The Effects of high vibration on the steam turbo-generator machine of the unit B1 339 MW in thermal power plant “Kosovo B” : [presented August 2009]

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Capstone Project

The Effects of High Vibration on the Steam Turbo-Generator Machine of the Unit B1 339 MW in Thermal Power Plant "Kosovo B"

Submitted as a Capstone Project in partial fulfillment of a Master's of Science Degree in Energy & Resource Development at the RIT Center for Multidisciplinary Studies

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ABSTRACT

This Capstone Project will address the analysis and diagnosis of the Steam Turbo-Generator (STG) vibration problems in the “Kosovo B” Thermal Power Plant (TPP) that are degrading the normal operations of the unit B1 339 MW. This Steam Turbo-Generator is a steam turbine directly connected to an electric generator for the generation of electric power and where unplanned shut downs can have catastrophic consequences for the production of electricity.

The purpose of my analysis of the high level of vibrations in the STG-B1 machine is to extend the life of the machine and preventing any catastrophic failures. This implies not only detecting if the machine is developing problems, but also to identify the specific nature of those problems or sources of vibration while the machine is running because most of the vibration problems are present during normal operations. While implementing my vibration analysis on the machine I hope to improve the reliability and longevity of the machine.

I will conduct vibration analysis on this machine by using **off-line** and **on–line** vibration instruments such as **VIBROMETER VM 600** (on-line monitoring vibration system), **VIBSCANNER** and **BRUEL & KJÆR data collector 2526** (off-line instruments) for monitoring and analyzing the machine’s condition and I will make recommendations on upgrades as a result of my findings.

I believe that analyzing and diagnosing the vibration issues on the STG-B1 machine will help in developing the steps for planning continuous maintenance and establishing the correct actions necessary to insure the continued, safe and productive operation of this machine.

1. INTRODUCTION

I would like to emphasize that this Capstone Project will not only demonstrate my ability to identify and solve real-world problems but I also hope it will be a very important project for TPP "Kosovo B" management and its business. I anticipate that they will take immediate corrective actions based on my recommendations and expertise as a machine diagnostician and maintenance engineer in TPP "Kosovo B".

The Thermal Power Plant, "Kosovo B", consists of two units, B1 and B2, with a capacity of 339 MW each. The units, like many industrial technological industries in the world are composed of different machines such as turbines, mills, feed pumps, condense pumps (Figure 1) that require continued maintenance. Through the **Asset Management** class of the Master's of Professional Studies Program at the American University in Kosovo we learned that one way of maintaining a machine's life is by predictive maintenance that involves the trending (to compare parameter reading against previous readings for the same measurement point) and analyzing the machinery performance parameters to detect and identify developing problems before failure and extensive damage can occur.

Regarding predictive maintenance, machine vibration is one of the main indicators that detects when a machine is developing problems or something isn't right with the machine's normal operation in comparison with other indicators such as oil analysis, infrared thermograph. Machine vibration is simply the back and forth movement of machines or the machine's components.

By understanding how important predictive maintenance is for one machine or for a Power Station like TPP "Kosovo B" I decided to conduct an analysis of the vibration problems in one of its vital machine in unit B1, where the monitoring vibration instruments indicated problems.

By allowing this machine to run until failure, the repair costs will escalate dramatically. In general, repairing a machine will involve more parts, longer shutdown periods, and more labor to complete. Fortunately, modern vibration analysis instruments and software predict developing problems so that repairs can happen before disaster strikes.

I hope that the benefits of analysis and diagnoses of vibrations in this machine will result in recommendations for corrective action that will include a work schedule, requirements for personnel, tools and replacement parts to be made available before the shutdown of the machine to thus avoid extensive damage.

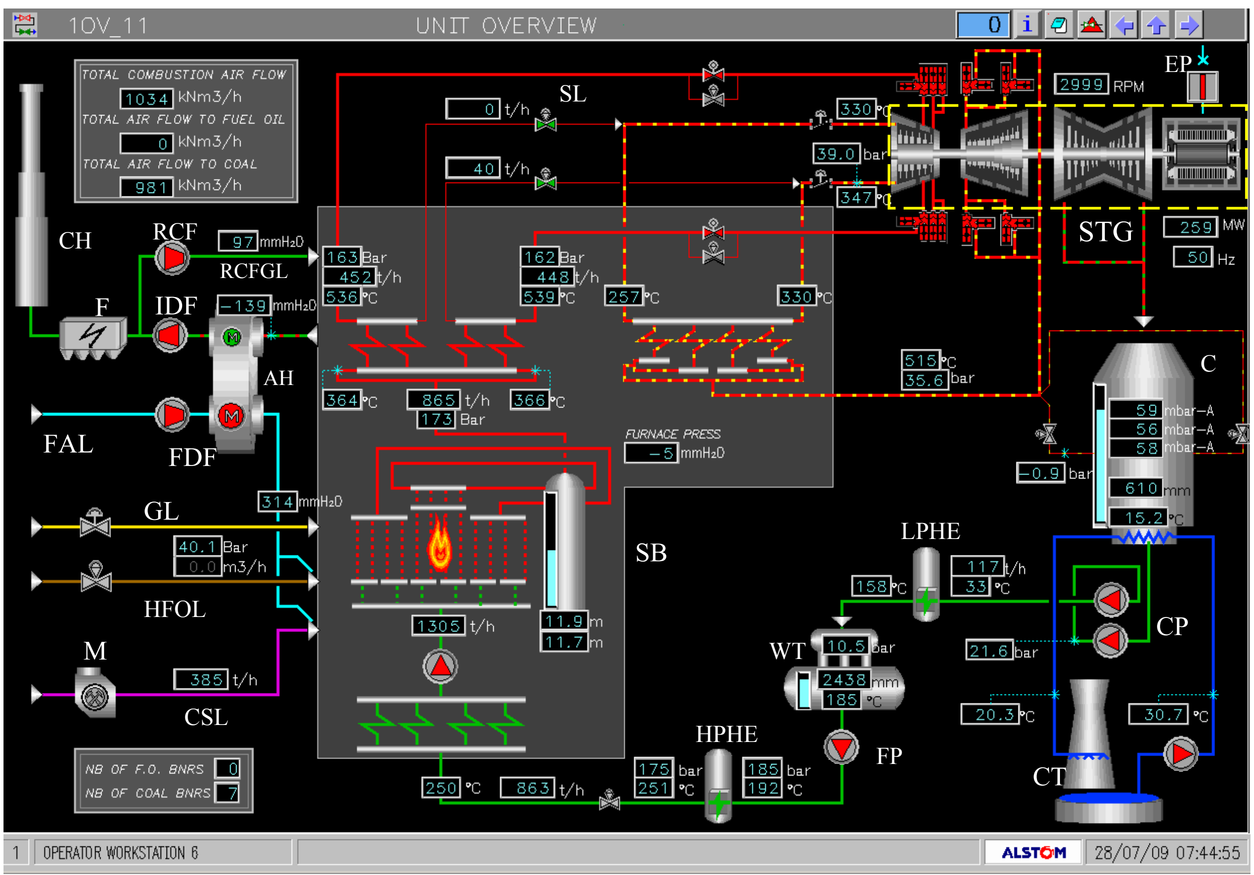


Figure 1: The scheme of TPP “Kosovo B” Unit B1 with operational parameters for capacity 259 MW

(Taken from the TPP “Kosovo B” control room)

2. PROBLEM BACKGROUND

The Steam Turbo-Generator machine STG (**Figure 1**) was installed in 1983 as part of the unit B1 339 MW in TPP "Kosovo B" and is one of the most important machines for generating electricity. This machine is the heart of unit B1, therefore, to monitor its condition during start up, normal operation and shut downs, a new sophisticated **on-line** monitoring vibration system called VIBROMETER **VM600** was installed 2003. This system permanently monitors the relative shaft vibration up to the 10^{th} (tenth) journal bearing of the STG-B1 machine. Each bearing is fitted with 2 relative shaft vibration sensors (one in the vertical direction Y and the other in horizontal X (Figure 2)).

Besides the on-line monitoring shaft vibration system in the STG-B1 machine, TPP "Kosovo B" is also organized to receive monthly preventive **off–line** vibration measurements on the bearing pedestals in three directions -horizontal, vertical and axial – (see Figure 10), in order to increase control of the machine's condition because unfortunately, some problems show up only in one direction.

Monitoring the vibration characteristics of a machine gives us an understanding of the health condition of the machine and we use this information to detect problems that might be developing. To detect any machinery problems we use vibration signals as indicators of a machine's condition because each mechanical component or defect generates vibrations.

The problem my Capstone Project will attempt to answer is what are the preventive results stemming from the vibration measurements on the bearing pedestals (absolute vibration) of the STG-B1 machine in TPP "Kosovo B" compared to ISO standard 10816-2 (a standard for Steam Turbo-Generator machine vibrations by measuring non- rotating parts with a rated power higher than 50 MW) that show higher vibration values (see **Table 1** and **Figure 3**).

The level of vibration is a useful guide to the machine's condition and an increase in the vibration is a direct result of failing elements in the machine.

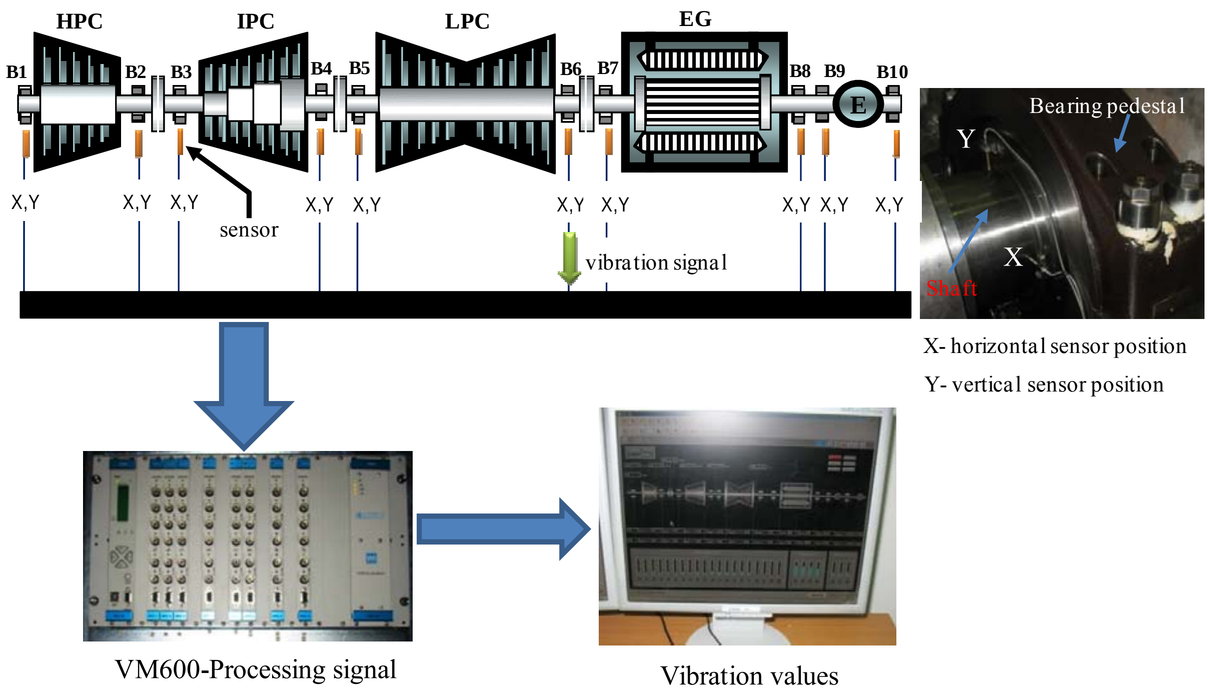
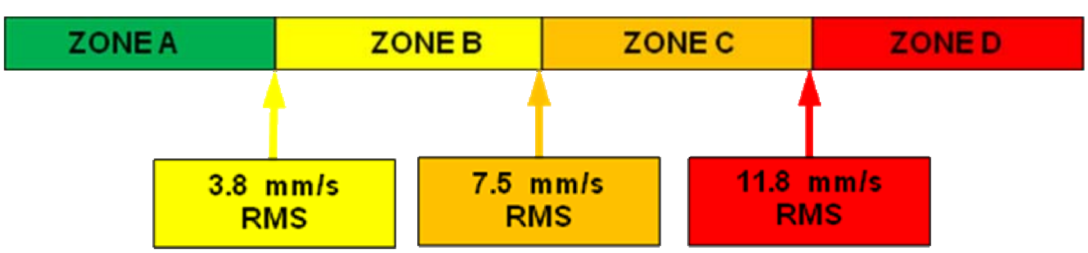


Figure 2: The on-line monitoring vibration system VM600 in the STG-B1 machine

The ISO standard 10816-2 gives the following limits:



The evaluation vibration zones are shown as follows:

<u>Zone A</u>: The vibration of a newly commissioned machine will normally fall within this zone.

<u>Zone B</u>: A machine with vibrations within this zone is normally considered acceptable for unrestricted long-term operation.

<u>Zone C</u>: A machine with vibrations within this zone is normally considered unsatisfactory for long-term continuous operation. Generally, the machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.

<u>Zone D</u>: Vibration values within this zone are normally considered to be of sufficient severity to cause damage to the machine.

If we analyze the results of the absolute vibration measurements on the STG-B1 machine taken on 11 March 2009 by TPP ”Kosovo B” personnel, as shown in Table 1, the critical value of vibrations is on bearing B6 in the axial direction (B6/A), and B7 and B8 in the vertical direction (B7/V, B8/V). The amplitude of vibrations recorded on the off-line instrument was in mm/s - RMS unit (see Figure 12).

Table 1: The results of the absolute vibration measurements on the STG-B1 machine in unit B1 339 MW in TPP ”Kosovo B“ taken on 11 March 2009

TPP ”Kosovo B” STG-B1 machine		Bearing							
		B1	**B2**	**B3**	**B4**	**B5**	**B6**	**B7**	**B8**
		v(mm/s) RMS	v(mm/s) RMS	v(mm/s) RMS	v(mm/s) RMS	v(mm/s) RMS	v(mm/s) RMS	v(mm/s) RMS	v(mm/s) RMS
Direction	Vertical	**1.30**	**2.35**	**0.72**	**2.27**	**2.79**	**2.57**	**7.65**	**9.33**
	Horizontal	**0.66**	**0.92**	**0.93**	**0.90**	**1.33**	**2.25**	**4.94**	**3.67**
	Axial	**1.18**	**0.75**	**0.81**	**1.18**	**1.12**	**13.45**	**3.60**	**3.80**

The results in Table 1 not only shown high vibration values but also the previous vibration measurement results that indicated the machine had vibration problems. As is shown in the following diagram, Figure 3, from after the war in Kosovo in 1999 until now the absolute vibration (vibration measurements on the bearing pedestal) on the three measurement points B6/A, B7/V and B8/V were high. Sometimes the vibrations were very high and sometimes much better, but at all the times they were bad.

By analyzing the above results we can ask:

1. What is wrong with the machine?

2. What is the machine’s vibration problem?
3. Why is this machine running at considerable risk?
4. What should be the corrective actions?

These are questions that require answers.

The effects of high vibrations in the STG-B1 machine are not only degrading the machine’s operation by increasing the risk for normal operation and shortening the machine’s life but also for operational and maintenance personnel.

The high level of vibration on the bearing pedestals of the STB-B1 machine is an indicator that something isn’t right with this machine and its normal operation. Therefore, one or more problems (such as an unbalance rotor, misalignment of the coupling, mechanical looseness, bent shaft, electrical problems etc.) exist that are causing these high vibrations that require solutions. Based on the results of the absolute vibration measurements and the ISO standard 10816-2, the STG-B1 machine may be operated for a limited period because high vibration indicates problems for the machine and may also cause other problems. Therefore, a diagnosis of the vibration problems together with planning maintenance and corrective action is necessary.

As shown in Figure 1, the STG-B1 machine is directly connected to the production of electricity and unplanned shut downs can have catastrophic consequences for the unit. We should also take into consideration that if the STG-B1 machine stops running for any problem, there will be a loss of approximately 260 MW electrical power per hour. On the other hand, depending on the problems that may be present, this machine may require expensive repairs over a long period of time.

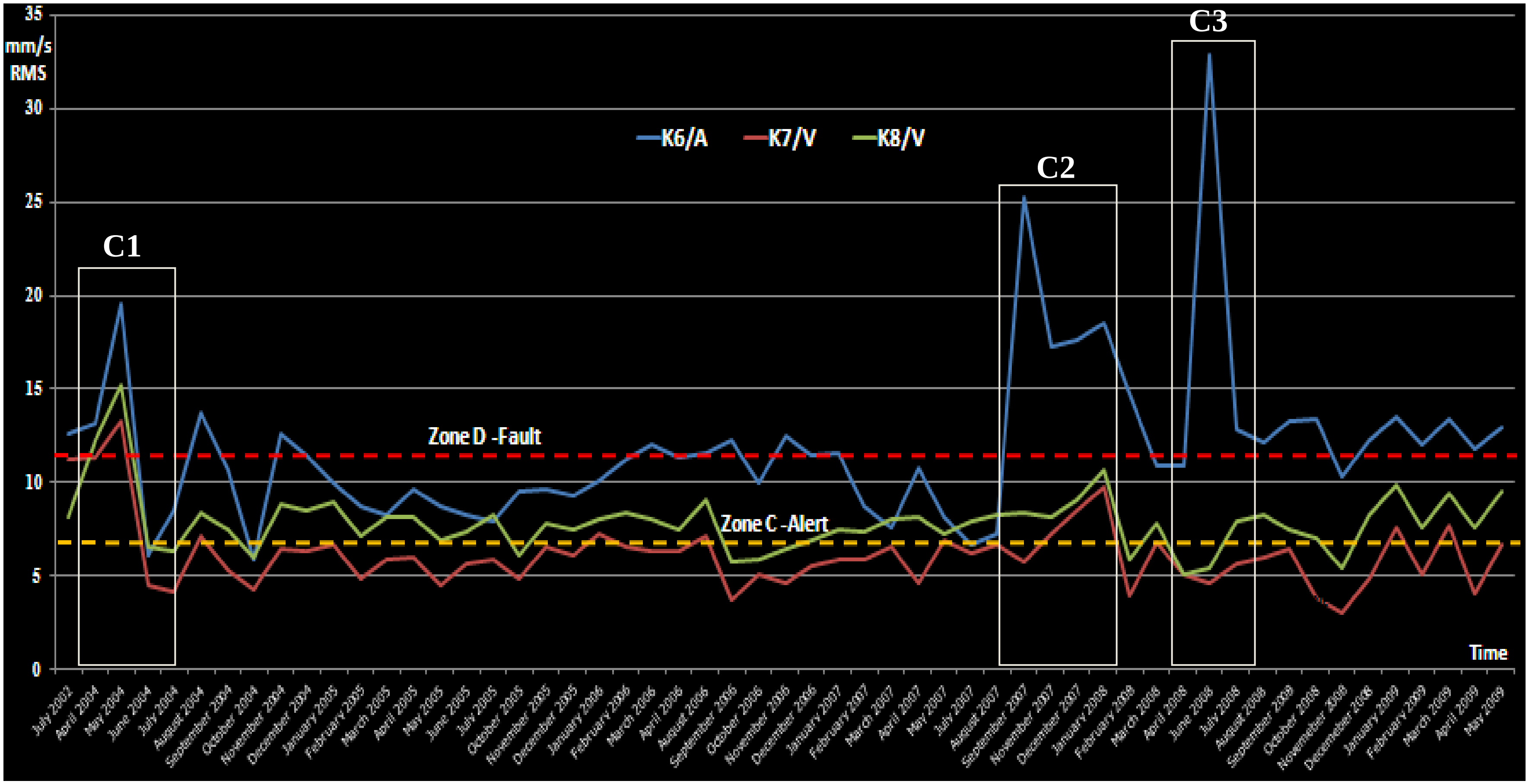


Figure 3: The trend of vibration measurement points B6/A, B7/V and B8/V for 5 years of the machine operation

C1- the event when a balancing generator rotor was done (vibrations are decreased)

C2 and C3 – the events when the blades of Low Pressure Rotor (LPR) in stage 27A and 29A were cracked (see Fig. 8)

3. APPROACH TO SOLVING THE PROBLEM

For my Capstone Project, I plan to conduct an analysis and diagnosis of the vibration problems in the STG-B1 machine in unit B1 339 MW in TPP "Kosovo B". To do that, it will be necessary that I co-operate with the maintenance and operational personnel in TPP "Kosovo B" to collect the operational and technical data of the machine, and conduct all vibration measurements in the STG-B1 machine.

To develop a plan on how to conduct this project, the knowledge obtained from the Introduction to Project Management, Advance Project Management and the Asset Management classes of the Master's of Professional Studies Program at the American University in Kosovo were useful.

I used acquired knowledge such as project planning, scheduling, budgeting and cost estimation from the Project Management class, and preventive and predictive maintenance from the Asset Management class for conducting my Capstone Project.

The following will described how I went about solving the problem, and what research methods and resources I used to conduct the analysis of the vibration problems in the STG-B1 machine.

In order to analyze vibrations in this machine and to make the best recommendations for corrective action I planed to conduct the following three steps:

- Analysis of the machine's construction design and technical data
- Analysis of the machine's history
- Conduct and collect vibration data and operational parameters on the machine

3.1 **Analysis of the Machine's Construction Design and Technical Data**

The first step of my project research was to analyze the machine's construction design and technical data. That helped me to be informed how the machine works, what were the machine's elements, what was the machine's running speed, what vibration data to collect and store, what vibration measurement units to take (displacement, velocity or acceleration) and where and how to take them. For example, where should I place the transducer for measuring the absolute vibrations?

The information collected about the machine's construction design was a useful tool to identify all the components of the machine that could cause vibration. That was necessary before an analysis of the vibration problems could be done because the components that could cause vibrations within the machine must be known.

The STG-B1 machine (Figure 4) is composed of:

- Steam Turbine
- Electrical Generator

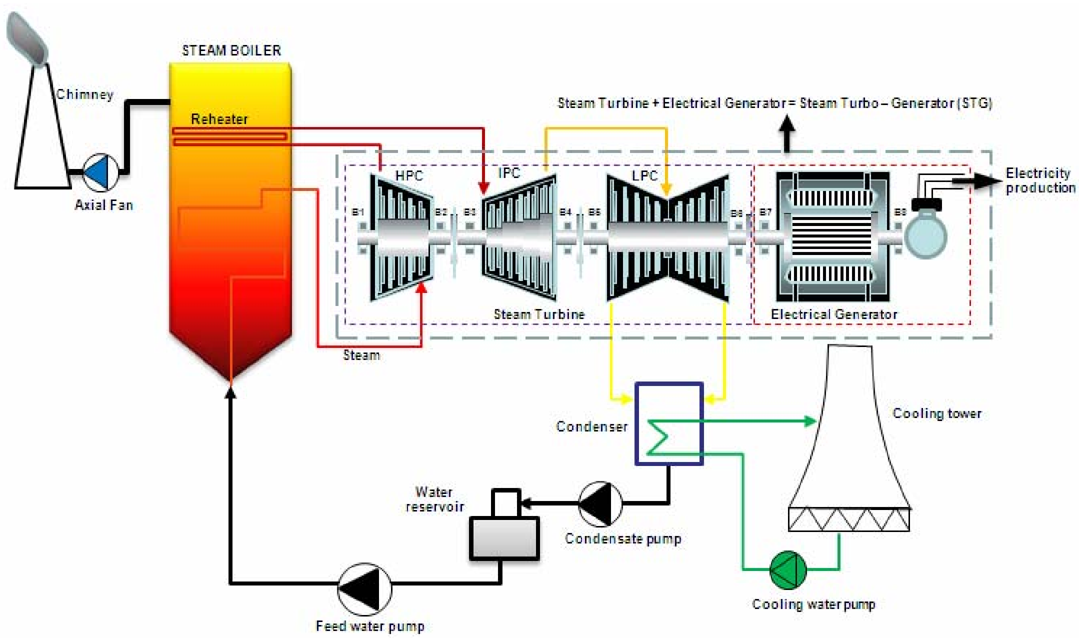


Figure 4: The Unit B1

3.1.1 **Steam Turbine**

The steam turbine is a mechanical device that extracts thermal energy from pressurized steam, and converts it into rotary motion or mechanical energy. The rotary motion from the steam turbine through shafts and rigid couplings is extracts to the rotary motion of the generator rotor, and that results in electrical power (Figure 4). The steam turbine of the STG-B1 machine is a low pressure cylinder (LPC) section rigidly coupled to an intermediate pressure cylinder (IPC) and high pressure cylinder (HPC) as shown in Figure 5. Each cylinder consists of two casings, up and down. All the rotors are supported on the elliptical journal bearings with a center for oil lubrication (see appendix B). Only B2 is a radial-axial journal bearing with all the others being radial.

The main components of the turbine are: a high pressure rotor (HPR), intermediate pressure rotor (IPR), low pressure rotor (LPR), journal bearings (B), casings, bolts, rigid couplings, labyrinth casket (see figure 5).

The turbine characteristics are as follows:

- The total turbine weight 641 tons
- The LPC cylinder 49 tons
- The LPC stage number 2x5 (5 left and 5 right)
- The IPC stage number 12
- The HPC stage number 12

As shown in **Figure 1**, the technological steam from steam boiler (SB) with parameters: pressure 163 bar, temperature 539 ºC and quantity 900 tons per hour (these parameters are for 259 MW of the unit capacity) through stop and control valves comes into HPC by extracting a part of its thermal energy. After the expansion process, the steam with parameters of 39 bar, 330 ºC, goes from HPC to SB for reheat, and in IPC and LPC to extract more of its thermal energy.

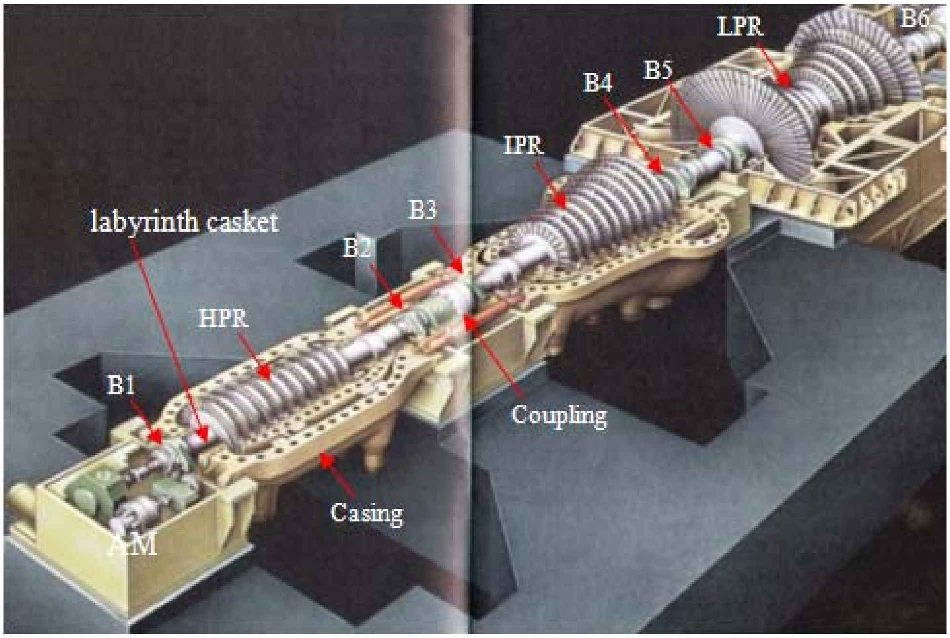


Figure 5: Steam turbine

3.1.2 **Electrical Generator**

The electrical generator (Figure 6) is a device the converts the rotary motion by the steam turbine into electrical energy, generally using electromagnetic induction. The electrical generator is rigidly coupled to the steam turbine and is supported on the two journal bearings with a center for oil lubrication. The stator and rotor windings are cooled with hydrogen and with water through a water exchanger device.

The main components of the electrical generator are: a generator rotor, stator, journal bearings, stator windings, rotor winding bars, retaining rings (see Figure 6). It rotates at 3000 rpm.

The constructive characteristics are: generator weight, 320 tons; generator rotor weight, 55 tons; and generator stator weight, 265 tons. The manufacture is ALSTHOM - ATLANTIQUE, Belfort, France.

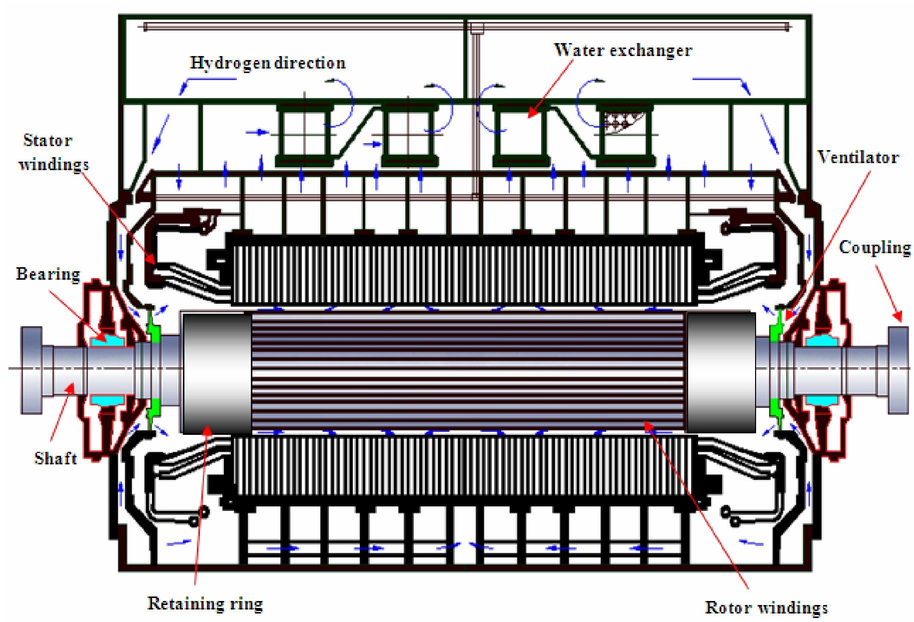


Figure 6: Electrical Generator

3.2 Analysis of the Machine's History

The second step of my research project was collecting and analyzing the information on the machine's history. I conducted this step because it was important for me to be informed regarding the machine's previous condition and to help me to diagnose the machine's vibration problems. Based on the documentation recorded and contacts with responsible engineers and chiefs in TPP "Kosovo B" I understood that the history of the STG-B1 machine would give me no satisfaction. The following will be three interesting events that happened after the war in Kosovo in 1999, and they helped me to gain batter results in diagnosing the vibration problems.

3.2.1 Generator Rotor Repair (2001)

My contacts with the responsible maintenance engineer for the STB-B1 machine in TPP "Kosovo B" , Mr. Ali Mehanja, and researching the documentation in the TPP "Kosovo B" library for the STG-B1 machine, I found **a generator rotor repair report** of the STG-B1 machine in which was written: "Based on the attention of the station personnel that the rotor of the Electrical Generator B1 is running with high relative shaft vibration, one inspection was done from Consortium Lurgi Lentjes Service and Babcock Borsig Power".

In this inspection report done by Consortium No.2 and dated 23/07/ 2001, it was reported that some of the generator rotor winding bars were distorted and elongated and were causing thermal instability. The repair work on the generator rotor winding bars was started on 7 September 2001. Some of winding generator rotor bars were shortened and repaired (see Figure 7b). After that all rotor winding bars were checked for dimensional correctness and realigned.

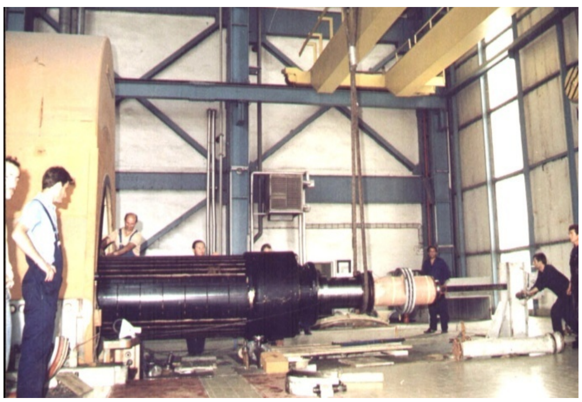


Fig. 7a: The generator rotor during disassembly
. (Photo taken from Ali Mehanja)

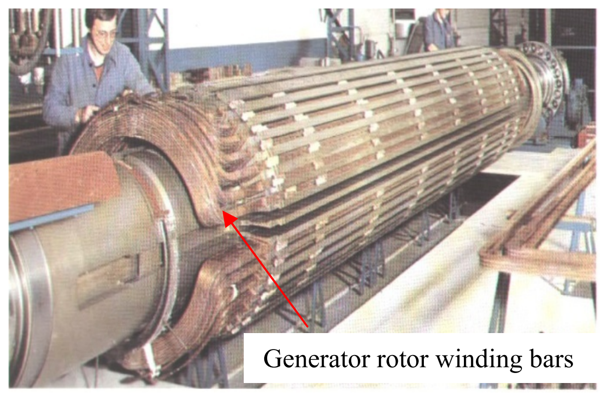


Fig. 7b: An example of the generator rotor winding bars repair (taken from ALSTHOM ATLANTIQUE catalog)

3.2.2 Balancing Generator Rotor (2004)

Based on the results of the absolute vibration measurements on the bearing pedestal of the electrical generator in the STG-B1 machine taken in the month of April 2004 as shown in Table 2,-prior to balancing, the TPP "Kosovo B" management had taken corrective action to bring the permissible level of vibration in accordance with ISO standard 10816-2. After taking into consideration all measurements vs. different working condition, an expert from PRUFTECHNIK Company (Poland) concluded that the main reason for high vibration levels on the bearing pedestals of the electrical generator was the dynamic rotor unbalance and to make better the vibrations, a balancing of generator rotor in nominal speed (3000 rpm) should be carried out.

Table 2. The results of vibration measurments prior and after generator rotor balancing in 2004

TPP "Kosovo B" STG-B1 machine Balancing Generator Rotor 2004		Bearing					
		B6	B7	B8	B6	B7	B8
		Prior balancing			**After balancing**		
		v(mm/s) RMS	v(mm/s) RMS	v(mm/s) RMS	v(mm/s) RMS	v(mm/s) RMS	v(mm/s) RMS
Direction	Vertical	**2,66**	**13,11**	**15,12**	**2,65**	**4,21**	**5,92**
	Horizontal	**5,57**	**13,08**	**7,46**	**2,88**	**3,58**	**3,03**
	Axial	**19,51**	**5.98**	**5,03**	**6,98**	**2,32**	**1,50**

Unfortunately, the vibration results after the generator rotor balancing changed and after two months of the STG-B1 machine operation, the vibrations were again high (Figure 3- case C1).

3.2.3 Shortened the LP Turbine Blades

The other interesting events found from the machine history were the periods of time in 2008, where during the normal operation of the STG-B1 machine a blade of the low pressure turbine was cracked causing damages to other blades (Figure 8). This had consequences that caused unit B1 to be shut down for several days. To return the machine to operation again, a corrective action was taken by shortening all blades to 30 cm length in stages no.28A, 28B, 29A, 29B and all blades in the rotor (to root) in stages no. 27A and 27B as shown in Figure 9. This corrective action in the LP rotor with no balancing had consequences by increasing the very high vibrations in the STG-B1 machine during start up (Figure 3, case C2 and C3) and restricted the continuous work of the machine. Balancing the LP rotor in-situ was necessity. This corrective action (LP rotor balancing) had effects not only in decreasing the relative shaft vibration but also in decreasing the absolute vibration in the bearing no.B6 in the axial direction. The decreased level of absolute vibration measurements in the bearing no.B6/A was not satisfactory (Figure 3) and the turbine continued to work with limited capacity due to the LP's shortened rotor blades (the capacity was limited to 260 MW).

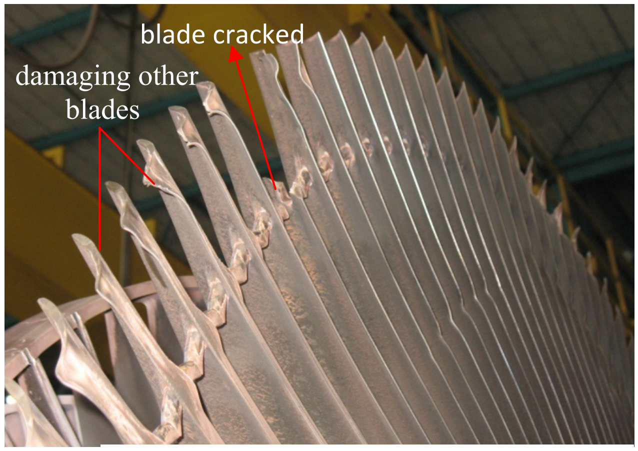


Figure 8: The LP rotor blade cracked

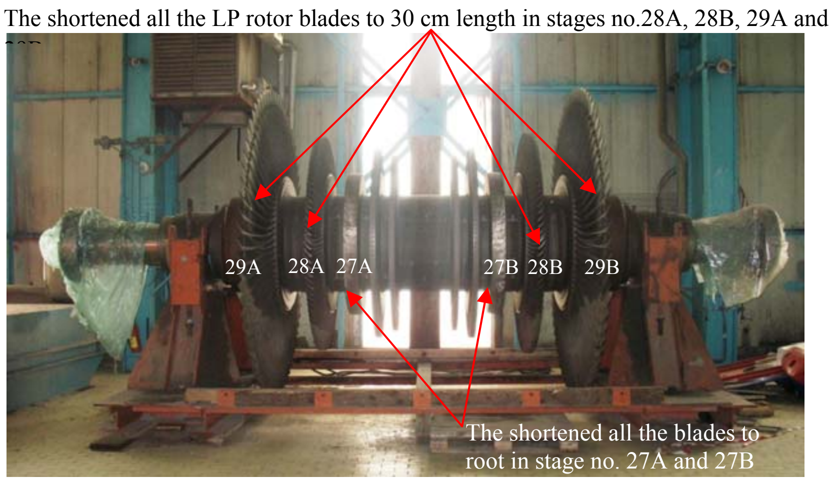


Fig. 9: The LP rotor during its repair

3.3 Conduct and Collect Vibration Data and Operational Parameters on the Machine

The third and most important step of my project research was measuring and collecting vibration data on the STG-B1 machine. This implies obtaining the overall absolute vibration measurements and full-spectrum vibration signature in the vertical, horizontal and axial directions (see Figure 10) on the bearing pedestals number B6, B7 and B8. The vibration analyses and readings taken in these three positions provided insight as to what may be causing any excessive vibration.

Before conducting the vibration measurements on the STG-B1 machine I was aware that the key consideration in obtaining a vibration signal that accurately represents the vibration was selecting the right type of transducer, measurement unit, and locating and installing it correctly to get the right data. Table 3 that follows will show the selecting measurement units and definition depending on the machine's rotating speeds (Source: On-line machinery condition monitoring diagnostics, Author: Peter W. Hills, White paper Dec. 2005).

In our case the measurement unit should be velocity with the scale factor RMS or zero-peak.

Table 3: The selecting measurement units

Measurement Units	Definition	Units	Machine Speed Range in RPM
Displacement	The total distance a mass travels back and forth as it vibrates	Microns peak-peak	<600
Velocity	A measure of how fast the mass is moving back and forth	mm/s Peak or RMS	600 -120 000
Acceleration	The rate of change of velocity	mm/s^2	120 000

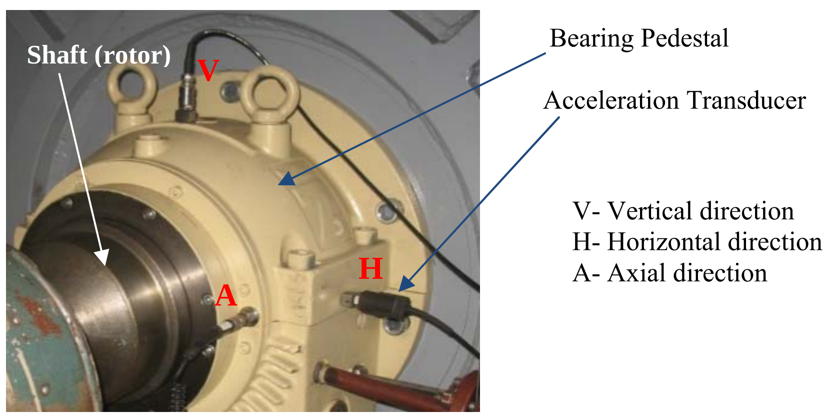


Fig. 10: An example of measuring the absolute vibration in three directions

3.3.1 **Instruments Used**

In order to conduct the measuring and vibration collection data on this type of machine I utilized sophisticated instruments such as:

1. **<u>VIBSCANNER</u> –** an off-line instrument with two channels for measuring and collecting the absolute vibration data on the bearing pedestals (Figure 11a). This type of transducer is an accelerometer with a frequency range of 2Hz - 20 kHz.

Fig. 11a: VIBSCANNER instrument

Fig. 11b: Collecting vibration data with VIBSCANNER

Before I started collecting vibration data on the machine, it was very important to know what vibration data to collect and store. Therefore, to analyze vibration problems in the STG-B1 machine I choose to collect:

<u>**Overall vibration**</u> - the total vibrations measured within a frequency range of 2-1000 Hz with scale factor in RMS. Other scale factors used in overall vibration measurements that indicate the vibration severity are: **peak** or **zero-peak, peak to peak** and **average** (Figure 12)**.**

The **Zero-Peak (O-P)** value represents the distance to the top of the waveform measured from a zero reference.

The **Peak to Peak (P-P)** value is the amplitude measured from the top of the waveform to the bottom.

The **Average** value is the average amplitude measured for the waveform.

The **RMS** (**Root Mean Square)** amplitude of a vibrating machine tells the vibration energy in the machine. The RMS amplitude is always lower than the peak amplitude. When the waveform spectrum is sinusoidal the RMS= (0.707) x (0-P)

It is really a matter of personal choice if we select the RMS or Zero-Peak as scale factors of vibration measurement units because if vibrations are measured with the scale factor in RMS, it is very easy to calculate this value in Zero-Peak or Peak to Peak (software).

Waveform spectrum – a waveform spectrum is a graphical representation of how the vibration level changes with time (the plot of vibration vs. time). While the graphical display of electrical signals from a person's heart (electrocardiogram or ECG) is useful for analyzing the medical condition of the person's heart, the waveform spectrum is a useful tool for analyzing the nature of the vibration severity.

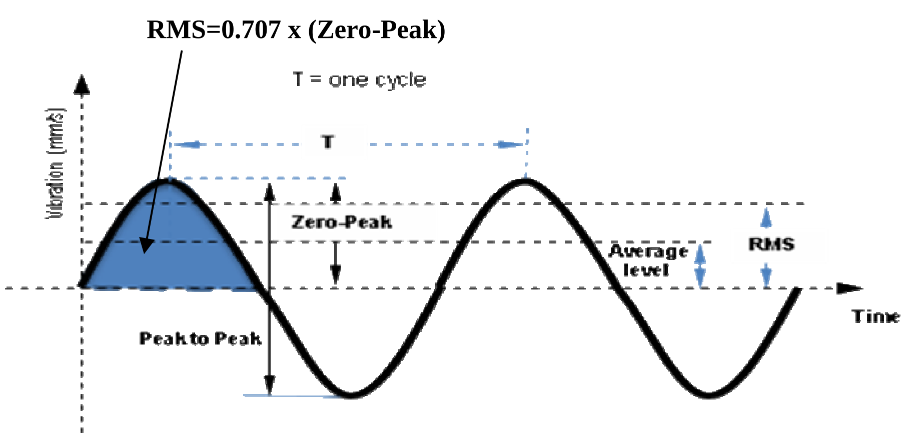


Figure 12: When a waveform spectrum is sinusoidal

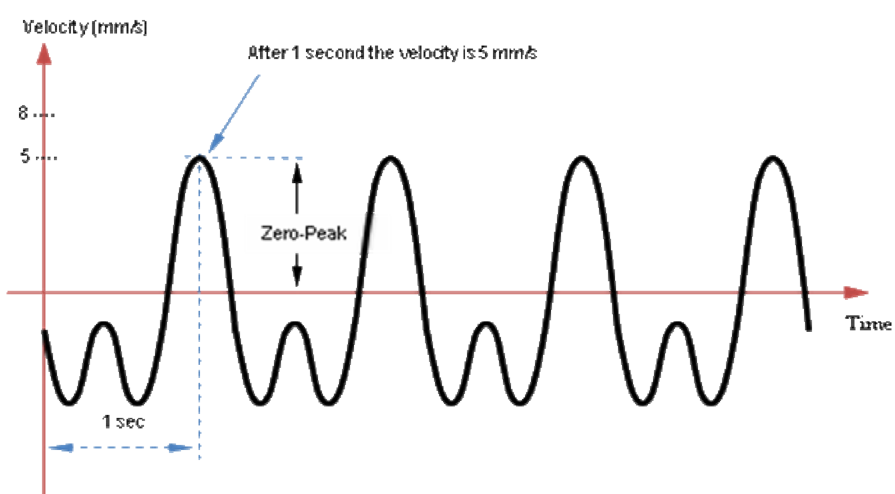


Figure 13: When a waveform spectrum isn't sinusoidal

<u>FFT Spectrum</u> <u>(Fast Fourier Transformation Spectrum) –</u> is a calculation method for converting a time waveform to a frequency display that shows the relationship of discrete frequencies and their amplitudes (Fig. 14, the plot of vibration vs. frequency). This spectrum is a useful tool for analyzing vibrations and provides information to help determine the location of the problem, type of problem and the cause of the problem. This spectrum allows us to analyze vibration amplitudes in various component frequencies. In this way we can identify the amplitude vibration in the specific frequency because machinery problems generate vibrations at specific frequencies.

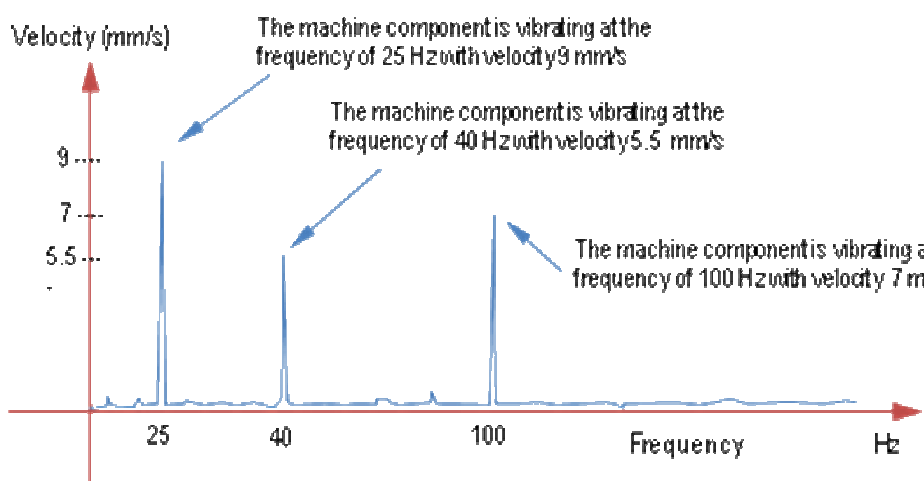


Fig. 14: An example of a FFT spectrum velocity

<u>Phase</u> - the angular difference between a known mark on a rotating shaft of the machine and the vibration signal. This relationship is useful for balancing and analysis purposes.

These plots and phase are required to analyze and determine the machine's vibration problems.

2. **<u>VIBROMETER VM600</u>** – is an on-line monitoring vibration system that was installed in TPP "Kosovo B" in 2003 for monitoring on-line relative shaft vibrations in the STG-B1

machine. This system permanently monitors relative shaft vibrations in the STG-B1 machine by showing amplitude vibration values and insuring the machine's protection if the vibration excesses its limits. This system doesn't have installed analysis vibration software (FFT spectrum) that will help a vibration expert to diagnosis vibration problems in the machine. The on-line vibration data collections are usefully to see what is happening with relative shaft vibrations.

3. **DCS (Distribution Control System) –** was installed in TPP "Kosovo B" for monitoring the temperature of the bearings in the STG-B1 machine, steam pressure and other operational parameters of the unit (see Figures in appendix B).

4. **BRUEL & KJEAR Data collector 2526** (Figure 15a) - I used this instrument to receive signals from the on-line monitoring vibration system, VM600, and processing them in the FFT vibration spectrum because in this instrument a channel is installed for connection to on-line monitoring vibration system (VM600) Figure 15b.

Fig. 15a: Bruel & Kjear 2526 instrument

Fig. 15b: Collecting shaft vibration data with Bruel & Kjear instrument

3.3.2 Collected Vibration Data with VIBSCANNER Instrument

The work of collecting vibration data with the VIBSCANNER instrument began with putting reflective tape on the shaft (Figure 16) before the start run of the machine because it was necessary to read the phase and analyze vibration. The other activity was putting all accessories of the VIBSANNER instrument on the machine Figure 11b.

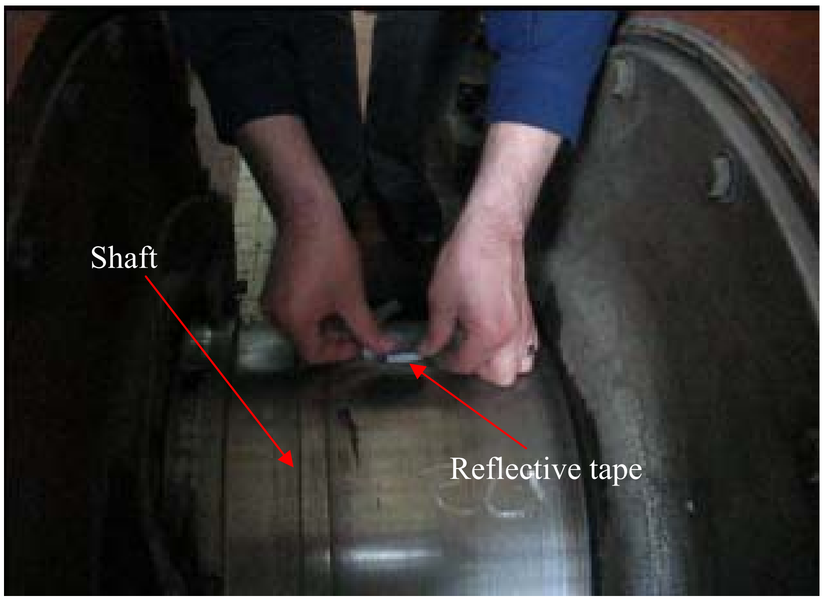


Fig. 16: Reflective tape on the shaft

In order to obtain a good analysis of the vibration problems on the STG-B1 machine, I measured vibrations on the machine during start up, idle running and maximum allowed loading because I needed to see how the vibrations changed under different operational regimes. In this case I was restricted to the full analysis vibration in the STG-B1 machine due to the LP rotor that has a limited unit capacity of from 300 MW to 260 MW. However, I completed all vibration measurements in different loads to 260 MW, and that was useful for me in conducting an analysis of the vibration problems.

The results of the overall vibration measurements taken from the VIBSANNER instrument on the bearing pedestals B6, B7 and B8 of the STG-B1 machine during start up, idle running and maximum allowed loads are shown in Table 4.

Table 4: The results of vibration measurements on the bearing pedestal B6, B7 and B8 for different loads

STG-B1 machine		Measurement unit velocity (mm/s) - RMS									
B	Load / Position	1000 rpm	1500 rpm	3000 rpm	100 MW	152 MW	200 MW	250 MW 89 MVar	250 MW 115 MVar	250 MW 130 MVar	253 MW 70 Mvar
B6	Vertical	0,73	1,38	3,05	2,01	1,74	2,14	2,33	3,57	3,38	2.21
	Horizontal	0,28	0,68	3,96	2,33	2,77	3,82	4,41	5,12	4,56	2.40
	Axial	0,82	0,97	12,7	12,07	13,96	13,66	13,78	14,59	12,27	12.28
B7	Vertical	0,22	0,36	4,34	4,04	5,05	4,73	5,99	5,13	5,55	5.87
	Horizontal	0,31	0,34	3,47	3,07	3,71	3,32	3,57	3,15	3,3	3.53
	Axial	0,22	0,32	1,98	2,26	2,28	2,38	2,37	1,47	3,53	2.48
B8	Vertical	0,24	0,3	5,17	4,99	7,03	6,87	7,13	6,72	8,1	8.46
	Horizontal	0,4	0,47	3,7	3,11	2,25	1,92	2,78	2,86	4,71	4.23
	Axial	0,19	0,24	3,04	2,79	3,74	2,66	3,29	2,99	3,04	3.22

The following will also show the waveform spectrum, FFT spectrum with a frequency range (2-400 Hz), and cascade spectrum taken on the bearing pedestal B6/A, B7/V and B8/V.

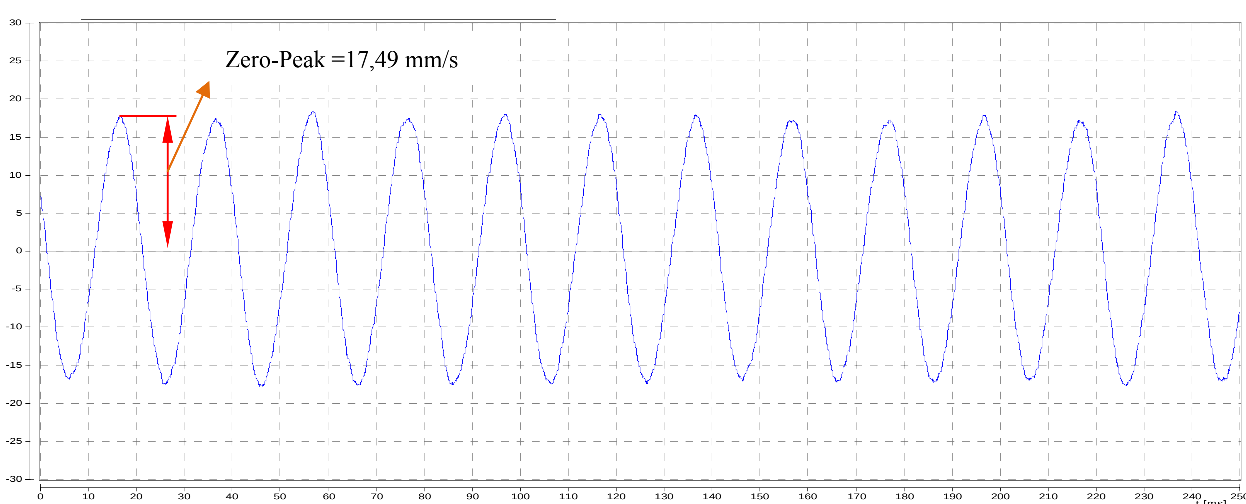


Figure 17a- The waveform spectrum collected from B6/A vibration signal (250 MW)

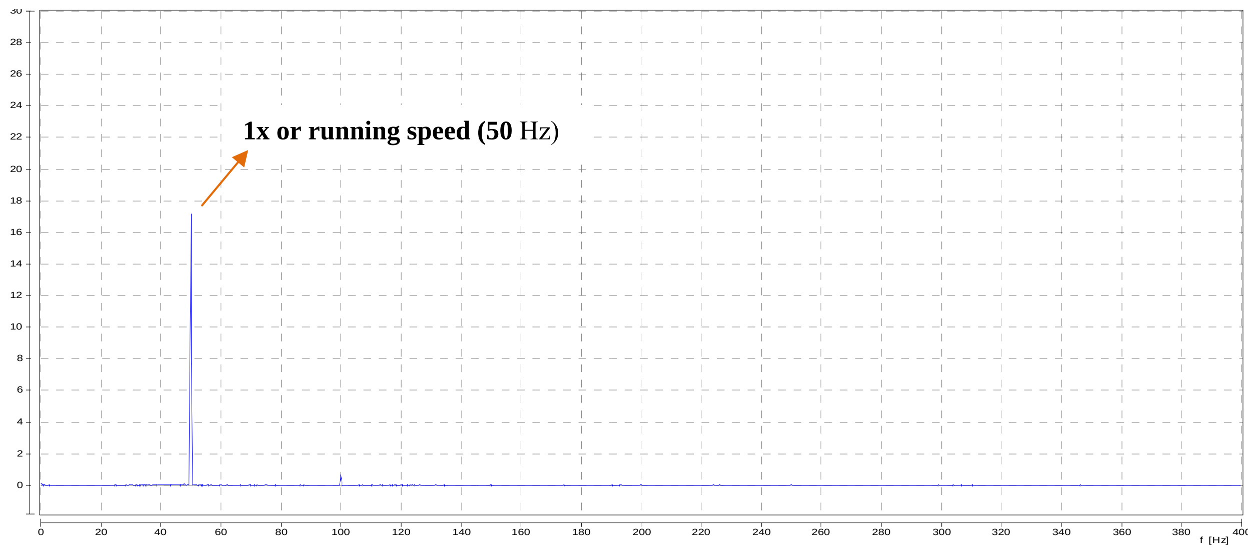


Figure 17b: The FFT spectrum collected from B6/A vibration signal (250 MW)

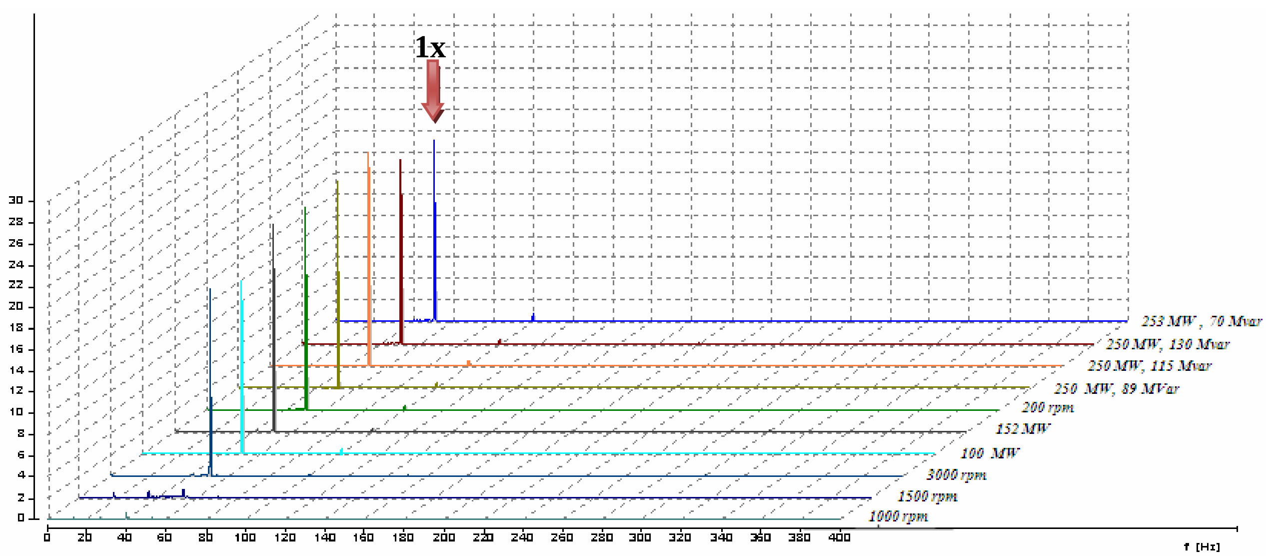


Figure 17c- The cascade FFT spectrum collected from B6/A vibration signal for different loads

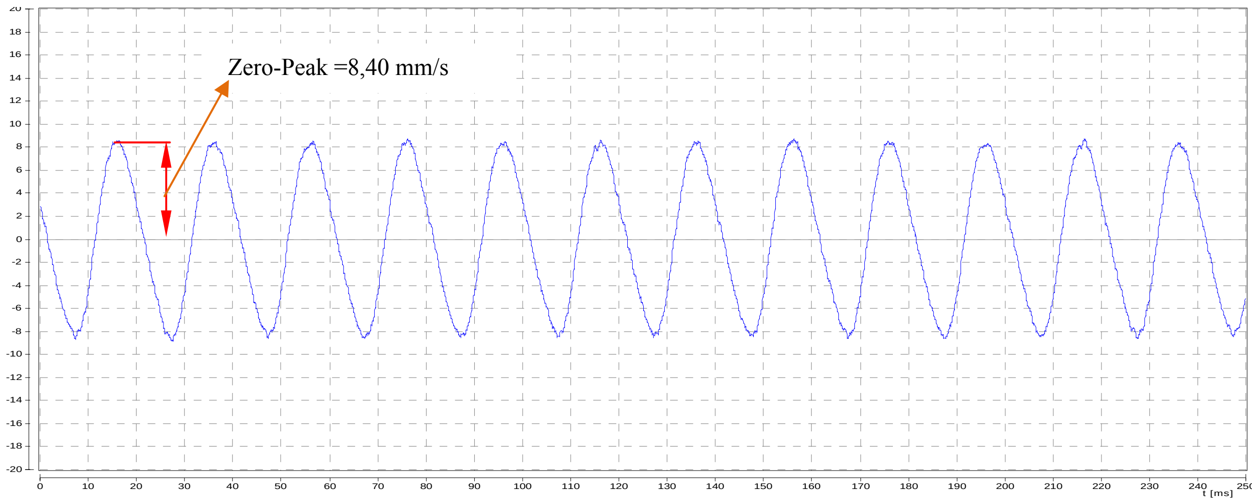


Figure 18a: The waveform spectrum collected from B7/V vibration signal (250 MW)

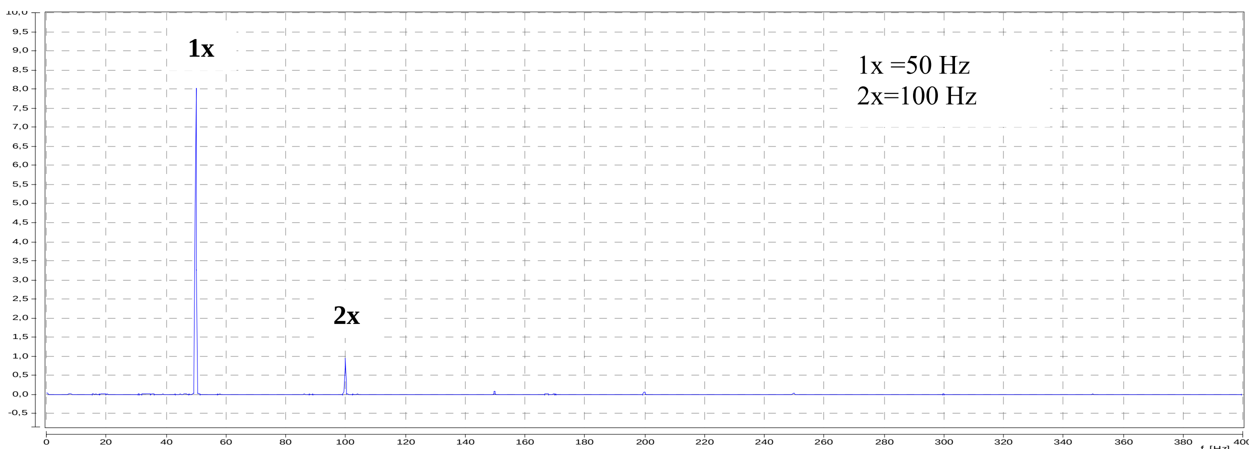


Figure 18b: The FFT spectrum collected from B7/V vibration signal (250 MW)

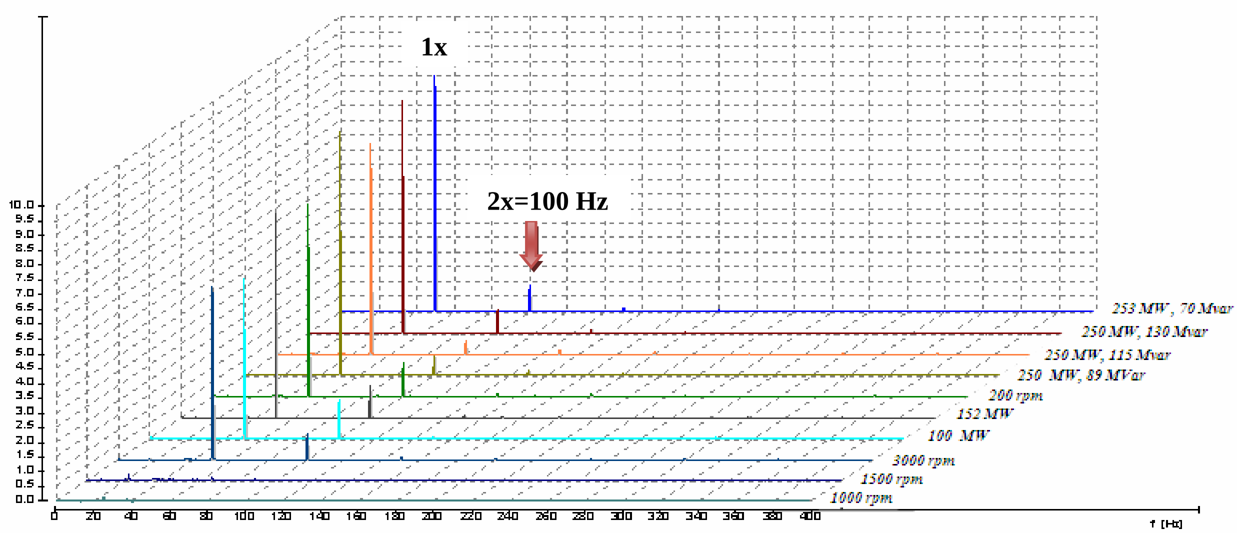


Figure 18c- The cascade FFT spectrum collected from B7/V vibration signal for different loads

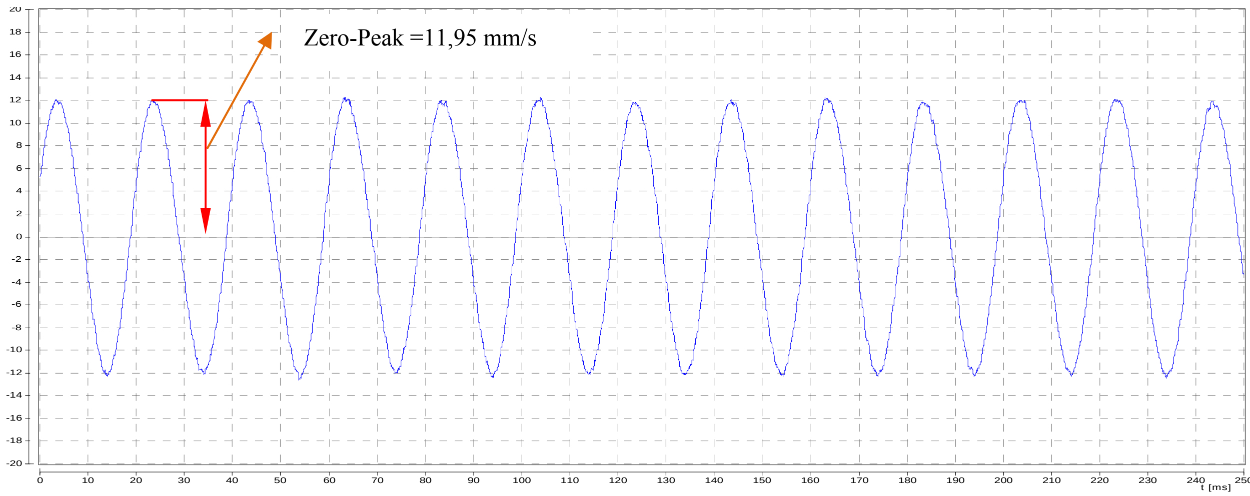


Figure 19a- The waveform spectrum collected from B8/V vibration signal (250 MW)

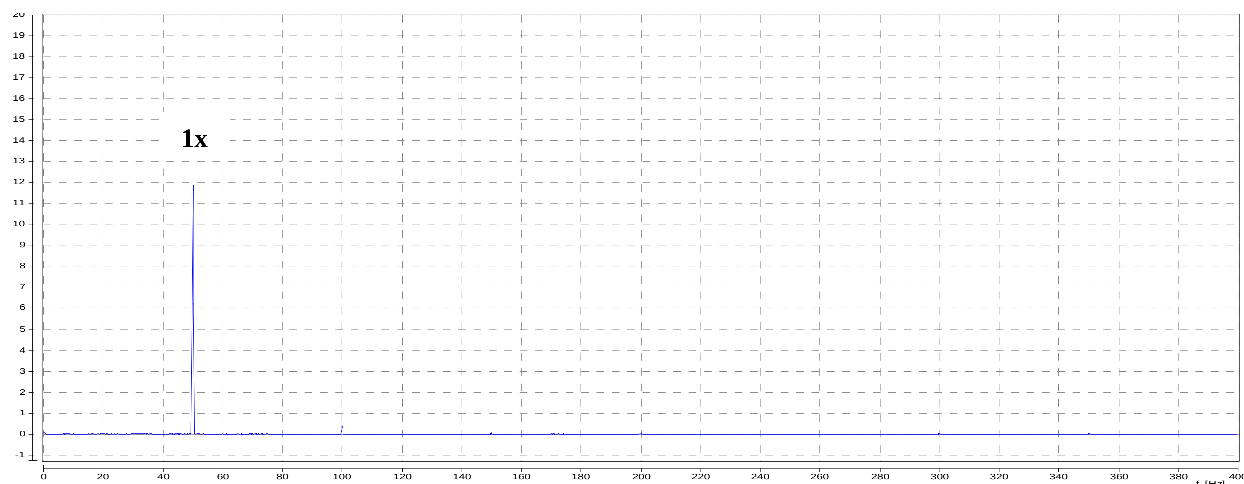


Figure 19b: The FFT spectrum collected from B8/V vibration signal (250 MW)

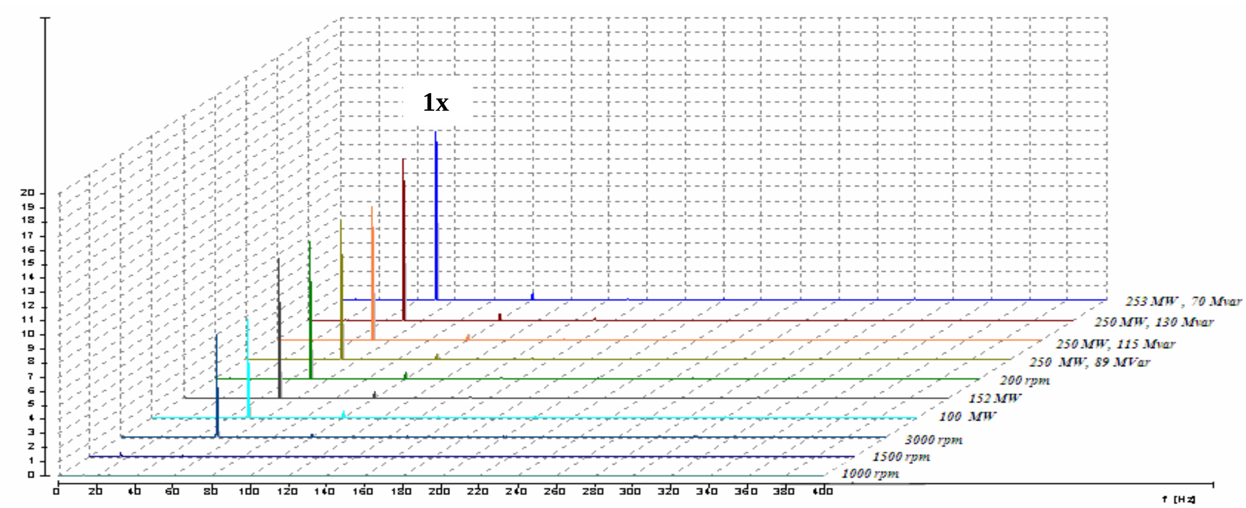


Figure 19c- The cascade FFT spectrum collected from B8/V vibration signal for different loads

During the start up of the machine I also measured vibration and phase of the two measurement points on the bearing pedestal B7/V and B8/V because it was important for me to get additional information on what was happening with the generator rotor (Table 5).

Table 5: The results of vibration measurements and phase taken on the bearing B7/V and B8/V during start runs of the machine.

B7/V			B8/V		
Rotation Speed (RPM)	Velocity (mm/s, 0-P)	Phase (º)	Rotation Speed (RPM)	Velocity (mm/s, 0-P)	Phase (º)
800	0, 05	40	800	0.01	0
1500	0, 32	331	1500	0.02	356
1893	1, 79	23	1937	2, 67	231
2029	1, 91	9	2071	2, 95	209
2174	3, 23	74	2216	2, 69	255
2316	2, 39	88	2360	1, 58	259
2472	1, 65	78	2515	1, 86	245
2687	2, 85	73	2711	2, 26	239
2815	3, 22	73	2858	3, 07	204
2968	5, 55	55	2998	9, 01	231
3000	7, 47	61	3000	9, 21	232
100 MW	5, 65	58	100 MW	7, 89	221
150 MW	6, 89	66	150 MW	9, 27	231
200 MW	6, 40	59	200 MW	9, 28	226
250 MW/89 MVar	9, 82	56	250 MW/89 MVar	12, 54	225
250 MW/115 MVar	7, 50	64	250 MW/115 MVar	9, 26	228
250 MW/130 MVar	8, 44	55	250 MW/130 MVar	10, 96	222

3.3.3 Collected Vibration Data with BRUEL&KJEAR (B&K) Instrument

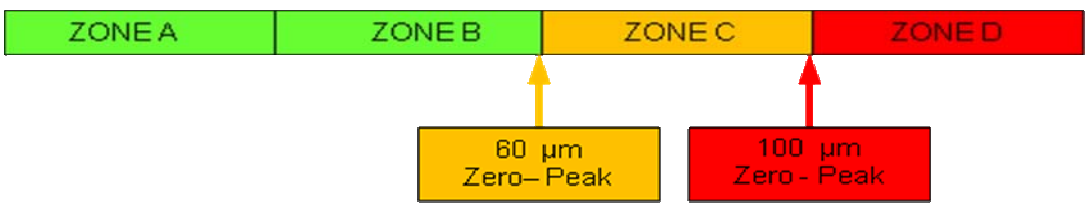
In order to see what is happening with the relative shaft vibrations on the STG-B1 machine in the radial position I also collected vibration data and FFT spectrum for four vibration measurements B7/V, H and B8/V, H. To do that, I needed to be connected to the B&K instrument in the VM600 system (Figure 15b). Table 6 shows the results of the shaft vibration measurements taken with the B&K instrument for 250 MW / 130 MVar power.

The measurement unit of the relative shaft vibration measurements is displacement (µm –micro meter) because the installed transducer type of shaft vibration is a radial displacement.

Table 6: The results of relative shaft vibration measurements in the STG-B1 for capacities 250 MW and 130 MVar

Bearing	**B1**	**B2**	**B3**	**B4**	**B5**	**B6**	**B7**	**B8**
Displace. / Position	0-Peak (µm)	0-Peak (µm)	0-Peak (µm)	0-Peak (µm)	0-Peak (µm)	0-Peak (µm)	0-Peak (µm)	0-Peak (µm)
	B&K connected to on-line monitoring vibration system VM 600							
Vertical	38	21	30	58	44	18	**40**	35
Horizontal	47	49	49	22	48	28	57	65

The VM600 system gives the following limits:



Zones A, B, C and D have the same meaning as was explained for the vibration measurements on the non rotating parts with a rated power higher than 50 MW (page 7 and 8). The following will show the FFT spectrum taken on the vibration signal B8/H, B8/V, B7/H and B7/V (250 MW, 130 Mvar).

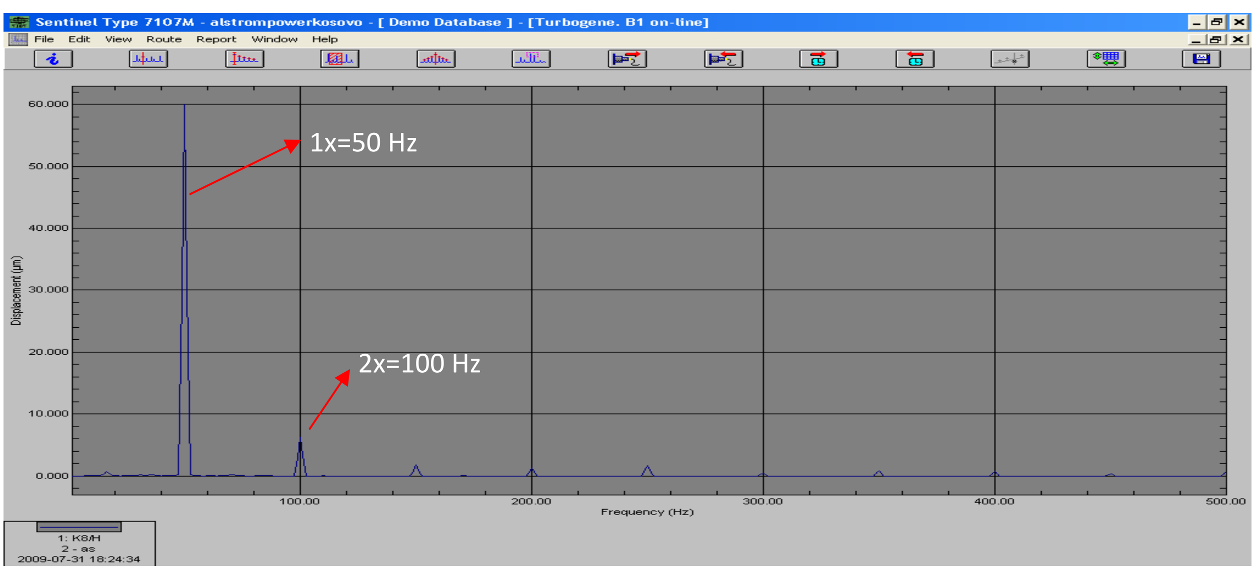


Fig. 20a: The FFT spectrum taken from B8/H vibration signal

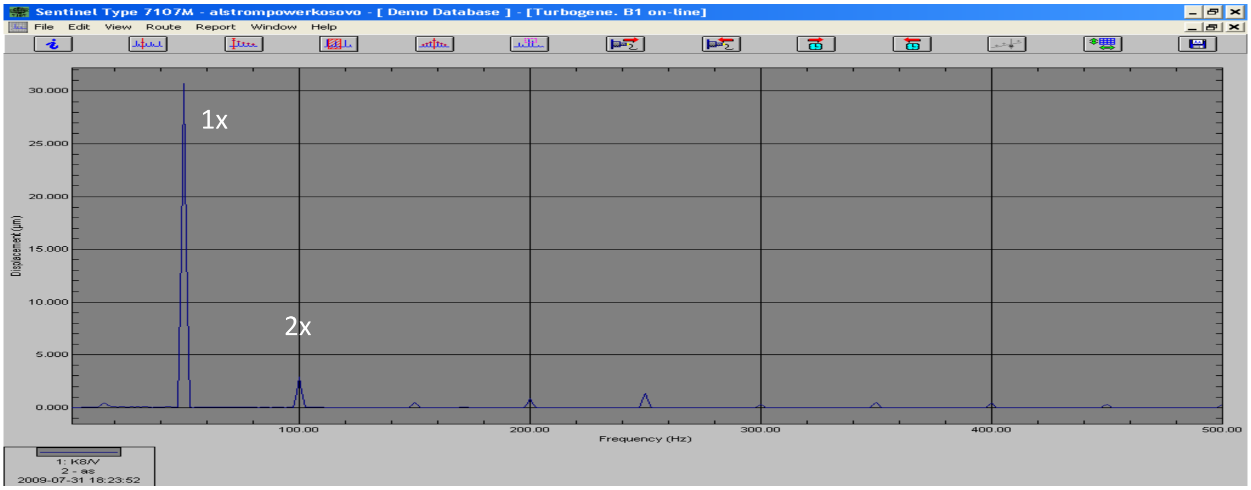


Fig. 20b: The FFT spectrum taken from B8/V vibration signal

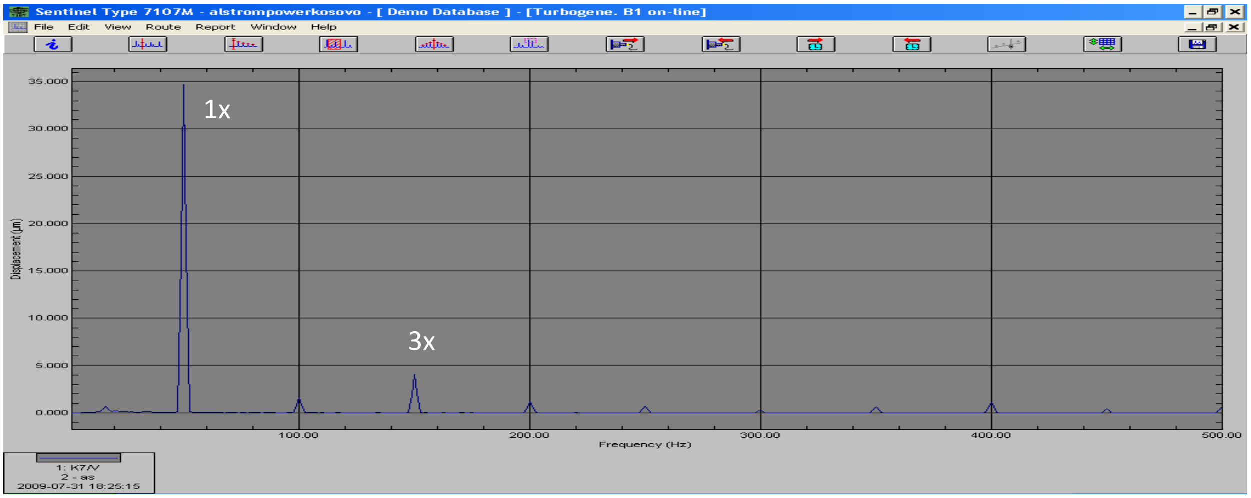


Fig. 21a: The FFT spectrum taken from B7/H vibration signal

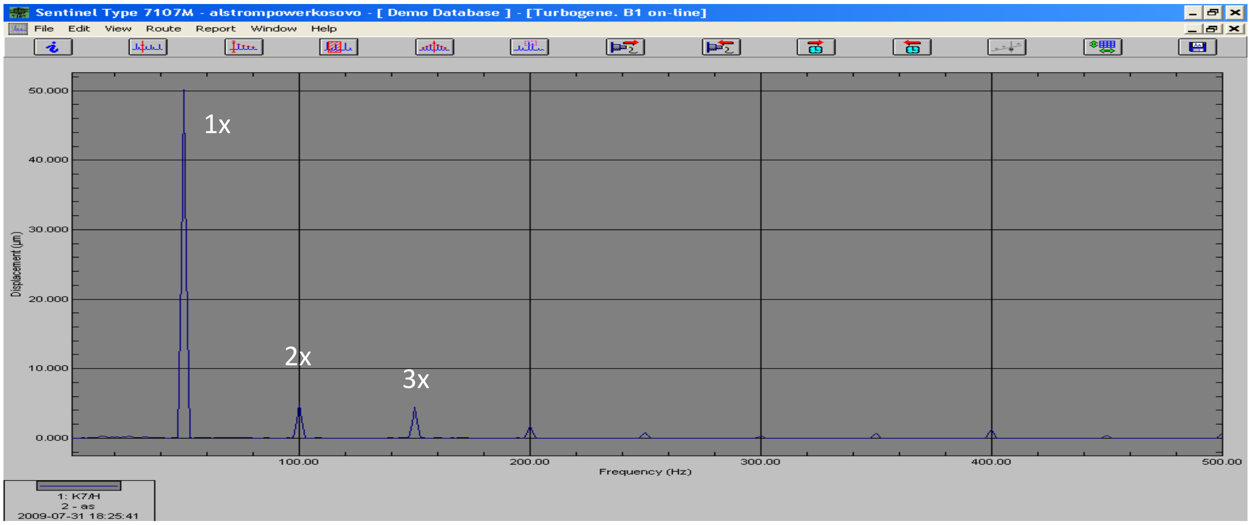


Fig. 21b: The FFT spectrum taken from B7/V vibration signal

As shown in Table 6 and from the FFT spectrums collected, the relative shaft vibration results didn't show any critical value as much as absolute vibration measurements but indicated the same nature of vibration problems. The relative shaft vibration measurements on the bearing B7 and B8 changed when the reactive power was 115 and 130 MVar (Figure 22, on-line monitoring vibration system VM600).

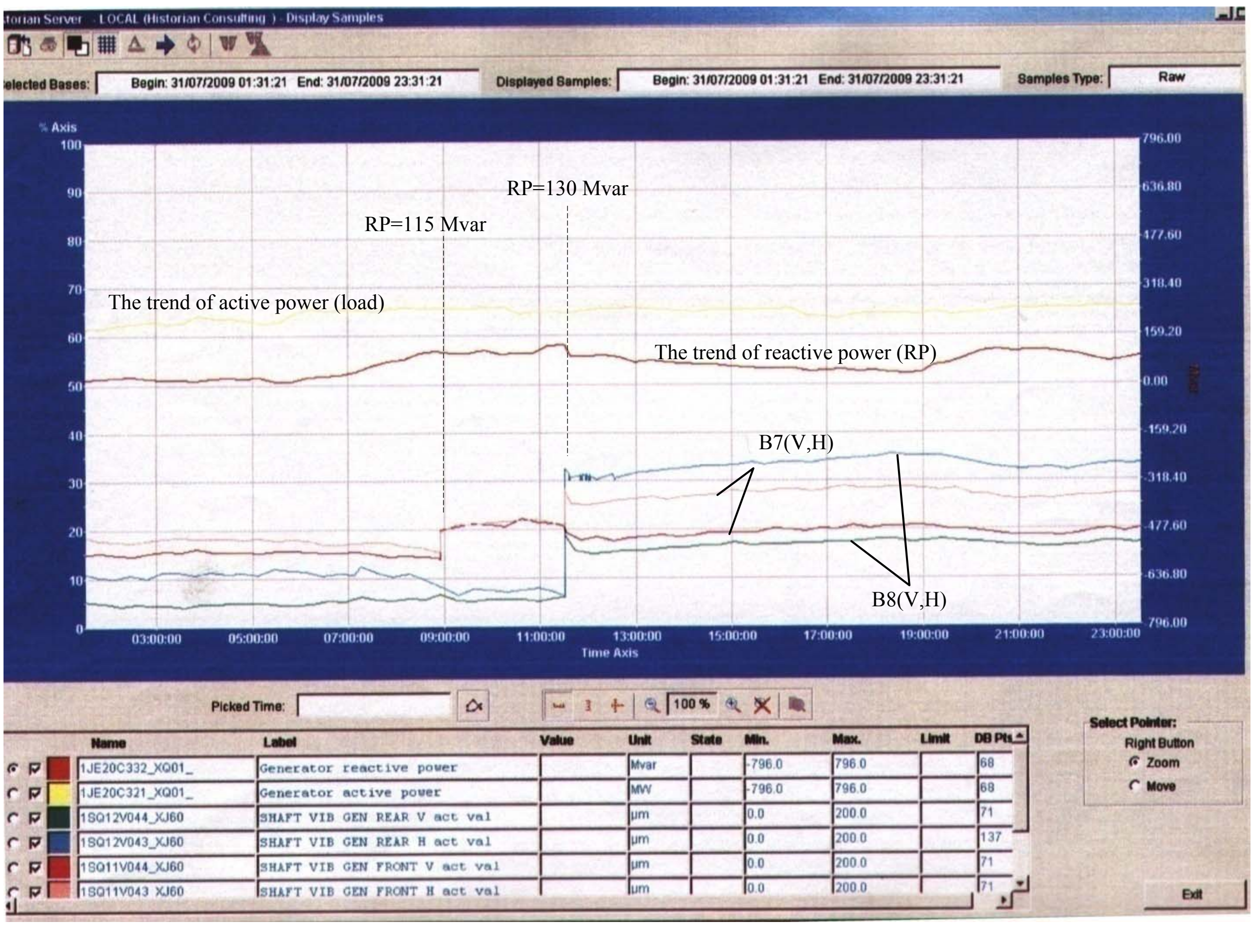


Figure 22: The trends of active and reactive powers, and relative shaft vibration measurements of the two support bearings of electrical generator (B7/V, H and B8/V, H), taken from the TPP "Kosovo B" unit B1 control room

4. PROJECT FINDINGS

4.1 The Analysis of all Information Collected and Diagnosis Vibration Problems

When analyzing the vibrations of the machine we should look at three main components of the vibration signal: its amplitude, its frequency and phase.

The amplitude - is the size of the vibration signal and determines the severity of the fault. The higher amplitude indicates the bigger problem.

Frequency (Hz) - describes the oscillation rate of the vibration (how frequently an object vibrates back and forward). The frequency at which the vibration occurs indicates the type of fault (unbalance, misalignment, mechanical looseness). By establishing the frequency at which the vibration occurs, we get a clear picture of what could be causing it.

Phase (Angle) - is one of the most important vibration analysis tools an analyst can have at his/her disposal. The analyst uses phase when trying to balance an unbalanced rotor to locate the heavy spot. Phase is also a useful tool to determine the types of unbalance, misalignment, looseness, soft foot, bearing misalignment, resonance and other machinery faults.

These three components of vibration signals provide a basis for identifying the root cause of vibration.

Based on the results of vibration measurements, waveform spectrums, FFT spectrums and phase data collected from VIBSCANNER and B&K instruments I could identify the following information to the measurement points B7/V and B8/V:

- The waveform spectrum is sinusoidal for the two measurement points B7/V and B8/V (fig. 18a and 19a)

- The highest amplitude vibrations on the FFT spectrums are in 1x or 50 Hz, excluding the FFT vibration spectrum on the B7/V, where there is also a little vibration in 2x or 100 Hz (fig. 18b)

- At the idle running (3000 rpm with no electricity produced) the phase and vibration are constant (that was checked during the vibration measurements).

- The angle between two vectors (amplitude vibrations) of vibration measurements B7/V and B8/V at 1x (3000 rpm, idle running) is opposite, nearly 180º (see Table 5 and Fig. 24). The phase is read in the opposite direction of the generator rotor rotation.

- The highest amplitude vibration of the two measurement points is in the vertical direction (Table 4).

- The amplitude vibrations were depending on the reactive power (the amplitude vibrations changed when the reactive power was increased) Figure 22.

Many vibration books such as “Machinery vibration – Balancing” written by Victor Wowk; The Vibration Analysis Handbook (A Practical Guide for Solving Rotating Machinery Problem) written by James L. Trylor; Vibration Diagnostic Guide –SKF reliability and others emphasize that the key characteristics of vibration caused by rotor unbalance are: 1) its waveform is sinusoidal, 2) the vibration amplitude is present at a frequency of once per revolution or 1x, 3) the amplitude increases with speed, 4) the amplitude and phase are constant at running speed. By comparing the above characteristics with our vibration results collected from the off-line and on-line instruments we can conclude that the main problem of increase vibration on the bearing B7/V and B8/V is **dynamic generator rotor unbalance.** In our case the generator rotor is twice bent because the amplitude vibrations on the two measurement points (B7/V and B8/V) are opposite Figure 23, 24 and 25.

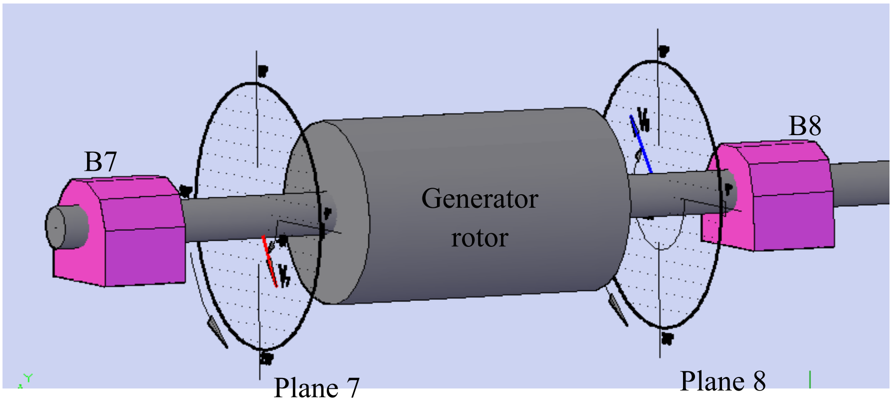


Fig.23: The plane 7 and 8 in Generator Rotor

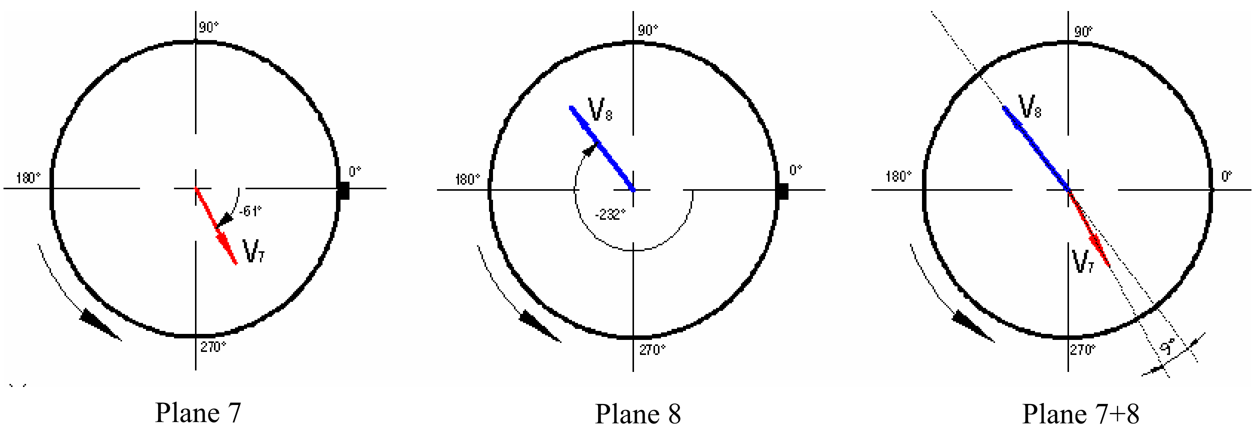


Fig. 24: The amplitude vibrations (vectors) and phase for two measured vibration points B7/V and B8/V at the idle running - 3000 rpm

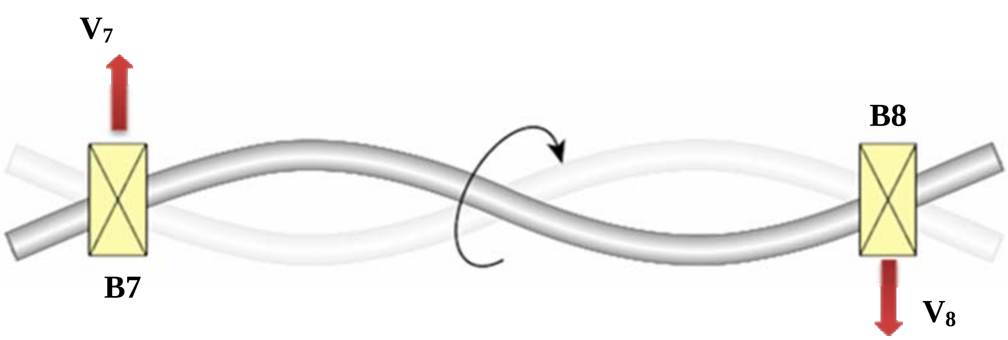


Fig.25: The bent form of Generator Rotor

The only dynamic generator rotor unbalance isn't the main problem of increase vibration in the electrical generator. During our test of how the amplitude vibration changes if the reactive power is increased for the same load (this phenomenon caused heating up the generator rotor), the relative shaft vibration on the electrical generator changes (Figure 22). When the reactive power was increased to 115 MVar, the amplitude of shaft vibration on the measurement points B7/V, B7/H was increased, while the vibration on the B8/H was decreased (B8/V-constant). By increasing the reactive power to 130 Mvar, again the shaft vibration changed. Now the amplitude of vibration on the measurement points B8/V, B8/H and B7/H increased, while the amplitude vibration on the B7/V was approximately constant. With this phenomenon, we could conclude that the vibrations on the electrical generator depend on the reactive power. This is a symptom of **thermal rotor instability**. Some of the rotor winding bars are elongated when the rotor is heating up and after the cooling process the vibrations do not return to the same position (an example is shown in Fig. 26). To return the amplitude vibrations to the previous value one needs to increase the reactive power to 155 MVar (see appendix B, Fig. 33).

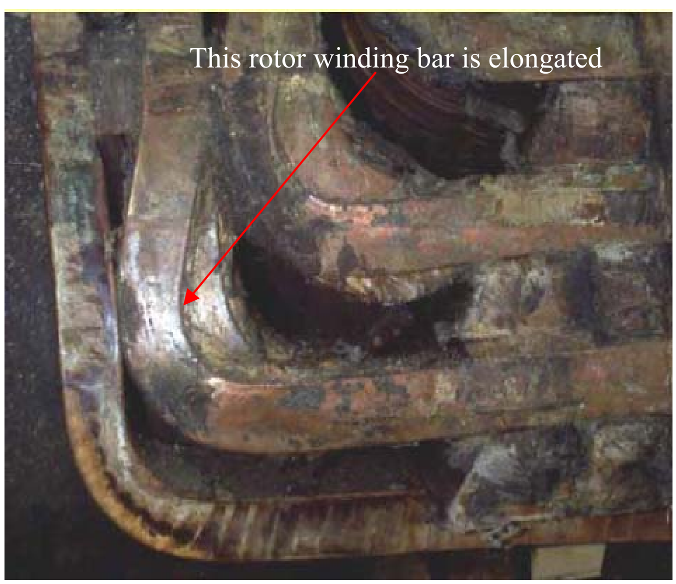


Fig. 26: An example, when Generator Rotor winding bars are elongated

The vibrations on the bearing B6/A have the same symptoms as on the bearing B7/V and B8/V. From the FFT spectrum taken on the B6/A vibration signal we can identify that the high amplitude vibration is one per revolution or 50 Hz (Figure 17b). The difference is that the high vibration value on the bearing B6 is in the axial direction, which indicates another source of vibration problem. The books “Machinery vibration-Alignment” written by Victor Wowk and “The Vibration Analysis Handbook” written by James L. Trylor also emphasize that if the axial vibrations are high in 1x, or once per revolution, the problem of increased axial vibration should be angular misalignment (Figure 27). The angular shaft misalignment is said to occur when the centerlines of rotation of two machine shafts are supposed to be collinear but are not in line with each other.

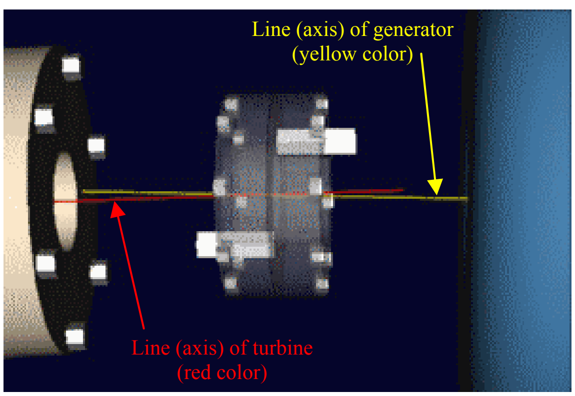


Figure 27: Angular misalignment

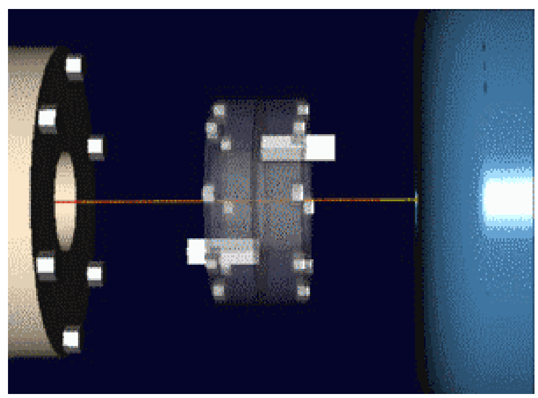


Figure 28: Good alignment

Finally, from the analysis of vibration problems in the STG-B1 machine we have found two main results:

- **The main problem of increased vibration in the electrical generator (B7/V, B8/V) might be generator rotor unbalance and its thermal instability.**
- **The increase vibration on the B6/A might be the result of coupling misalignment between the LP rotor and generator rotor, and high vibration levels in the electrical generator (B7/V and B8/V).**

5. CONCLUSIONS AND RECOMENDATIONS

Based on the results of these vibration measurements, and analysis and diagnosis of the vibration problems in the STG-B1 machine, I come to the following conclusions and recommendations:

5.1 Conclusions

1. The main problem of increase vibration on the measurement points B7/V and B8/V should be dynamic generator rotor unbalance. To reduce the vibration a dynamic generator rotor balancing process on the two planes in-situ should be made (see appendix C).

2. Working only on the dynamic generator rotor balancing will not solve the vibration problems for the long-term machine operation due to the generator rotor thermal instability (elongated winding bars when the reactive power goes up)

3. The generator rotor winding bars should be repaired because the results of the vibration analysis measurements indicated this is a problem.

4. For the long-term of the STG-B1 machine operation, a general generator rotor repair is necessary (rotor winding bars repair and balancing).

5. Based on the trend of the vibration measurements in Figure 2, I hope that after the dynamic generator rotor balancing is done, the vibrations in the B6/A will be decreased, but not to an acceptable level.

6. The LP rotor must be repaired or changed with a new one because it is limiting the unit’s capacity.

7. The cause of the increase vibrations in the B6/A is due to the shaft misalignment between the LP rotor and generator rotor, and the high vibration level in the Electrical Generator.

5.2 Recommendations

The following will be my recommended solutions with an implementation plan for corrective action in the STG-B1 machine.

To solve the vibration problems in the STG-B1 or to decrease vibration in the measurement points B6/A, B7/V and B8/V I recommend two options:

Option-1

1. In-situ generator rotor balancing (that should solve the vibration level on the measurement points B7/V and B8/V.

2. Correcting the shaft misalignment between the LP rotor and generator rotor (that should solve the vibration problem in the measurement point B6/A).

Option-2

1. **General generator rotor repair** should include: generator rotor winding bars repair and balancing in the workshop. This option also includes correcting the shaft misalignment between the LP rotor and generator rotor.

With the corrective action option-1, the vibration problems in the STG-B1 machine will be solved much more quickly than with corrective action option-2. However, option-1 will not guarantee the long-term operation reliability of the machine due to the other problems such as the generator rotor thermal instability.

The corrective action option-2 will solve the vibration problems on the STB-B1 for the long-term operation because the three main vibration problems such as the generator rotor unbalance, generator rotor thermal instability and misalignment between the LP rotor and generator rotor will be solved. This corrective action, as shown in the implementation plan for corrective action option-2 will requires more time and budget than corrective action option-1.

Remark:

The other problem that is degrading the normal operation on the STG-B1 machine is Low Pressure (LP) rotor (limited the unit capacity). The vibration problems on the STG-B1 machine are not a direct result of the LP rotor; therefore it will not be included in my project plan.

5.2.1 Recommended Implementation Plan for Corrective Action Option-1

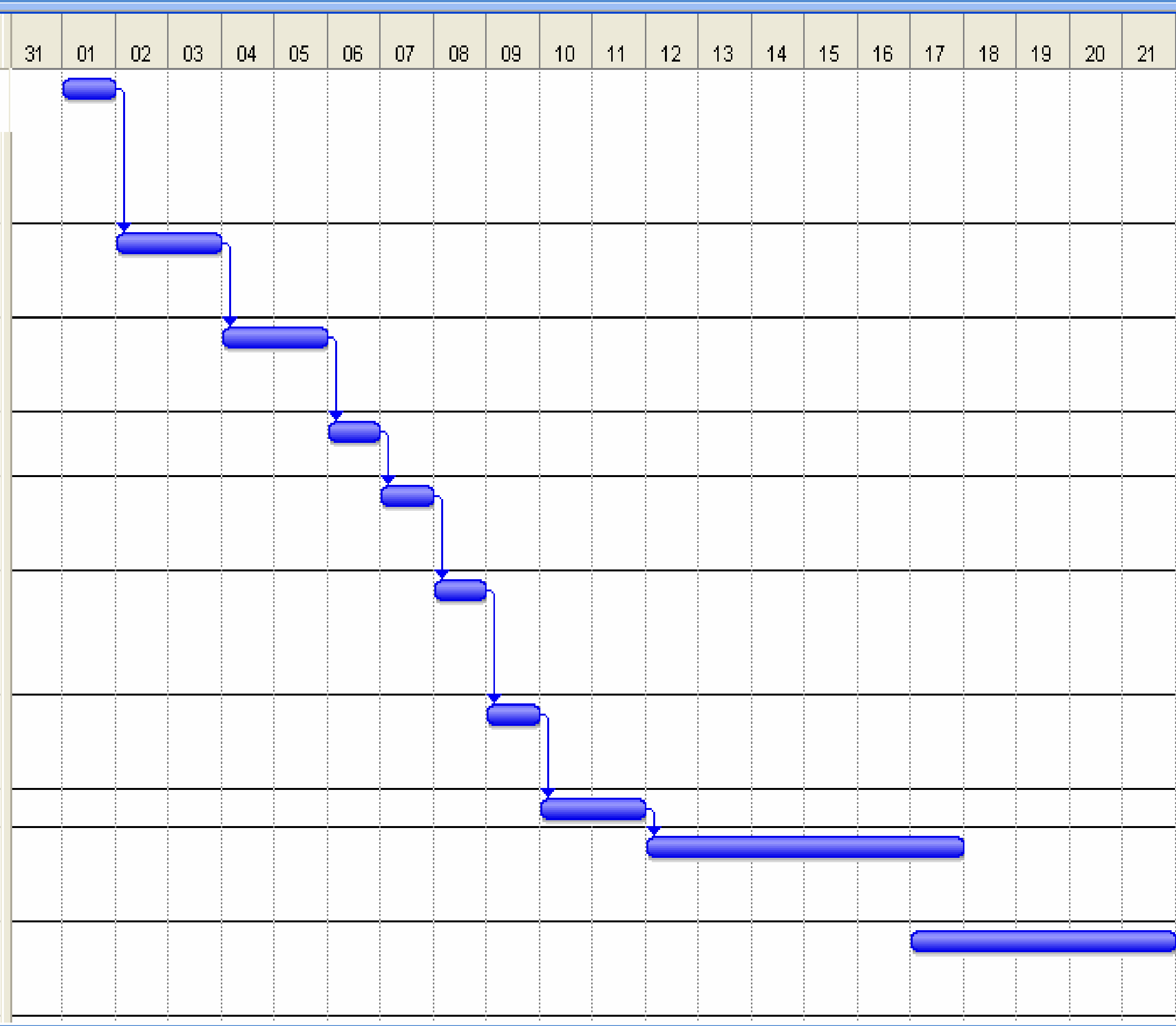
The implementation plan for corrective action option-1 includes: in-situ generator rotor balancing and correcting the misalignment between the LP rotor and generator rotor. In Introduction to Project Management class (Meredith, R. J., and Mantel. S. J. (2005)) we learned that for the successful implementation of a plan we should take into consideration the following three objectives: performance, time and budget. Regarding these, I prepared the plan of work, timeline and budget for this corrective action as shown in Table 7.

If the vibrations issues in step 7 (Table 7) are not improved, the activity will continue until the corrections reach an acceptable level. Here, we should also take into consideration the time schedule risk.

Table 7: The plan of work and timeline for corrective action option-1 (Gent Chart)

Gantt Chart

	Task Name	Duration
1	Start run the STG-B1 machine to 3000 rpm and measure the vibration level and phase on the two bearing pedestals of the electrical generator (B7 and B8)	1 day?
2	Stop run the machine, cooling the turbine and discharge H2 from electrical generator	2 days
3	Disassembly necessary elements on the electrical generator and attach the test weight on the plane 7 and 8	2 days
4	Assembly necessaty generator elements and prepare it to start run	1 day
5	Start run the machine to 3000 rpm and measure the vibration level and phase with attached test weight	1 day
6	Stop run the machine, disassembly necessity elements on the generator, remove test wight and attached correction weight	1 day
7	Start run the machine to 3000 rpm and check the improvement vibration level	1 day?
8	Cooling turbine	2 days
9	Mounting all elements in electrical generator and prepare it for long-term operation	4 days
10	Checking and correcting shaft alignment between LP and generator rotor	5 days



Budget:

For corrective action option-1 the following is necessary:

1. An expert for measuring vibration and balancing	10 days x 3000 €/day = 30 000 €
2. Maintenance workers	
- For balancing generator rotor	8 workers x 10 days x 80 €/day =5600 €
. - For correcting shaft misalignment	5 workers x 5 days x 80 €/day=2000 €

This calculation is predicated on a 10 hour work day

3. Unit operation expenditures (heavy fuel oil, chemicals, water, electricity, operation personnel) for a start run is approximately 100 000 €

3 start up x 100 000=300 000 €

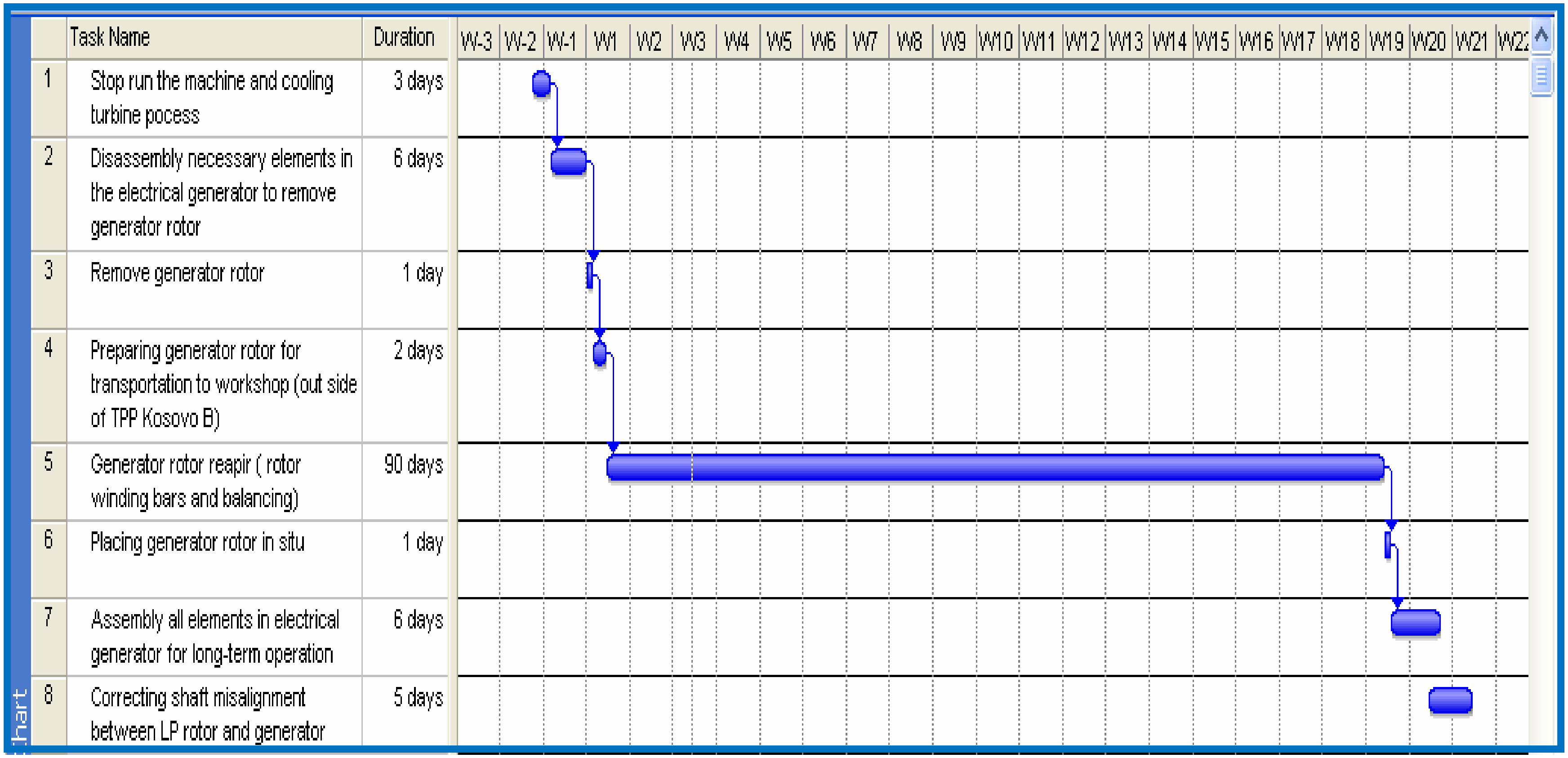
Total expenditure for corrective action option-1 will be: 337 000 €

In Appendix C, I will also described the recommended corrective action for in-situ generator rotor balancing, performed without the need to disassemble the rotor and balancing it in the workshop (source: a handbook taken from my consultant Mr. Remzi Shahini).

5.2.2 Recommended Implementation Plan for Corrective Action Option-2

The implementation plan for corrective action option-2 includes: general generator rotor repair in the workshop (Generator Rotor winding bars repair and balancing) and also correcting the shaft misalignment between the LP rotor and generator rotor. The plan of work and timeline for this corrective action is shown in Table 8.

Table 8: The plan of work and timeline for corrective action option-2 (Gent Chart)



	Task Name	Duration
1	Stop run the machine and cooling turbine pocess	3 days
2	Disassembly necessary elements in the electrical generator to remove generator rotor	6 days
3	Remove generator rotor	1 day
4	Preparing generator rotor for transportation to workshop (out side of TPP Kosovo B)	2 days
5	Generator rotor reapir (rotor winding bars and balancing)	90 days
6	Placing generator rotor in situ	1 day
7	Assembly all elements in electrical generator for long-term operation	6 days
8	Correcting shaft misalignment between LP rotor and generator	5 days

Budget: The budget for corrective action option-2 is very high. According to my consultant, Mr. Remzi Shahini, who has experience overhauling machines, the total costs for repairing the generator rotor winding bars, balancing, rotor transportation and other tasks are at approximately 2 million euro.

6. PROJECT EVALUATION

The results of measuring the vibrations in the machine after corrective actions have been taken by the TPP “Kosovo B” maintenance personnel or others is an indicator that the project of analysis and diagnosis of the vibrations, together with corrective action taken, will show whether or not this has or hasn’t been successful. If vibration levels are decreased this will be a signal that the project was successful.

APPENDIX A: List of Symbols

STG- Steam Turbo-Generator

EG – Electrical Generator

STG-B1 Steam Turbo-Generator machine of unit B1

TPP – Thermal Power Plant

C – Condenser

CP – Condensate Pump

LPHE – Low Pressure Heating Exchanger

WT – Water Tank

FP- Feed pump

HPHE – High Pressure Heating Exchanger

SB – Steam Boiler

M – Mill

CSL- Coal Supply Line

HFOL- Heavy Fuel Oil Line

GL – Gas Line

FAL – Fresh Air Line

FDF – Forced Draft Fan

AH – Air Heater

F - Filter

RCF – Recirculation Fan

RCFGL- Recirculation Fuel Gas Line

IDF – Induced Draft Fan

CH - Chimney

SL – Steam Line

EP – Electricity Transformer

RMS – Rout Mean Square

OP – Zero Peak

PP – Peak to Peak

CT – Cooling Tower

RPM – Revolutions per Minute

B1, B2… B8 – Journal Bearings

LPC - Low Pressure Cylinder

IPC - Intermediate Pressure Cylinder

HPC - High Pressure Cylinder

APPENDIX B: Machine Schematics

In appendix A will be shown the scheme of turbine oil lubrication, turbine operational parameters (turbine control), temperature of the bearings, turbine glade steam.

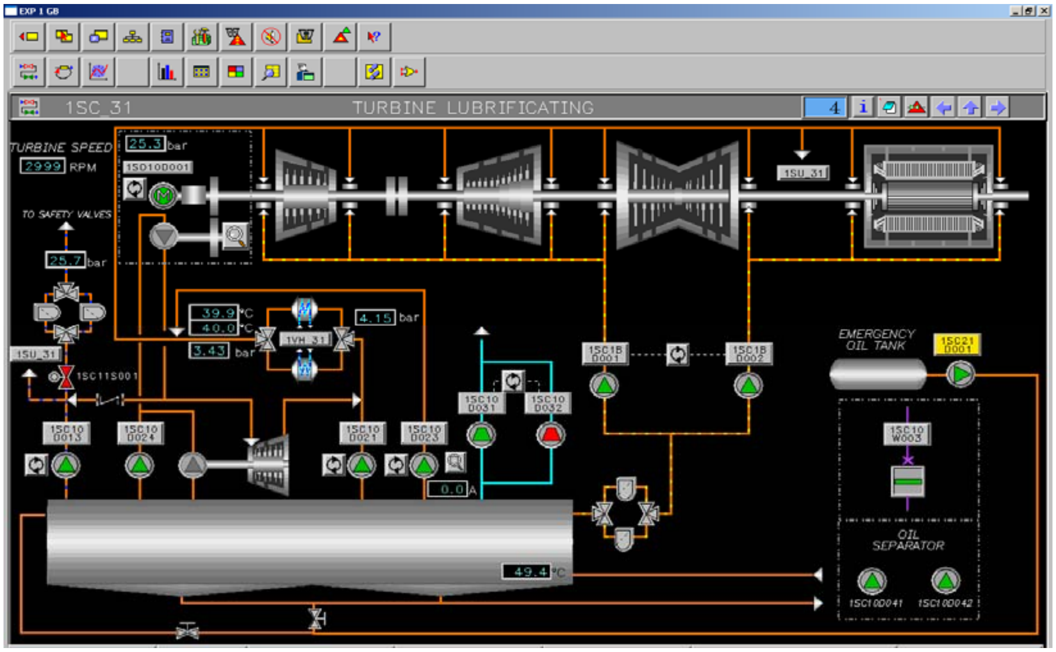


Fig.29: The scheme of bearing center oil lubrication

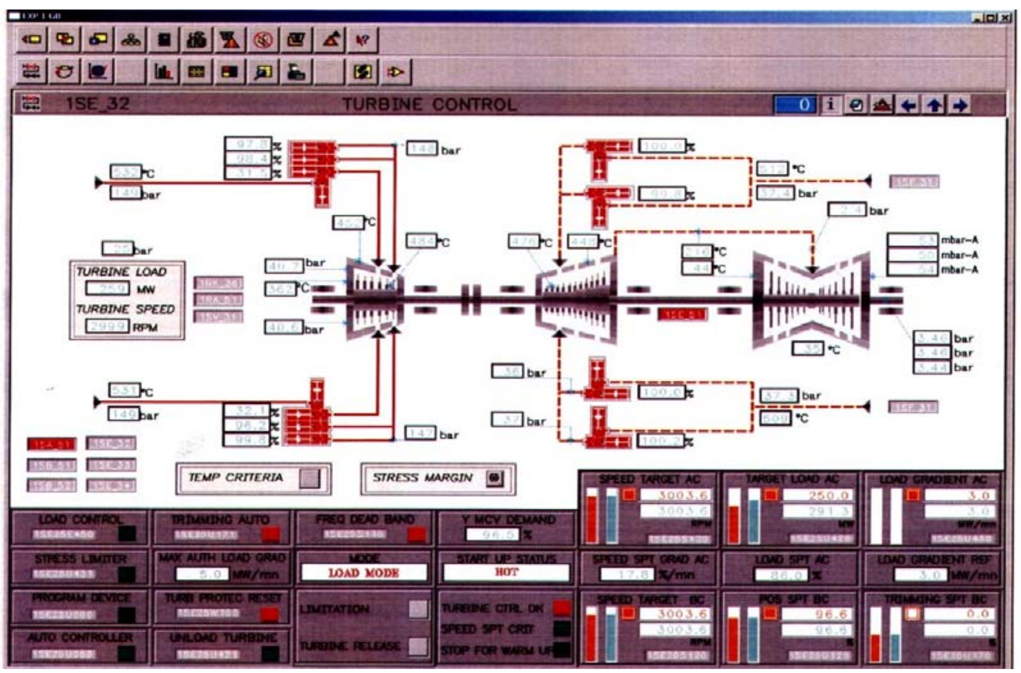


Fig. 30: The scheme of turbine control

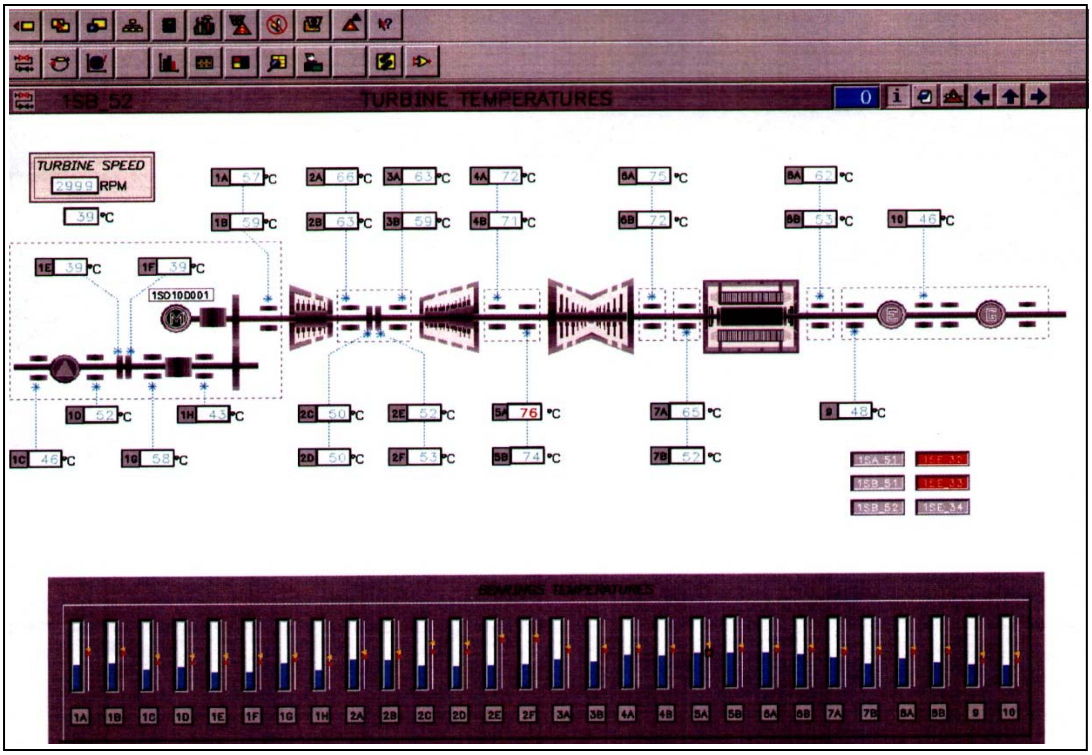


Fig. 31: The scheme of bearing turbine temperature (250 MW)

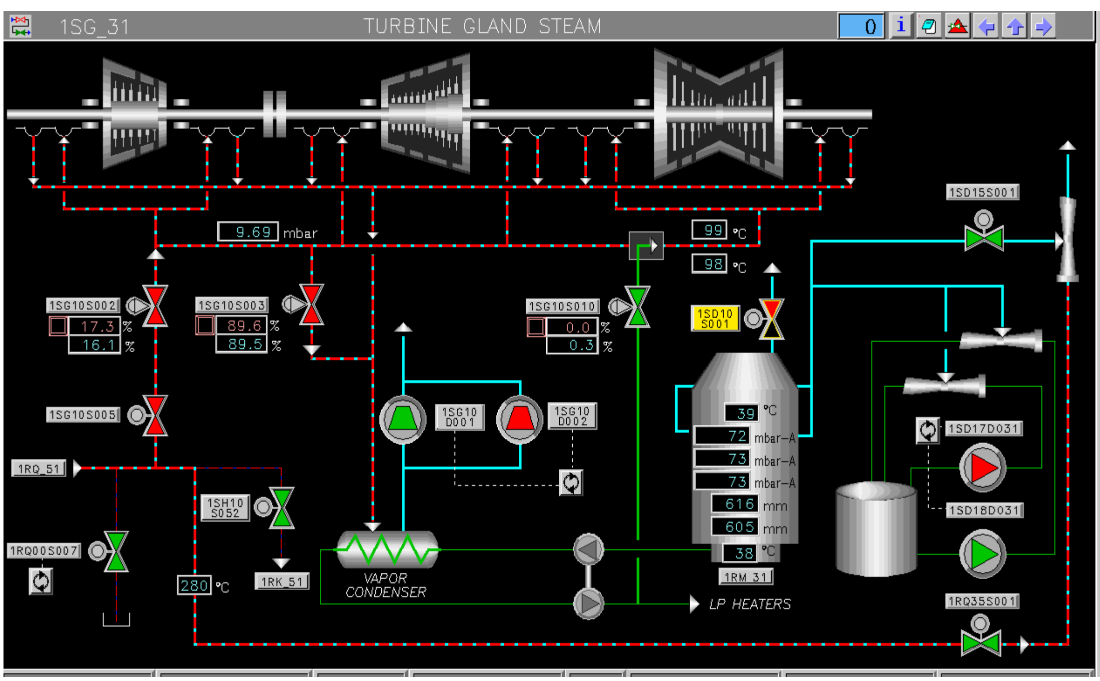


Fig. 32: The scheme of turbine gland steam

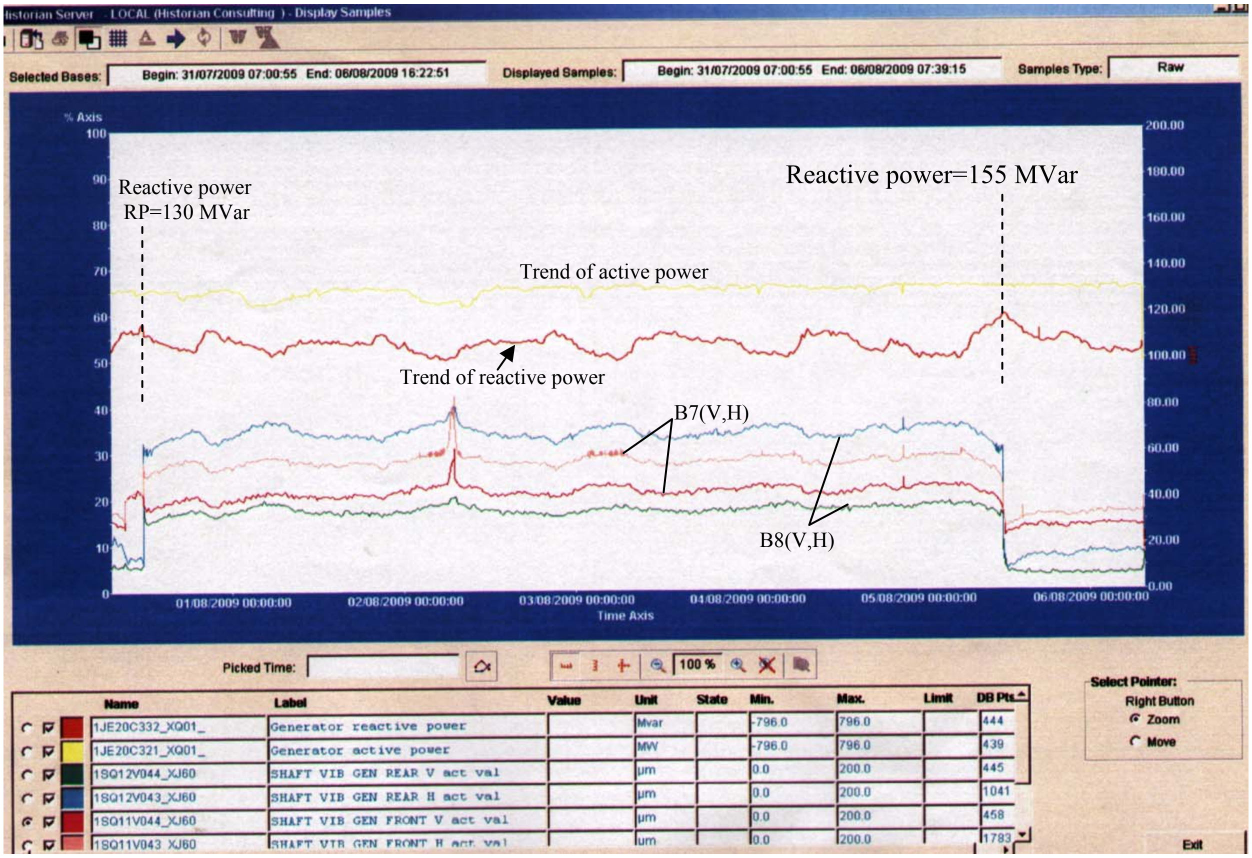


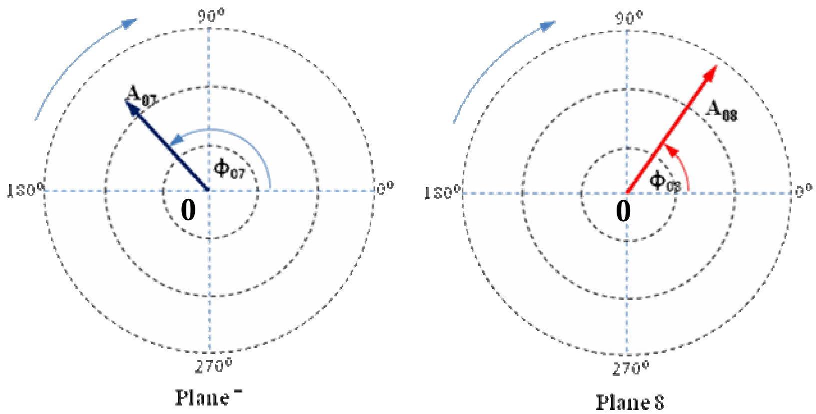
Fig. 33: When the reactive power was 155 MVar, the relative shaft vibrations in the electrical generator were reduced

APPENDIX C: Corrective Action (In-situ Generator Rotor Balancing)

The procedure of in-situ generator rotor balancing is as fellows:

1. Start run the machine to idle running (3000 rpm) and measure the vibration and phase in once per revolution or 1x (A_{07} @Φ_{07} and A_{08} @Φ_{08}) . The angle (Φ_{07} , Φ_{08}) is readings in the counter direction of the generator rotor rotation.

A_{07} and Φ_{07} A_{08} and Φ_{08}



2. Stop –run the machine and make calculation for balancing methods (static or dynamic balancing). In this case we should find the A_{0S} amplitude vibration and phase Φ_{0S}. The vector A_{0S} is the disatnce from **0** to point **c** **(**Fig. 34). The point **c** is the middle point of the segment **ab**.

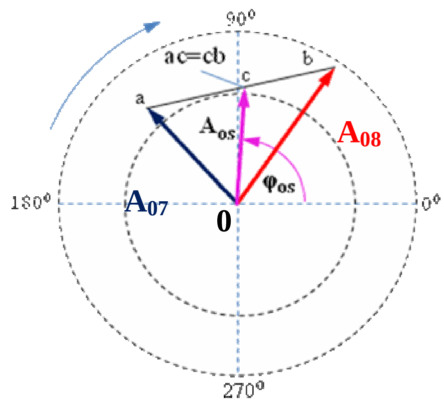


Figure 34. Plane 7+8

If we decide to make a static balance, the procedure will be as follows:

3. Attach the test (trial) weight M_{t7}@ Φ_{t7} and M_{t8}@ Φ_{t8} arbitrary in plane 7 and 8 in the same angle Φ_{t7}=Φ_{t8} (Figure 35 and 36). Normally, the attached test weight should change the vibration and angle approximately 25 % of the initial vibration.

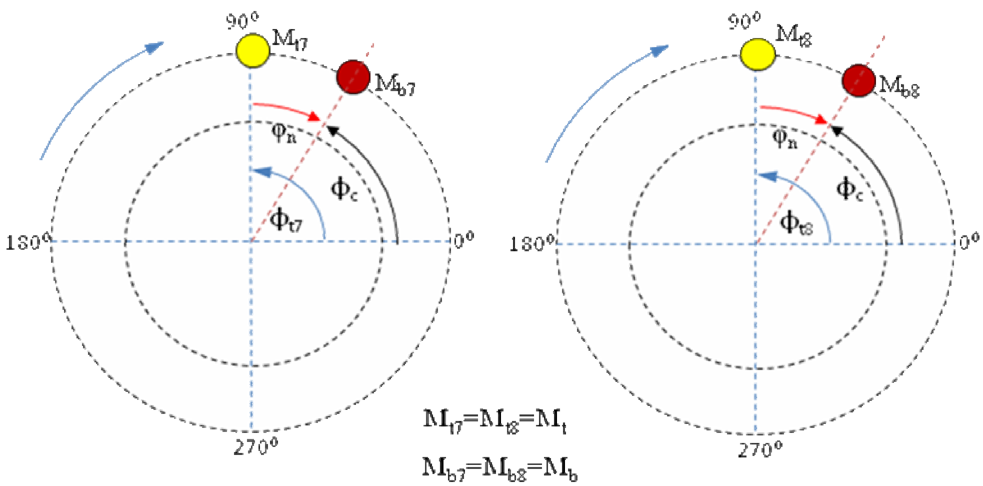


Fig. 35: The attached test weight in plane 7 and 8

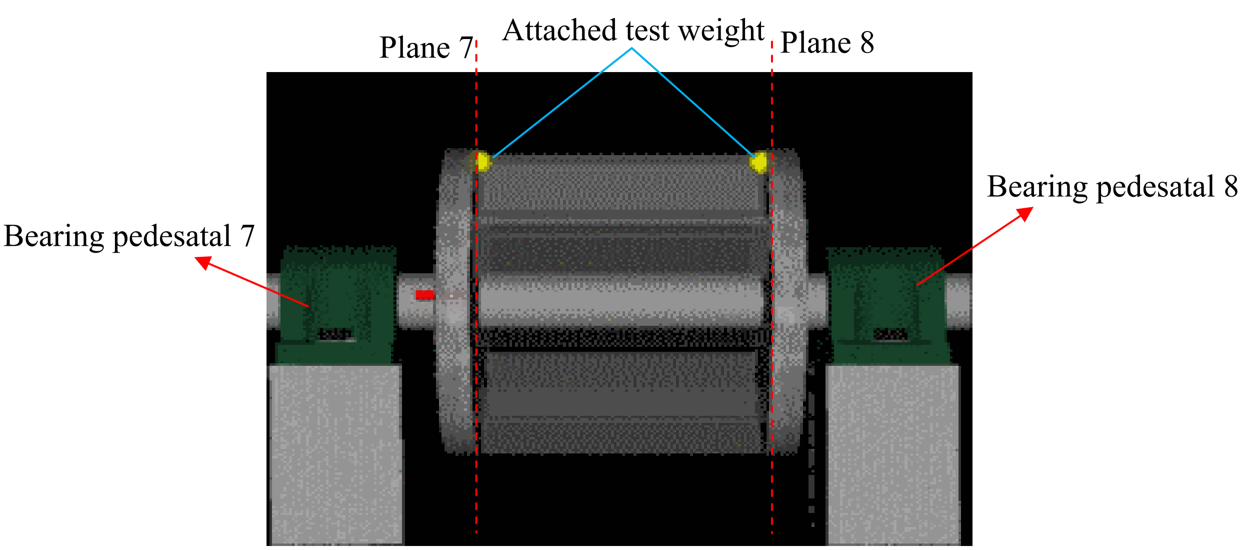


Figure 36: The attached test weight in the same angle in plane 7 and 8

4. Start run the machine and again measure the amplitude vibration and phase with the attached test weight in the rotor. During this run we have recorded A_{17} @ Φ_{17} and A_{18} /@Φ_{18} amplitude vibrations and phases (Figure 37). In the same way we also can calculate the vector A_{1S} and angle Φ_{1S} (Fig. 37).

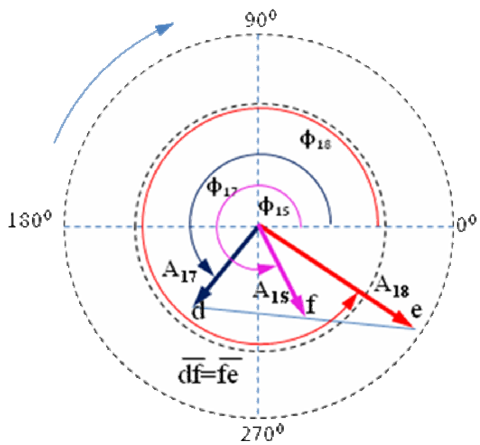


Figure 37: The calculation of A_{1S} vector

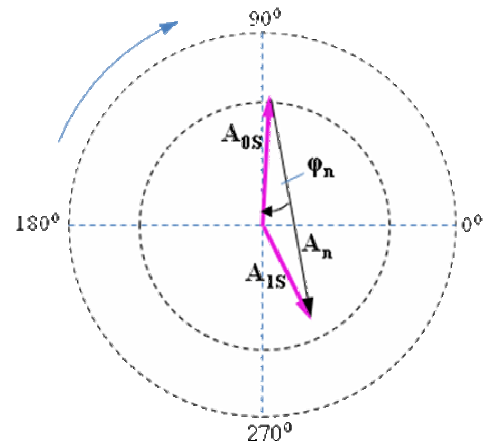


Figure 38: The calculation of A_n vector

5. Stop run the machine and removes the test weights. Calculate the vector A_n and angle φ_n (Fig. 38). The correction weight (balancing weight) will be calculated with formula:

$$M_{b7}=M_{b8}= M_t\,(A_{0S}/\,A_n)$$

While the angle of attached correction weight (Figure 35) will be: $\Phi_c=\Phi_t-\varphi_n$

6. Attach correction weight and start run the machine to check amplitude vibration improvement.

If we decide to make a dynamic balance, the procedure will be as follows:

3' After are known the initial vibration measurements and phases, we can identify also the vectors A_{p7}@Φ_{p7} and A_{p8}@Φ_{p8} as shown in figure 39.

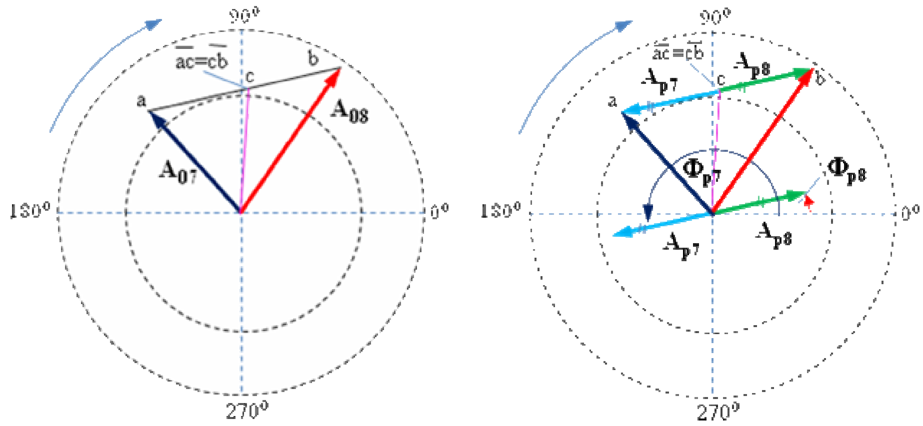


Figure 39: The calculation of $A_{p7}@\Phi_{p7}$ and $A_{p8}@\Phi_{p8}$

4'. Attach the test (trial) weight $M_{t7}@\Phi_{t7}$ and $M_{t8}@\Phi_{t8}$ arbitrary in plane 7 and 8 in the opposite angle (180º) figure 40.

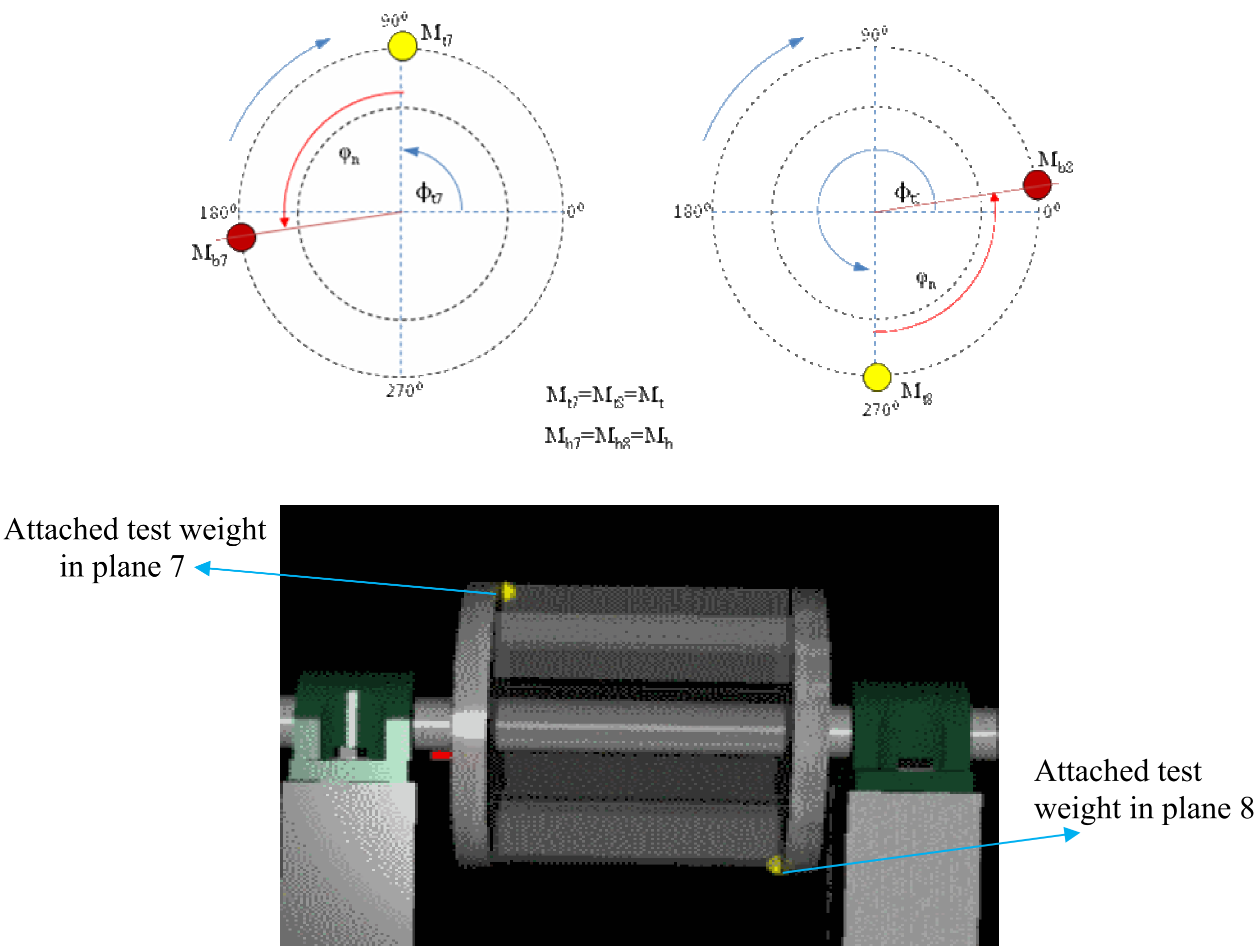


Figure 40: The attached test weight $M_{t7}@\Phi_{t7}$ and $M_{t8}@\Phi_{t8}$

5'. Start run the machine and again measure vibration amplitude and phase with the attached test weight in the rotor. During this run we have recorded vibration amplitude and phase $A_{11}@\Phi_{11}$ and $A_{12}@\Phi_{12}$. In the same way we can calculate the vectors A_{q7} and A_{q8} (Fig. 41).

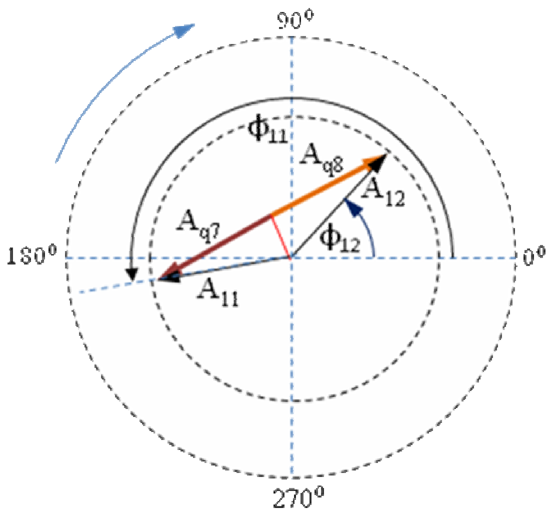


Fig. 41: The calculation of A_{q7} and A_{q8}

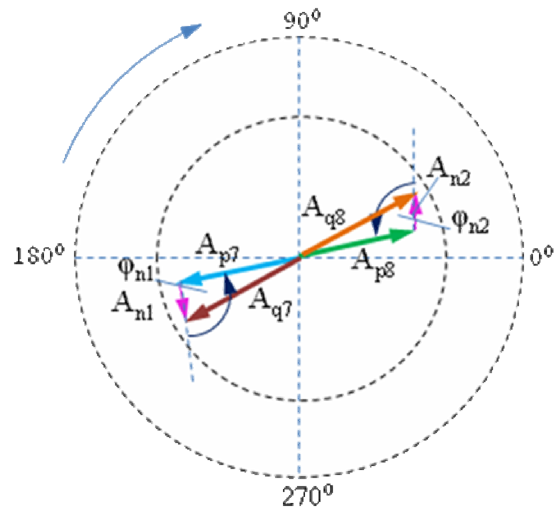


Fig. 42: The calculation of $A_{n1}@\varphi_{n1}$ and $A_{n2}@\varphi_{n2}$

6'. Stop run the machine and calculate the vector $A_{n1}@\varphi_{n1}$ and $A_{n2}@\varphi_{n2}$ (Fig. 42). The correction weight (balancing weight) will be calculated with formula:

$$M_{b7} = M_{b8} = M_{t7}\,(A_{p7}/ A_{n1}) = M_{t8}\,(A_{p8}/ A_{n2})$$

The location of attached correction weight is offset for angle φ_n from the test weight

7'. Run the machine and check the amplitude vibration improvement.

APPENDIX D: List of References

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