Condition Monitoring of Kaplan Turbine Bearings Using Vibro-diagnostics

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***Abstract*— Vibration diagnostics is an indispensable method to evaluate the technical condition of machinery. If the conditions of machines are regularly monitored, the problems can be corrected even before they arise. The most often overloaded components of the turbine are bearings that similarly like other machinery are subject to degradation wear. The article deals with the bearings´ condition of the Kaplan turbine. Within the research, the frequencies at which maximum accelerations were achieved and their impact on plant operation were observed. Time records were converted to the frequency domain by means of Fast Fourier Transformation and the records were processed by means of filters. Acceleration peaks have determined the frequencies responsible for the outer and inner ring damages, but based on an evaluation of the operating state of the bearing using Root Mean Square (RMS) values could be concluded that that damage to any part of the bearing is not demonstrable and the turbine is operability.**

***Index Terms*—diagnostics, vibration, turbine, bearing, operating conditions**

I. Introduction

Each machine, if it has to work reliably throughout its planned life, must be maintained. For all large and expensive equipment, operational life is an essential and often neglected part of the machine characteristics. Monitoring the vibration characteristics of a machine gives us an understanding of the 'health' condition of the machine. This information can help us to detect problems that might be developing. If the conditions of machines are regularly monitored, the problems can be corrected even before they arise. Because machine vibration monitoring finds potentially damaging vibration, a lot of time, money, and frustration can be saved. [1]

Machine condition analysis followed naturally from the early successes of vibration measurement and fault diagnosis in the 1960s. It is interesting to trace the job tasks involved with operating machines including maintenance, reliability, and life expectancy. The sequence of tasks following vibration measurement involves signal processing, fault analysis, condition evaluation, and prognosis. Condition monitoring began with routes using handheld meters and hand recorded data. The advent of the microprocessor brought sophisticated data collectors that could record, trend, process, and store volumes of data. However, the high-level analysis still remains the forte of the vibration analyst. [2, 3]

II. State of the Art

Vibration diagnostics or called vibro-diagnostics is an indispensable method to evaluate the technical condition of machinery. Vibro-diagnostics of rotating machines and mechanisms is carried out to determine the technical condition of the rolling and sliding bearings, gears and piston hydraulic pumps, motors, reciprocating, screw and vane compressors, broken and unfortified foundation and others. Each machine has vibrations, but they must be within a certain standard set out in the relevant standards and technical specifications. By vibro-diagnostics are measured vibrations and are identified different types of malfunctions during operation of the machines, namely: unbalance, abnormal alignment between shafts, presence of gaps in the camps, analysis of the rolling and sliding bearings, analysis of the gear, belt and others. gears, analysis of the state of electrical machinery and so on. [4]

Many papers dealt with vibro-diagnostics of rotational components and machines. A principle of spectral vibro-diagnostics for fatigue damage of structural elements is described by Matveev [5]. Cereska [6] in his article

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employs a methodology for performing the analyses and vibro-diagnostics measurements of the mechanical-dynamic elements of mechatronic systems. Monkova [7] realized strengthening of the station frame for motor testing based on vibration analyses results and so the operating mode of the station has been improved. Ziaran & Darula [8] in their paper presented basic procedures and methods used for determining the state of wear of high contact ratio (HCR) gear sets comparing frequency spectra and Cepstra recorded throughout their lifetime tests, through vibro-acoustic diagnostics. Budynkov [9] studied gas turbine engines by means of Cepstrum analysis within vibro-diagnostics. Sedmak & Veg [10] had developed a portable device for vibro-diagnostics and its service application for heavy-duty rotating equipment. Milovancevic et al. [11] used vibro-diagnostics to analyze pumping systems for waterworks.

The bearings ´condition of Kaplan turbine was studied within the article. In most cases, this turbine is used in hydroelectric power plants, where it is one of the most important parts of the plant. It is an overpressure turbine that is used for smaller gradients - up to 80 meters and for larger flow rates. Power regulation can occur in two ways. The first method of regulation is by means of the impeller blades and the second method is the possibility of turning the blades of the camshaft. The impeller and timing blades are rotated such that each blade position corresponds to a certain angle of the propeller blades. The propeller blades are rotated by means of a control rod that passes through the hollow shaft. The speed can be controlled by tilting the special brake blades, which extend automatically when the maximum permitted speed is exceeded, the coupling between the impeller blades and timing blades is disengaged or the flow is closed. The advantage of the turbine is that it achieves the highest speed of all turbines, even at small gradients. The heavyweight of the turbine makes it possible to use one turbine where two different types of turbines would have to be used. The disadvantage is problems with cavitation, which causes vibration and corrosion. The principle parts of the turbine are presented in Fig. 1. [12]

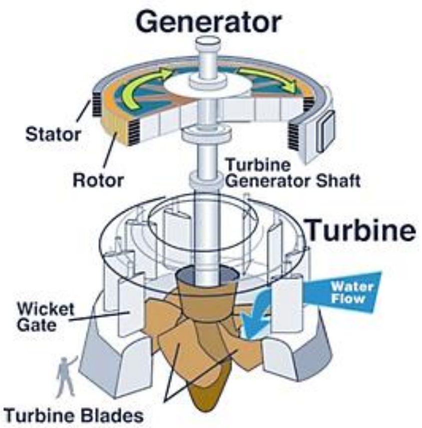

Figure 1. Kaplan turbine [12]

The most often overloaded components of the turbine are bearings that similarly like other machinery are subject to degradation wear. The wear of the bearings is caused by the interaction between the loads, the moving parts and the influence of the environment. From the point of view of vibro-diagnostics, it is possible to distinguish according to the methods used, at what stage of the damage the bearing is located. Damage is most often caused by improper assembly, storage, operating conditions and maintenance. What is important is the size and extent of the damage that determines whether the bearing has the required properties.

Rolling element bearings normally generate very little vibration amplitude with or without faults unless they are near failure. The fault frequencies are a function of the bearing geometry and the shaft speed. Depending on where the defect is located different waveforms will be generated. An outer race defect is stationary if the outer race is stationary and thus the rolling elements continuously roll through the defect yielding a more or less constant amplitude waveform. An inner race defect is rotating and will rotate through the load zone periodically yielding a modulated vibration signal. While impacts on the waveform are a sign of a bearing fault, the severity is difficult to determine from amplitude values. Judging severity has to be related to the frequencies in the spectrum. Fortunately bearing defects generate unique frequencies. To assess severity, one must examine the working mechanism of the bearing. [13,14]

The bearing constantly rotates through a loading zone where the Hertzian stresses will be maximum. For this reason, outer race defects are common. Cyclic stressing is only one cause of bearing failures. Lubrication, wear, design, environmental conditions, and installation can have a large effect on failure. However outer race defects are not the most fatal defects. Because of the fragility and location of the separator (cage), the run time to failure of such defects is unpredictable. Pieces of the separator are lodged between the rolling elements and the races and propagate other defects. The rolling elements are not as fragile but pieces of a metal shed from a defect can have the same effect as a cage failure. This all leads to a hierarchy of faults from the worst case when judging condition. [15]

Bearing failure frequencies are detected only in existing failures. Also, in the frequency spectrum, the failure of different bearing parts differs from one another. A failure on the outer ring occurs whenever the rolling element passes this failure. It also affects the inner ring at each pass, but the position and load also have an impact. A manifestation of rolling bearing failure in the frequency spectrum is shown in Fig. 2. [16]

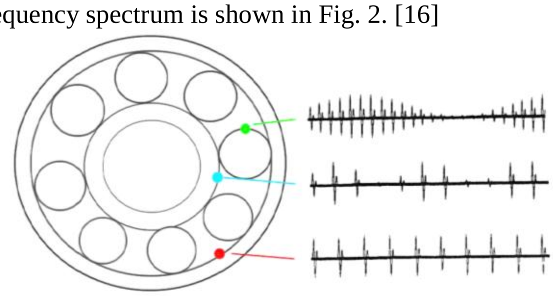

Figure 2. A manifestation of rolling bearing failure in the frequency spectrum (rolling element - up, inner ring – center, outer ring - down)

If the failure, when the rolling element passing, is in the loaded part of the bearing, the response is greater, what occurs once per revolution. The failure of the rolling element always appears in contact with the outer and inner rings and it also depends on the load. When measuring bearing vibrations, velocity is usually selected from three possible variables (position, speed, acceleration). This is because bearing failure frequencies occur in the high-frequency range (see Fig. 3). Since the deflection is most pronounced at low frequencies, the failure symptoms would be suppressed. Conversely, the measurement of acceleration could result in exaggerated accentuation. [17]

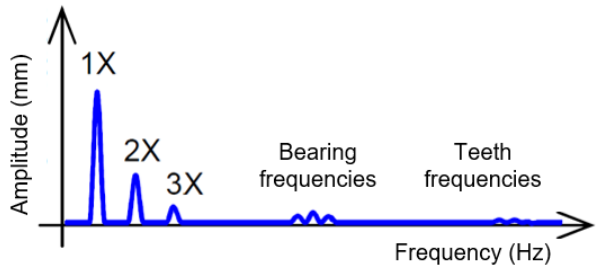

Figure 3. Position of teeth and bearing frequencies within a frequency spectrum

It is possible to locate damage of individual elements in the bearing (Fig. 4) on the basis of characteristic bearing frequencies specified on the bearing dimensions.

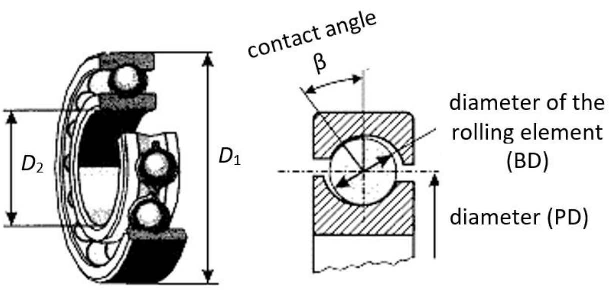

Figure 4. Elements of a bearing

The following formulas apply to kinematic frequencies are: [18]

- BPFO (Ball Pass Frequency of Outer Ring) – the frequencies responsible for damage to the outer ring:

$$f = \frac{N}{2}(1 - \frac{BD}{PD}\cos\beta)n \qquad (1)$$

- BPFI (Ball Pass Frequency of Inner Ring) – the frequencies responsible for damage to the inner ring:

$$f = \frac{N}{2}(1 + \frac{BD}{PD}\cos\beta)n \qquad (2)$$

- BSF (Ball Spin Frequency) – the frequencies responsible for damage to the rolling element:

$$f = \frac{PD}{2BD}\,(1 - (\frac{BD}{PD}\cos\beta)^2)n \qquad (3)$$

- FTF (Fundamental Train Frequency) – the frequencies responsible for damaging the bearing cage

$$f = \frac{1}{2}(1 - \frac{BD}{PD}\cos\beta)\,n \qquad (4)$$

where
BD – ball or roller diameter,
PD – Pitch diameter,
β – contact angle,
N – number of balls or rollers,
n – revolutions per second or relative speed difference between the inner and outer race

The formulas as described are valid for a bearing with a standing outer ring, otherwise, the sign in the calculation must be reversed (except for the rolling element calculation.

III. Methods and Measuring Equipment

A scheme of the set, on which the measurements were carried out, is in Fig. 5. An important issue within the measurement was a rotation of the outer and the internal turbine flaps and their inclination. The outer flaps of the turbine have been placed on the impeller and the inner flaps on the timing wheel. The turbine is connected to the generator by means of a shaft coupling, that is to say, the generator is essentially positioned perpendicular to the axis of the turbine.

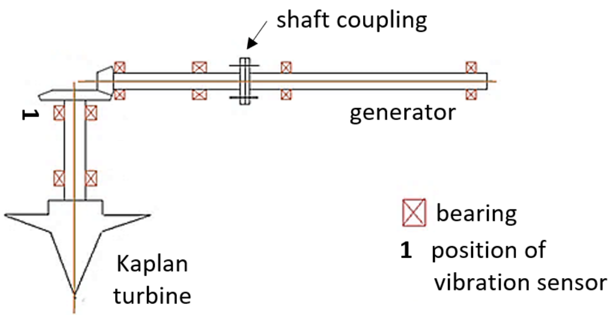

Figure 5. Schematic representation of the Kaplan turbine assembly

Measurements were carried out at a turbine power of 340 kW. During the measurement, it has been manipulated the rotation of the blades and found out how this affects the overall vibrations while increasing or decreasing the power, while the water flow through the turbine was constant. Changing the setting has changed the vibration amplitudes.

The vibrations were recorded at the bearing (point 1 in Fig. 3) in the time domain and the record was then processed and transformed into the frequency domain using the FFT method to obtain a dependence of amplitude on frequency. Undesirable noises were separated using filters.

The FFT analyzer of the CMXA 80 series (Fig. 6) was used for the measurement at the study of bearings behaviour. This device allows monitoring of the device status by analyzing vibration signals and variables using four channels.

Accelerometers MTN/1100 and MTN/1830 (Fig. 6) produced by Monitran company were also used at the experiments. They are characterized by small dimensions, side entry, constant current accelerometer for on-line and off-line vibration analysis. Because of their stainless-steel construction and robustness, they are suitable for use in harsh industrial environments.

Figure 6. Measuring equipment (from the left side: FFT analyzer CMXA 80, accelerometers MTN/1830 and MTN/1100)

An essential step in signal processing is filtering. Thus, by using filters, it is possible to adjust the signal to include only the information necessary for evaluation. There are a large number of filters that differ in the suitability of the application, as well as in the complexity and method of signal processing. Depending on the pulse response length, digital filters are divided into FIR (Finite Impulse Response) and IIR (Infinite Impulse Response) filters. These are then divided into other types according to permeability. [19]

The measured signal within the research was processed by a so-called envelope method with the Hilbert transformation that enables to create an analytical signal from a real signal.

IV. Results and Discussion

In order to assess the actual condition of the machines, to identify and locate their damage or emerging faults, it is necessary to analyze the frequency results. Using it not only information about the real speed range of revolution frequencies can be obtained, similar to the time-domain analysis, but the frequency analysis also points on types of problems.

As it was already said, the analysis can be divided into three main areas: low-frequency bands up to 5 Hz, middle frequency bands from 5 to 100 Hz and high-frequency bands. The high-frequency bands from the upper limit of the mid-frequency bands contain information about starting rolling bearing failures [20]. In bearings, surface fatigue of the bearing element material causes damage to the surface layer. It is possible to locate damage of individual elements in the bearing on the basis of characteristic bearing frequencies determined from the bearing dimensions.

On the bearing, where the vibration was scanned, the following frequency values were calculated for each damage type:

BPFO = 97.82 Hz
BPFI = 116.46 Hz
BSF = 40.11 Hz
FTF = 3.26 Hz

Figs. 7 – 10 represent processed dependencies of the accelerations in the frequency domain at four various inclination of turbine blades caused by their rotation, while the water flow was constant as already mentioned.

Fig. 7 shows the spectra in the range of 0 - 400 Hz. Within this range, it can be seen that between 5 and 20 Hz, the highest acceleration peak value is about 0.06g. A higher acceleration peak value is due to the generator starting up. In the 85 Hz, 115 Hz and 350 Hz areas, there are further peaks with acceleration values of 0.015 - 0.025g. Based on the calculated bearing damage frequencies, it can be seen that for a BPFO value of 97.82 Hz, an increased acceleration peak value is visible in a given spectrum, which may indicate a possible beginning outer ring damage. Other elevated values are likely due to vibrations on other parts of the equipment and should not affect operation.

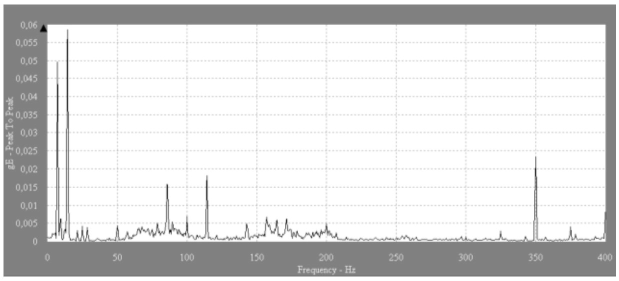

Figure 7. Measured accelerations at the first rotation of blades

In Fig. 8 it is possible to see the spectrum display in the range 0 - 800 Hz and in the range 0 - 0,18g for acceleration peaks. In this range, it can be seen that the highest value is in the range of 350 Hz and 0.16g. Furthermore, it can be seen that at the frequencies 300 Hz, 375 Hz, and 700 Hz, the acceleration also increased to 0.04g - 0.05g. At calculated values of frequencies for BPFO = 97.82 Hz and BPFI = 116.46 Hz, the values of monitored frequencies have increased approximately in 0.01g. This corresponds to incipient damage to the outer and inner rings.

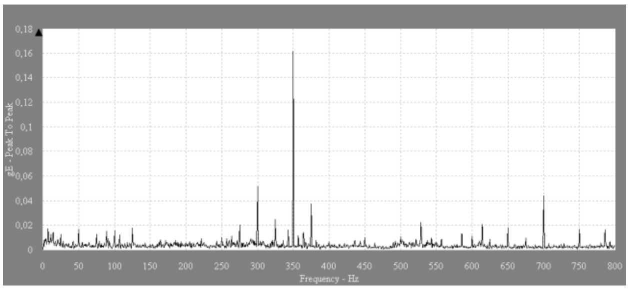

Figure 8. Measured accelerations at the second rotation of blades

At the third angle of the blades ´rotation, the frequency spectrum (in the range of 0 – 800 Hz) is a very similar character compared to this that is presented in previous Fig. 9. Maximal acceleration peak is also at the frequency of 350 Hz. At higher frequencies, the peak peaks did not exceed 0.04 g. In the 375 Hz and 700 Hz ranges, the value rose to 0.035 g, and in the 300 Hz range to 0.025 g.

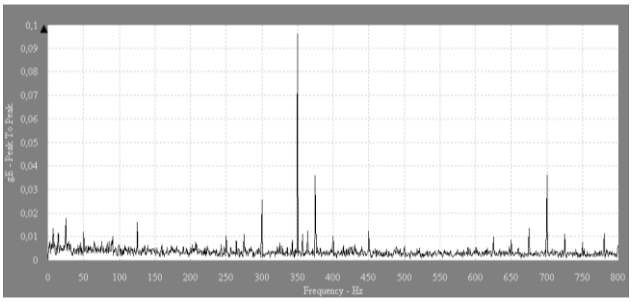

Figure 9. Measured accelerations at the third rotation of blades

At the fourth set-up of the turbine blades ´rotation, the acceleration reaches a maximum value of 0.0005 - 0.006g within the observed frequency range of 0 - 800 Hz (Fig. 10). The maximum value is in the range up to 10 Hz and its value is 0.006g. Deflections are repeated regularly and are at approximately the same level, so no deflection is significantly higher than others.

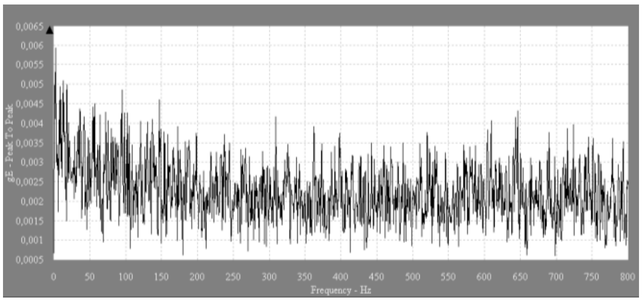

Figure 10. Measured accelerations at the fourth rotation of blades

Due to an evaluation of the results, RMS (Root Mean Square) values have been calculated. The RMS value is generally the most useful because it is directly related to the energy content of the vibration profile and thus the destructive capability of the vibration. RMS also takes into account the time history of the wave form.

Calculation of vibration velocity RMS in a given frequency band can be accomplished in many ways. The most intuitive approach is taking an acceleration time-domain signal, perform integration, detrend integrated signal and then use windowing and zero padding before FFT. Then the RMS value in the band of interest from obtained spectra can be calculated.

Although there are many methods for calculating RMS values, the standard method is still Calculus integration technique for continuous data stated as: [21]

$$f_{RMS} = \sqrt{\frac{1}{T}\int_{t_1}^{t_1+T} f^2(t)\, dt} \qquad (5)$$

where f_{RMS} is the RMS value of $f(t)$ between the domain interval of t_1 to $(t_1 + T)$, where T is the period of $f(t)$.
The dependency of RMS on frequency is presented in Fig. 11.

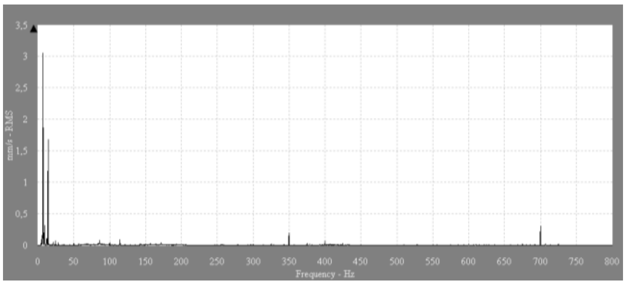

Figure 11. The dependency of RMS on frequency

The RMS values were compared with the limit values that determine the permissibility or impermissibility of vibration. The given limit values are given in Table 1. Based on a comparison of the highest achieved RMS value, given in mm/s, it is 3.1 mms^{-1} and for the power value of the analyzed Kaplan turbine 340 kW, it was possible to conclude that the RMS value is in band B. Zone B means permissible vibrations in unrestricted operation with increasing attention. [22]

TABLE I. LIMITED VALUES [15]

mm/s	< 15 kW	(15–75) kW	> 75 kW Solid unyielding	> 75 kW Solid yielding
0.28	A	A	A	A
0.45	A	A	A	A
0.71	A	A	A	A
1.12	B	A	A	A
1.8	B	B	A	A
2.8	C	B	B	A
4.5	C	C	B	B
7.1	D	C	C	B
11.2	D	D	C	C
18	D	D	D	C
28	D	D	D	D
45	D	D	D	D

V. CONCLUSIONS

Diagnostics is the art or act of identifying a condition from its signs or symptoms. Vibration signatures are extracted from measurements of the external casing (bearing casing) or shaft displacements contain information of the vibration response of the machine, which if correctly interpreted, could identify the condition of the machine.

Operating condition of Kaplan turbine bearings was investigated by vibration diagnostics. Within the research, the frequencies at which maximum accelerations were achieved and their impact on plant operation were observed. Time records were converted to the frequency domain by means of Fast Fourier Transformation and the records were processed by means of filters.

At the same time, the frequencies that were responsible for damage to different parts of the bearing were calculated on the basis of the bearing dimensions and these frequencies were subsequently compared with the measured records. The frequencies were measured at various Kaplan turbine blade rotations, while the turbine output was still constant at 340 kW.

When diagnosing the bearing with the first tilt of the blades, an acceleration peak was increased at BPFO = 97.82 Hz, which determines the frequencies responsible for the outer ring damage, which may indicate the beginning of the damage. Measurements at the second rotation of the blades also indicated the increased the acceleration peak values at BPFO = 97.82 Hz, which is responsible for the outer ring damage, and also at BPFI = 116.46 Hz, which is responsible for the inner ring damage. These values may indicate that the bearing is being damaged. Other deviations were caused by the influence of individual parts of the device.

After calculating the velocity and RMS values, the highest value was compared to the allowable limit, and the value of 3.1 mm/s was found to be within the permissible values for operation, but increased bearing attention should be paid and the measurement repeated more frequently. At that time, it was possible to conclude that damage to any part of the bearing was not demonstrable and the turbine was operability.

Conflict of Interest

The authors declare no conflict of interest.

Author Contributions

Conceptualization, methodology, supervision, measuring and article writing: K. Monkova. Measuring and equipment set-up: P. Monka. Resources: S. Hric. Validation and editing: D. Kozak and M. Katinić. Software and data processing: I. Pavlenko and O. Liaposhchenko.

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Marko Katinić was born on November 25, 1966, in Slavonski Brod, Croatia. He graduated at the Mechanical Engineering Faculty in Slavonski Brod, Josip Juraj Strossmayer of Osijek, Croatia in 1990. He achieved his titles Doctor of Philosophy in 2014 also at the Mechanical Engineering Faculty in Slavonski Brod. Thesis research was focused on the interaction of twin edge cracks in a finite plate subjected to the tension under creep condition.
After graduation, he worked for 2 years at an oil refinery as a Project Engineer. Then he worked for almost 22 years at a mineral fertilizer plant as a Chief Engineer for Turbines and Compressors. As a Lecturer, he worked for two years at the College in Slavonski Brod. He is currently employed as an Assistant Professor at the Mechanical Engineering Faculty in Slavonski Brod. His current research interests are fracture mechanics, mechanical vibration, rotor dynamics and vibration diagnostics. Dr. Katinić is a member of the Croatian Society for Mechanics and Croatian Society for Maintenance. He is a Certified Design Engineer.

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- implementation innovation engineering education by using 3D modelling, integrated engineering, rapid prototyping and reverse engineering, augmented/mixed reality and virtual reality (AR/MR/VR) technologies in the education process;
- mathematical modeling and numerical simulation, static and dynamic modeling, simulation-based research;
- CAD/CAE-systems for chemical engineering and plant design, SCADA process control system for the chemical industry;
- expert in an industrial inspection of oil and gas, chemical and petrochemical plants, alternative and renewable energy, bio-based fuels, food technologies.