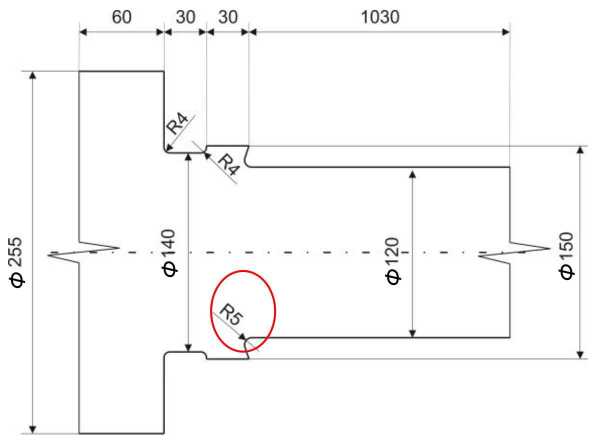


Fig. 10 Dimensions of turbine rotor shaft.

Fig. 8 documents that during crack spreading the pressing of material occurred which were invoked by cyclic loading resulting from rotation of shaft and axial deformation of flange clutch. In Fig. 11a is shown the separated part of turbine rotor from which a material for tests was taken. The material for mechanical and metallographic tests, see Fig. 11a, was taken according to plan given in Fig. 11b.

a) 

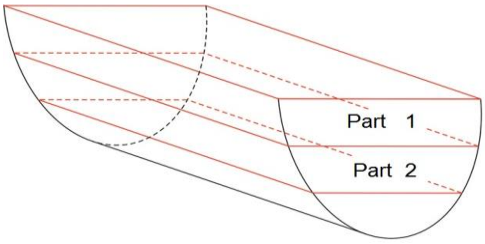
b) 

Fig. 11 a) Location on turbine rotor shaft from which was taken the material for tests, b) plan for preparation of specimens.

The segment of material separated by perpendicular section is shown in Fig. 12.

a) 

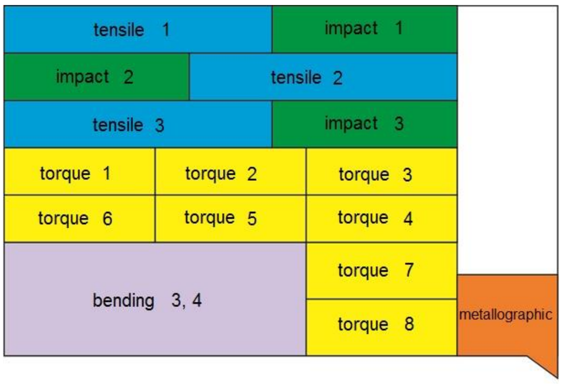
b) 

Fig. 12 Segment separated from damaged part of rotor.

On the left side of Fig. 12a is apparent area of fatigue failure and final crack caused by force. On the right side is a crack (Fig. 12b, failure in material) resulting from imperfect technology of shaft production. Strong plastic deformation along circumference of failure cross-section shows that in transversal macrocrack at the same time acted high normal stresses leading to plastic deformations during cyclic loading caused by rotation of turbine rotor.

The analysis of the mechanical properties of the material of the turbine shaft

The material for mechanical and metallographic tests was taken according to plan given in Fig. 13.

a) 

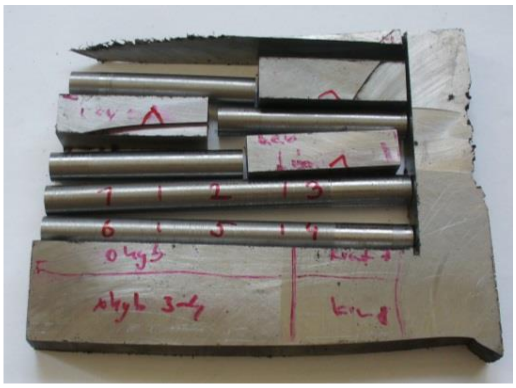
b) 

Fig. 13 Plan for preparation of specimens.

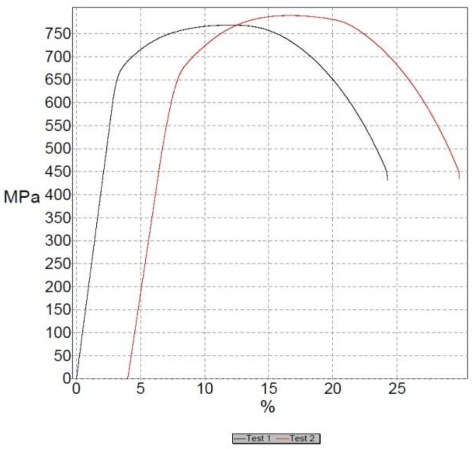
Tensile test

Basic mechanical properties were determined by static tensile test with specimens 6 x 30 STN EN 1000 2 -1 on the test machine ZWICK 1387 in accordance with treatment given in Standard STN EN 1000 2 -1. The measured results for given analysed material taken from broken shaft as well as properties of steel STN 41 6431.6 are given in Tab. No. 1. The testing velocity until reaching yield point of material $R_{p0,2}$ was 0,5 mm/min.

Table No. 1. Measured and prescribed mechanical properties of shaft material

Material 16431	Specimen	$R_{p0,2}$ [MPa]	R_m [MPa]	A_5 [%]	Z [%]
Measured value	1	645	777	22,3	71,4
	2	662	769	20,7	72,3
	3	613	789	22,3	73,4
Prescribed value	-	min. 569	686-834	20	-

Average value of yield point $R_{p0,2}$ was 640 MPa and strength R_m was 778 MPa. Shapes of tensile diagrams for specimens 2 and 3 are given in Fig. 14a, the test specimens after tests are shown in Fig. 14b.



a)

b)

Fig. 14 a) Tensile test diagram of specimens 2, 3, b) damaged test specimens 1, 2, 3.

Bending impact test

Toughness of material of damaged shaft was measured by impact test by bending for small bodies on the pendulum hammer PSW300 (Fig. 15) according to procedure given in STN EN 10045-1. The notches of shape Charpy „V“ and Mesnager „U2“ were used for crack initiation. The results of measurements are given in Tab. No. 2, which is completed with prescribed values for material 16 431.

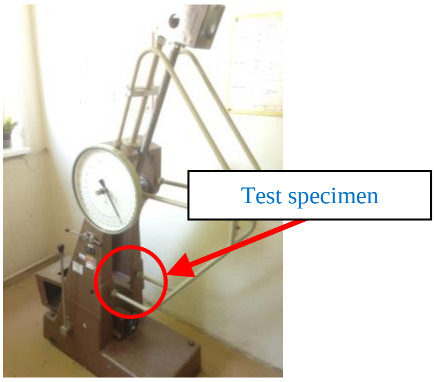


Fig. 15 Impact hammer PSW300 for toughness' test in bending for small specimens.

Table No. 2 Impact toughness of shaft material

Specimen	Impact work KV [J]	Toughness KCV [Jcm^{-2}]	Specimen	Impact work KU [J]	Toughness KCU [Jcm^{-2}]
1	169	211	4	201	251
2	186	233	5	196	245
3	167	209	6	194	243
Prescribed value	-	-	-	-	78

Average value of toughness with notch „V" is KCV=218 Jcm^{-2} and with notch "U2" is KCU = 248 Jcm^{-2}. Damaged test specimens after impact bending test are shown in Fig. 16.

Fig. 16 Damaged test specimens after impact test in bending.

The measured results show that the toughness of material fulfills demands given in material standard for the steel STN 41 6314.

Hardness

Hardness of shaft material was assessed by measurement of hardness according to Brinell by procedure given in STN EN 10003 – 1 (STN 42 0371). Prescribed value of the hardness is according to catalogue of steel STN 41 6314 HBS 208-253. Measured values were in range from 247 to 257 HBS.

Metallographic documentation

The samples were taken from damaged shaft for the metallographic scratch pattern in order to have possibility to investigate structure along circumference of shaft, or structure perpendicular to the shaft axis (see Fig. 12b, Fig. 13a). The investigation was accomplished on light microscope LEITZ. Microstructure of shaft material is given in Fig. 17a and Fig. 17b.

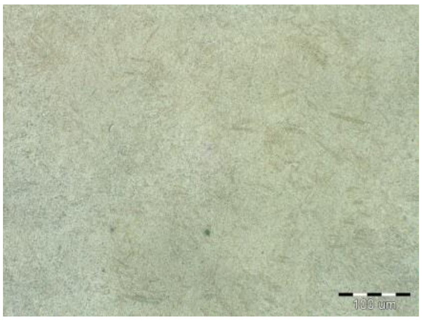
a)  b) 

Fig. 17 a) View to microstructure in direction of shaft, b) view to microstructure in direction perpendicular to the axis of shaft.

Microstructure consists of tempered martensite-sorbit that arose out during tempering of martensite over temperature 600°C. In general, the highest fineness of sorbit, the highest strength and hardness [9]. The documented state shows that the technology of temperature processing was appropriate.

Fatigue

The characteristics of fatigue properties were assessed by fatigue equipment PWO 0192N, product of company Carl Schenk, Fig. 18. The fatigue characteristics such as fatigue limit were assessed in conditions of torsion and plane bending. The shape of used test specimens is given in Fig. 19 for torsion and in Fig. 20 for plane bending. The diameter of test specimen for torsion test was 6 mm. The thickness of test specimen for bending test was 4 mm and its width 10 mm. The coefficient of cycle asymmetry was R = -1. The test was accomplished by Standard STN 42 0363.

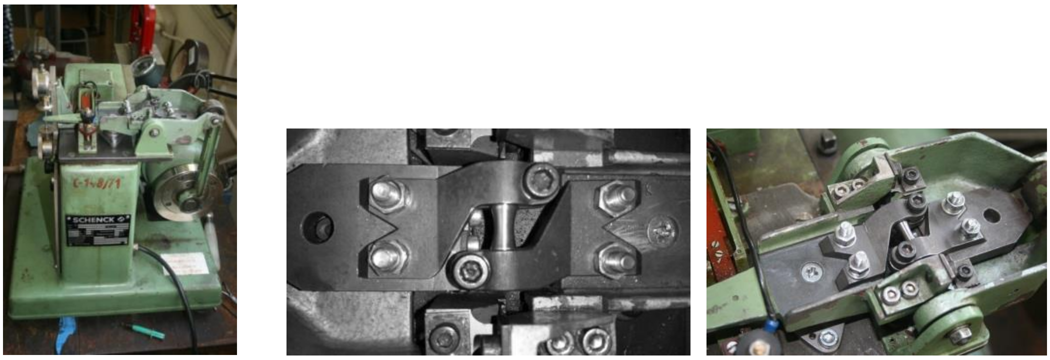


Fig. 18 View to microstructure in direction of shaft.

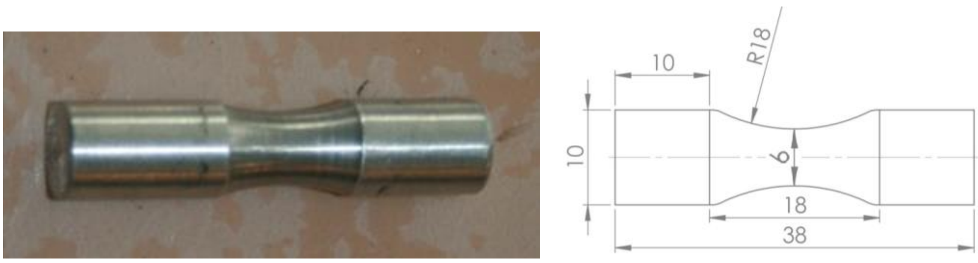


Fig. 19 Test specimens for torsion.

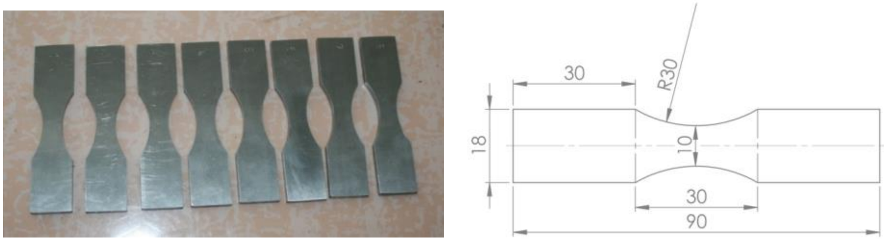


Fig. 20 Test specimens for plane bending.

Measured fatigue characteristics for torsion test are given in Tab. No. 3, and for the plane bending test in Tab. No. 4.

Table No. 3 Number of cycles to failure gained from measurements of torsional loading.

No. specimen	**2**	**6**	**7**	**4**	**1**	**9**	**5**	**3**	**8**	**10**
Stress [MPa]	150	174	181	186	205	214	237	258	274	291
Number of cycles [N]	10^7	10^7	10^7	981800	845600	335600	2081300	648600	742300	342600

Table No. 4 Number of cycles to failure gained from measurements of plane bending.

No. specimen	**2**	**5**	**4**	**1**	**3**	**7**	**8**	**6**
Stress [MPa]	318	330	348	368	393	405	411	427
Number of cycles [N]	10^7	10^7	473000	731200	522200	412000	254000	174800

Data scattering can be joined with specific location of specimen take off from the turbine rotor shaft, which was loaded over yield point of material (see Fig. 8 and Fig. 13).

Summary findings from the experimental determination of the mechanical properties of the material of the turbine rotor

Mechanical properties were determined by tensile tests. The tensile tests were realized on three specimens and there were determined magnitudes of yield point $R_{p0,2}$, strength R_m as well as plastic properties A5 and Z. Average values from three specimens are $R_{p0,2}$=640 MPa and R_m=778 MPa. These values created a base for the assessment of shaft strength under normal and emergency conditions.

Further test was realized for impact in bending. The tests were accomplished with notch of shape V and U. The tests were provided on three specimens for every of notch type. Material toughness of specimens with notches V and U did not exceed 209 J/cm^2, so the values prescribed by standards are fulfilled.

The hardness was in interval of prescribed values, too. Microstructure of material was determined in axial and transversal direction of shaft. Microstructure is created by tempered martnesite-sorbit that arise from steel tempering over 600 °C. In general, the highest fineness of sorbit, the highest strength and hardness. The documented state shows that the technology of temperature processing was appropriate.

The fatigue limit was measured for variable symmetric bending and torsion loadings. For the measurement were used 10 specimens for each type of loading and from the maximum measured amplitudes was determined number of cycles to damage. The measured values were used for creation of Wöhler curves from which is apparent fatigue limit for bending $\sigma_{co}^{*} = 330$ MPa and fatigue limit for torsion τ_c = 181 MPa. The ratios σ_{co}^{*}/R_m and τ_c/R_m corresponds to ratios determined by Research Institute of Materials in Prague [3].

On the basis of realized material analysis can be stated that:

a) Structure of damaged shaft corresponds to steel 16 431.6, while the prescribed technological procedures for tempering of given steel were obeyed [10].
b) Mechanical and fatigue properties correspond to the demands of thermal processing of given steel.
c) One of the reasons of failure can be a defect – inclusion in material that is a result of faults in production technology of shaft. This results to the stress concentration and initiation of fatigue process of failure of turbine rotor shaft in location where cumulated influences of notch as well as inclusion in material were.

Conclusion

The conclusions concerning reasons of failures and the answers to individual questions were elaborated on the basis of several visual inspections, analysis of data given by operator, but mainly on the basis of extensive experimental tests, chemical composition of material, determination of mechanical properties, assessment of microstructure, determination of mechanical properties under fatigue tests, extensive numerical and analytical modelling of tha shaft loading in location of damage. Definitely, the authors came out from general rules used for the design of steam turbines. The knowledge of educational institutions as well as production firms were used in formulation of conclusions.

From the analysis of fracture surface from the point of view of failure mechanism can be stated that despite of variable loading by torsion there was no clam-shell structure in the spreading crack in critical cross-section of shaft at circumferential direction, but the crack was non-concentric. From detailed analysis of fractured surface was found out that the location with the highest deepness of fatigue failure contains a crack in circumferential direction that con not be connected with loading, but it is joined with inappropriate production technology of turbine rotor shaft. With the highest probability, there was a failure as residuum from smiting. This error in the structure led to the crack towards surface and accordingly led to sharp notch which was reason of turbine shaft damage.

Acknowledgement

This paper was supported by Projects VEGA No. 1/0751/16 and VEGA No. 1/0393/14.

References

[1] P. Heinz, B. Murari, P. Singh, Steam Turbines: Design, Applications, and Rerating, Second Edition. New York, 2009 The McGraw-Hill Companies, Inc., ISBN 9780071508216.

[2] D. Japikse, Introduction to turbomachinery, 1997. 2. vydání. Oxford: Oxford University Press, ISBN 0–933283-10-5.

[3] F. Trebuňa et al., Posúdenie technických a prevádzkových parametrov turbogenerátora, vyhodnotenie príčin kolapsu, materiálová analýza súčiastok a konzultácia budúcich technických riešení, Final Report, TUKE SjF, Košice, 2015, 137 p.

[4] F. Trebuňa, P. Bigoš, Intensification of technical competence heavy supporting structures. Vienala, Košice, 1998.

[5] F. Trebuňa, F. Šimčák, Odolnosť prvkov mechanických sústav. EMILENA, Košice, 2004.

[6] S. Barella, M. Bellogini, M. Boniardi, S. Cincera, Failure analysis of a steam turbine rotor. Engineering Failure Analysis 18 (2011) 6, 511-519.

[7] Z. Mazur, R. G. Illescas, J. A. Romano, N. P. Rodrigue, Steam turbine blade failure analysis. Engineering Failure Analysis 15 (2008) 1 - 2, 129-141.

[8] E. Poursaeidi, M. R. Mohammadi Arhani, Failure investigation of an auxiliary steam turbine. Engineering Failure Analysis 17 (2010) 6, 1328 - 1336.

[9] F. Trebuňa, M. Buršák, Medzné stavy – lomy. Grafotlač Prešov, Prešov, 2002.

[10] I. Fűrbacher, K. Macek, J. Steidl, Lexikon technických materiálů. Svazek 1, Verlab Dashőfer, 2005.