The Design and Development of a Mechanical Faults Simulation Test Rig for Educational Purposes

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Abstract. The paper describes the process of designing and developing a mechanical faults simulation test rig to support courses on gas turbine diagnostic methods. In this context the test rig is designed to reinforce students understanding in rotor dynamics, instrumentation and measurements analysis, acquiring at the same time experience on mechanical faults symptoms and diagnosis. The design was undertaken by a student, as part of fulfilling the requirements for obtaining a Mechanical Engineering Degree. The design requirements for fulfilling the defined educational objectives are discussed, as are the construction details of the test rig. The instrumentation and the data acquisition system characteristics are also presented, along with sample results of simulated representative faults that have significant value for educational purposes. Finally the benefits accruing from the use of the test rig as part of an educational procedure are discussed, as are the educational benefits for the student that has undertaken the project.

Keywords: Educational experimental apparatus, Vibration measurements, Gas turbine diagnosis
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INTRODUCTION

Vibration measurements are a vital part of the gas turbine condition monitoring systems and of the preventive maintenance technique. Vibration level is representative of the engine mechanical condition and each system has a specific healthy vibration range, depending on the manufacturing tolerances. Mechanical faults or engine degradation are usually connected to increased vibration level, thus diagnostic information can be acquired by proper analysis of vibration measurements. According to Boyce [1] vibration analysis coupled with engine performance analysis is an unbeatable tool as a total diagnostic system leading to maximum power plant utilization and significant cost savings.

In order to fully exploit the diagnostic data available in vibration measurements the engineers should have a clear understanding of rotor dynamics, instrumentation characteristics, statistical techniques used in vibration analysis and if possible some hands on experience of the typical faults that occur in rotating turbomachinery components. A method to convey to students the needed knowledge and experience is by using a suitable experimental test rig to support the theory of vibration analysis and fault identification. Although such test rigs can be found in the market [2], an approach that can be followed is to build such a facility.

The decision of designing and building the test rig was based on a number of benefits that this approach offers. An obvious benefit is that the construction can be tailored made, to suit the educational vision of the teacher and the courses objectives. This approach also allows the cost to be kept well below the market price of similar commercial test rigs. The main drive however is the fact that the design and development process can itself contribute to the education. The first aspect is that the student responsible for the design of the test rig is involved on an integrated project that deals with many disciplined of his studies, while at the same time it provides the satisfaction of seeing something calculated and drawn on “paper” to be actually materialized. A second very important aspect is that having the outcome of a student project on display and usage to future generations give them confidence on their knowledge and motivate them to pursue similar integrated projects. Based on this reasoning the work was offered as a student diploma project and materialized by a final year student in cooperation with the Laboratory of Thermal Turbomachines of National Technical University of Athens (LTT/NTUA) staff [3].

The present paper describes the design and development of the test rig, which is capable to simulate mechanical failures that occur in turbomachines, in a controlled environment and is fitted with typical instrumentation found in the field. The instrumentation and the data acquisition system characteristics are presented, along with sample results of simulated representative faults that have significant value for educational purposes. Finally the benefits accruing from the use of the test rig as part of an educational procedure are discussed, as are the educational benefits for the student that has undertaken the project.

DESIGN OF THE TEST RIG

Malfunctions in turbomachines are categorized both by their causes and their effects in operation. Two of the most common and most frequently appeared malfunctions in turbomachines are unbalance and misalignment.

Unbalance is a situation in which some asymmetry exists in the geometry of a rotating part of a machine with respect to the axis of rotation which has the result that the axis of inertia of the rotating part is different from its axis of rotation. There are four different types of unbalance, as seen in Figure 1.

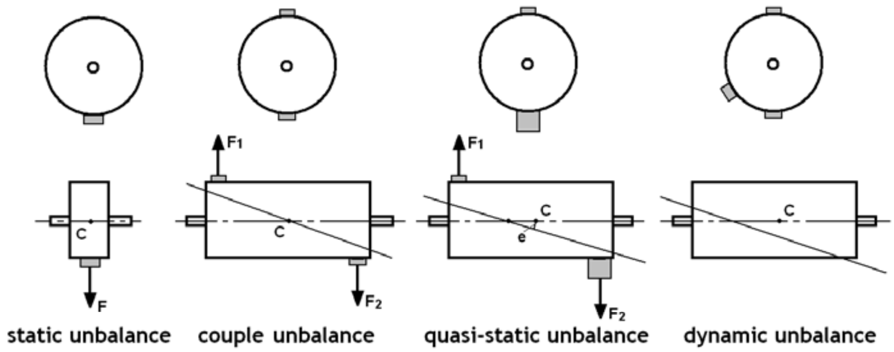


FIGURE 3. Types of Unbalance

Static unbalance appears when the centre of gravity lies off the rotation axis, while the axis of inertia is parallel to the axis of rotation. In this case the vibrations of both shaft ends will be in phase. Couple unbalance is defined as the condition of unbalance where the inertia axis crosses the axis of rotation on the gravity centre. Even though the shaft is statically balanced, it tends to vibrate about its center when it is rotated, while the vibrations of the shaft ends are in opposite phase (180 deg difference). Quasi-static unbalance is made up of a static unbalance and a couple unbalance. Quasi-static unbalance is the condition of unbalance where the inertia axis crosses the axis of rotation at a point different from gravity centre. The vibrations of shaft ends have unequal amplitudes but in phase. The fourth type of unbalance is dynamic unbalance and it is the most common type encountered in machinery. Dynamic unbalance is defined as unbalance where the axis of inertia and the axis of rotation do not intersect and they are not parallel. Dynamic unbalance often exhibits different amplitudes of vibration at each end of the shaft. In addition most often it exhibits phase angles that are neither in phase nor directly opposite from one another. Couple unbalance, quasi-static unbalance and dynamic unbalance cannot be corrected in a single plane but require corrections to be made in two or more planes.

Misalignment in couplings can appear in three basic types, as seen in Figure 2. The first is a parallel offset. In this type of misalignment, the two shafts can be offset vertically, horizontally or in a combination of both. The second type is angular misalignment. In this type of alignment, the angularity again can be in the vertical plane, the horizontal plane or in both planes. In most cases misalignment in couplings occur as a combination of both types.

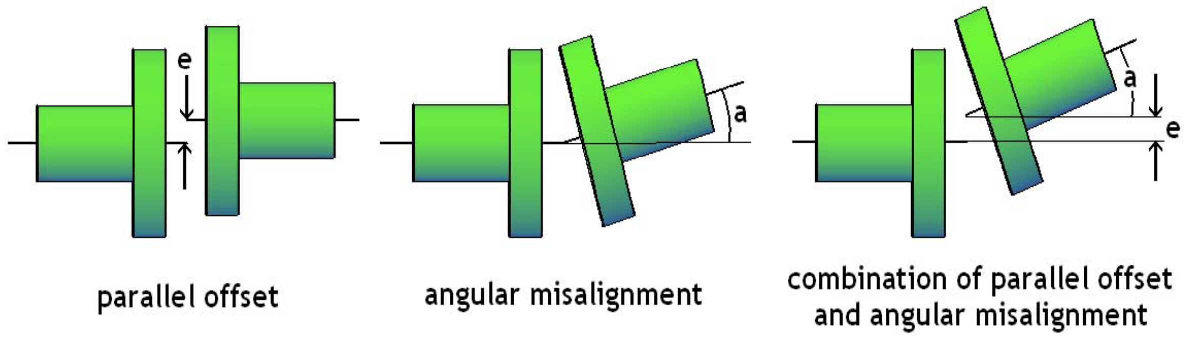


FIGURE 4. Types of Misalignment

Since misalignment and unbalance are the most common faults encountered in practice it was decided to design the test rig in order to simulate these two faults. An unbalanced shaft can easily be simulated by placing additional masses to specific positions to a properly balanced shaft. In order to simulate misalignment a coupling is needed in order to have the ability to move the shaft relative to the coupling. According to these rough guidelines the test rig should consist of a shaft with drilled holes at specific positions (unbalance), two bearing houses for the shaft, a motor, a suitable coupling, and a mechanism that will allow the movement of the shaft ends relative to the coupling (misalignment).

The test rig is tailored for supporting courses on gas turbine diagnosis, thus the shaft selected should be relevant to the topic, but also of relatively small dimensions.

Concerning the needed measurements the vibrations, in the form of velocity, or acceleration should be measured at the bearing positions. The use of proximity sensors could be problematic since the sensor must be close to the shaft, thus it wouldn't be possible to simulate cases of extreme misalignment without risking the probe integrity. Also rotational speed should be measured with good accuracy in order to characterize vibration frequency components.

Finally in order to increase the impact and the interactivity of the laboratory exercise, the results, in the form of faults signatures should be promptly available to the students. For this purpose a program that can analyze the vibration measurements in real time and produce power spectra in a graphical environment should be used.

DEVELOPMENT OF THE TEST RIG

The main part of the test rig is the rotating shaft. As discussed it should be relevant to the topic of turbomachinery. A small turbocharger shaft including the compressor and the turbine is ideal for this purpose, since it contains the turbomachinery components, giving a “real life” experience to the students. The selected shaft, which is a small turbocharger shaft, can be seen in Figure 3. The turbine rotor consists of 50 vanes and the compressor rotor consists of 20 blades. The total length of the shaft is 486 mm and its total weight is 6.9kg.

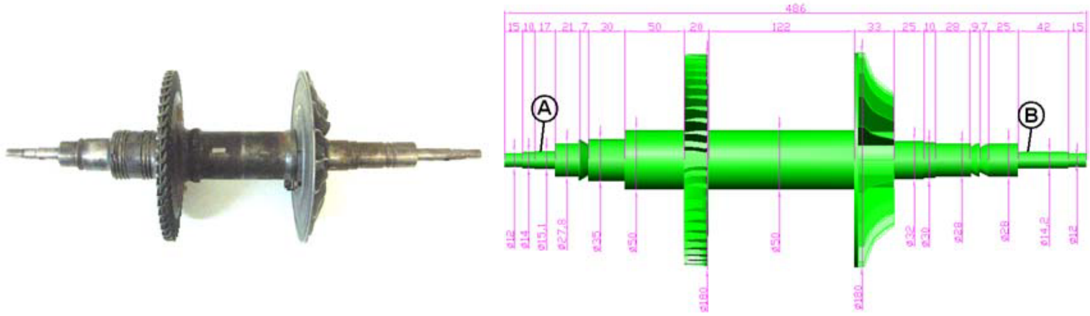


FIGURE 3. The selected shaft, its dimensions and seating positions (A, B)

The shaft was decided to be seated at two points (points A and B, Figure 3). The bearings used are deep groove ball bearings with an inner diameter of 15mm. The bearings can accept strong radial forces but can also accept axial forces. For their adaption to the shaft a ring was used.

In order to have the ability to balance the shaft, but also to simulate unbalance the addition or removal of masses from specific places around the rotating axis should be possible. In order to achieve this holes are drilled at the turbine and the compressor planes. At each plane 8 holes are drilled. Their radial position is 53.75mm for the turbine plane and 46mm for the compressor plane. The angle between two holes is constant and equal to 45 degrees for both planes. Added masses are small screws of different length.

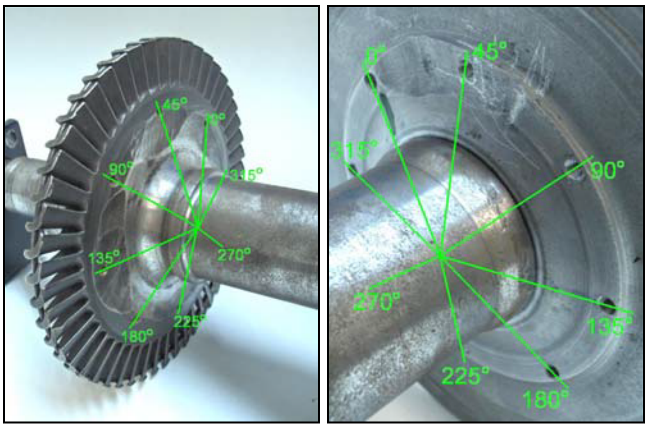


FIGURE 4. Places for balancing masses at turbine and compressor plane

In order to simulate misalignment the two shaft seatings should be capable of vertical movement. An appropriate mechanism identical for the two seatings has been constructed and can be seen in Figure 5. The main seating part is L shaped (item 1) which is also part of the bearing house. Side to side holes have been drilled in order to allow the pins (items 2) to slide inside item 1. Item 1 is risen by a screw rod (item 3). A clockwise turn of the rod results to the tightening of item 1 to the seating assembly. A counterclockwise turn of the rod results to the vertical rise of item 1. Since both shaft ends should be capable of vertical movement for the purpose of misalignment simulation the coupling of the shaft to the motor couldn't be a commercial one. This is due to the fact that commercial couplings use elastic rings in order to minimize misalignment effects. A non elastic coupling was designed and constructed. The coupling consists of two separate parts, one fitted to the shaft and one fitted to the motor. The electromotor shaft and the rotating shaft have been properly formed in order to make possible the conjunction of coupling parts to them with screws. Conjunction of coupling parts to each other uses M6 screws placed in side to side drilled Ø6 holes (6 holes drilled symmetrically to rotating axis). The constructed coupling can be seen in Figure 6.

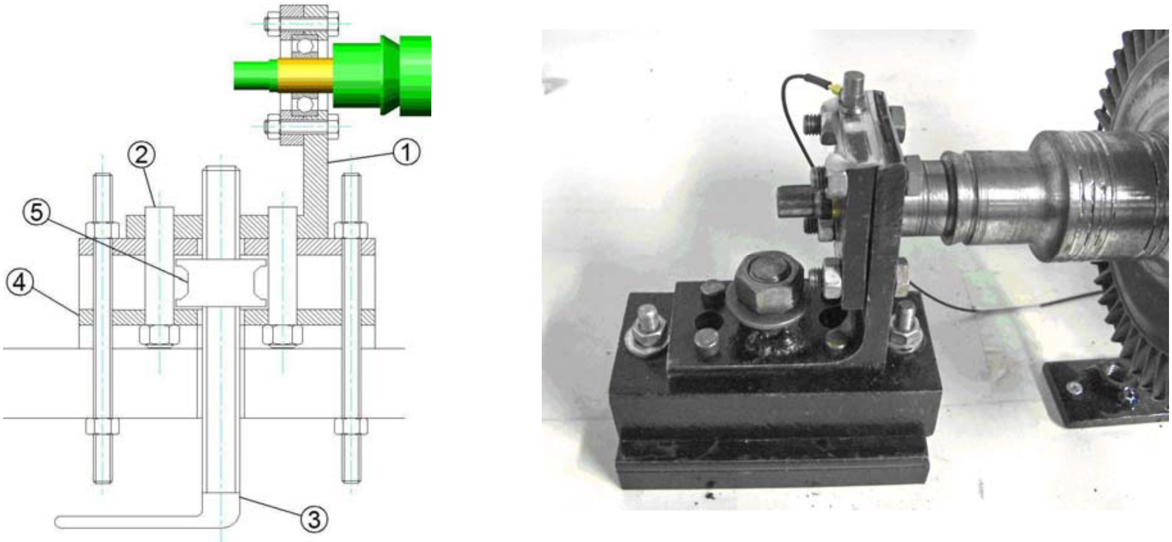


FIGURE 5. The seating assembly

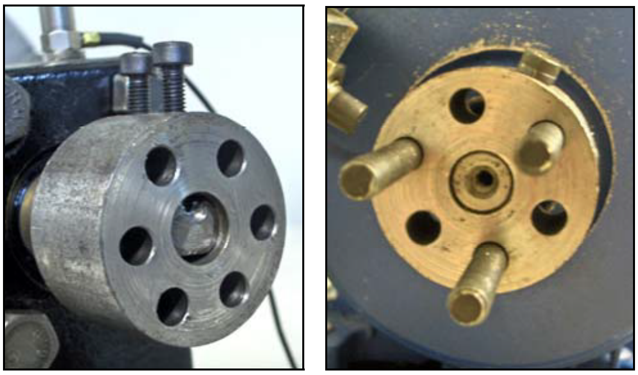


FIGURE 6. Coupling parts

The motor selected is a 3-phase induction electromotor. The electromotor has a rated power value of 0.18kW, it is connected with a 220V supply power, and the value of the field current is 0.9A with a frequency of 50Hz. Power factor value is cosφ=0.8. The rated speed is 2890rpm and the slide value is about s=5%. The electromotor is driven by an inverter, allowing the control of the shaft's angular velocity. The inverter has a frequency width of 5 to 50 Hz. An inductive sensor is used for the angular velocity measurement and is placed close to the coupling as seen Figure 7. The sensor signal is led to a frequency voltage converter and the angular velocity is presented in a LED display either in rpm or Hz units.

FIGURE 7. Coupling and inductive sensor

In order to measure vibration level, two accelerometers are used, one for each bearing. The accelerometer placed at A position (Figure 3) measures the vibrations for the turbine plane and the accelerometer placed at B position measures the vibrations for the compressor plane. Each accelerometer is mounted on top of the bearing housing using an appropriate magnetic disk, while it is insulated in order to prevent the current's frequency value to pass through the accelerometers and to the vibration signals. Accelerometers' sensitivity is 0.313 pC/ms^{-2}. Each accelerometer is connected with a charge amplifier. The amplifier use low-pass filters in order to cut off high frequencies and consequently to avoid "aliasing" frequencies appearance during the signal processing. Upper and lower frequency cut off limits, as well as the accelerometers' sensitivities are the basic amplifier parameters and can easily be determined by the operator via switches at amplifier's front side. Signal output uses a BNC port at amplifier's back side.

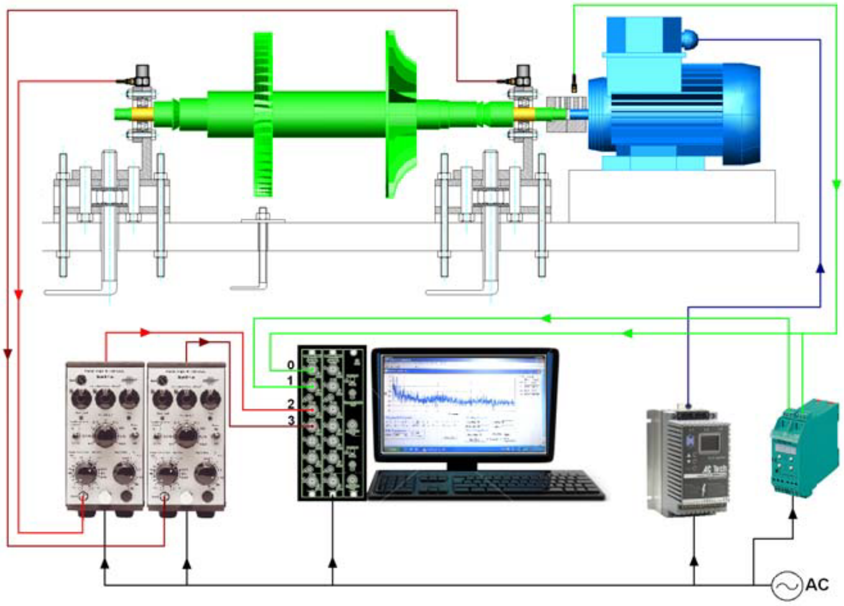


FIGURE 8. Test Rig Measuring Chain and Signals flow diagram

Measurements signals are led to the processing unit. The unit consists of a laptop, equipped with a data acquisition card having maximum total sampling frequency of 500 kHz and 12-bit accuracy. BNC ports are used as the card's interface with the signals. The signals acquired and analyzed are the angular velocity, inductive sensor pulse signal and the amplified vibration signals of the two accelerometers. The data processing is materialized using in-house software, which allows the analysis of acquired vibration signals using Fast Fourier Transformation (FFT) while the results in the form of power spectra and of power spectra differences from a reference condition are presented through a graphic interface environment. The parameters of the measurements analysis, such as the number of samples for FFT are set by the user, as is the range, gain and slope values for each channel. The measuring chain and the flow diagram of the measurements are presented in Figure 8.

Rotating parts of the test rig can be dangerous both for the operator and the students. So it was decided to place a plexiglas-made cover over the rotating parts. The cover is properly designed to allow the observation of the rotating shaft during operation, while a switch guarantees that the test rig wouldn't operate when the hood is open. The cover consists of an aluminum frame which upholds three 5mm plexiglas sheets. The finished test rig can be seen in Figure 9.

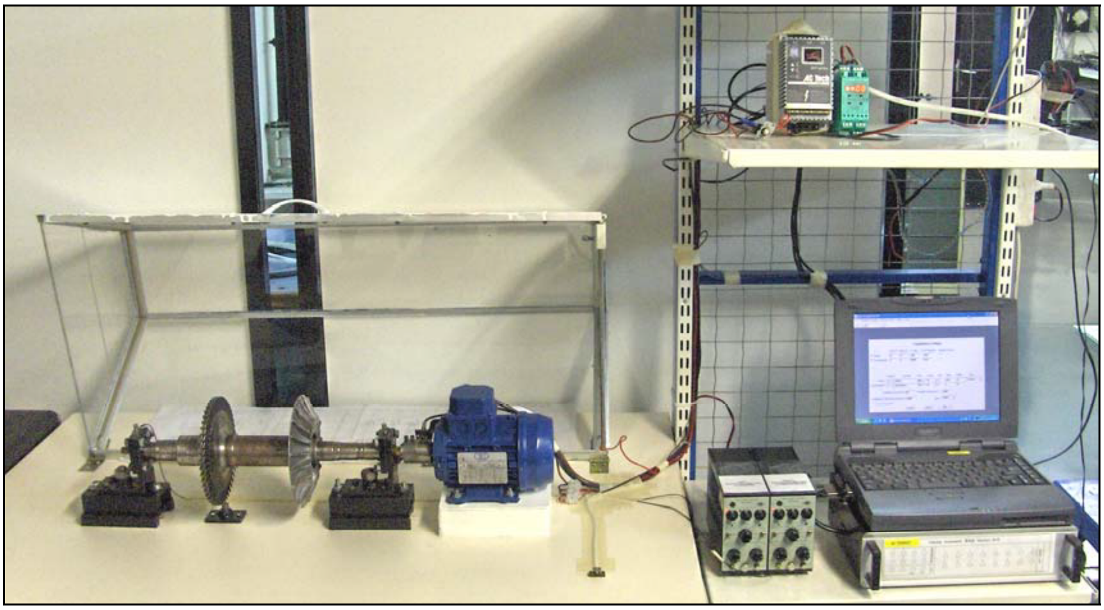


FIGURE 9. The test rig

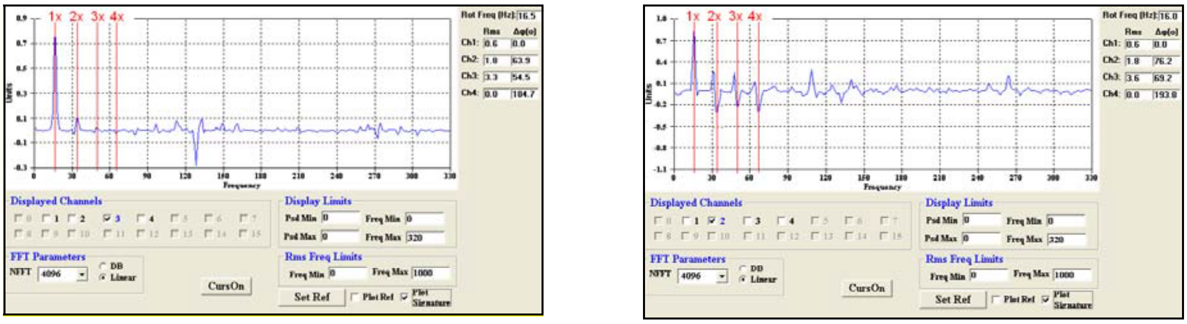
PROCEDURE AND RESULTS OF THE LABORATORY EXERCISE

A typical laboratory exercise routine, along with sample results is presented in order to verify that the test rig operates as expected and to highlight some of the educational objectives that can be accomplished. The primary objective of the laboratory exercise is for the students to understand the basic principles of diagnosis and to acquire knowledge and experience on the instrumentation, the measuring chain needed to perform vibration measurements and the analysis needed in order to gain diagnostic information. Also the importance of faults signatures should be clear, as should be the actual pattern of the simulated faults signatures. At the same time the students should understand the concept of balancing and its importance.

Prior to starting the laboratory exercise the measuring chain is presented and analyzed, while elements of rotor dynamics theory and vibration measurements, already covered during lectures, are discussed. Following, the "healthy" power spectra at the two planes (compressor and turbine) for a specific rotational speed are established. These power spectra are set as the reference condition in order to calculate power spectra differences when the faults will be implanted. This step highlights the basic principle of diagnosis: "A change on the condition of a mechanical system changes the physical quantities and parameters that describe the system's operation. Measuring the physical quantities change relative to their reference (healthy) values can lead to the finding of the root causes of the system's condition change". During the data acquisition the adjustments of the amplifier are examined, the students can directly see the effect of changing the settings to the power spectra and understand the logic and experience behind the selected setting.

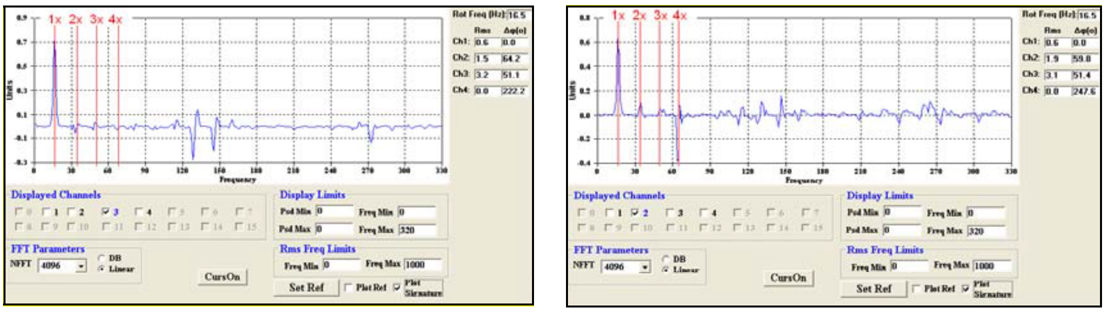
Having established the reference power spectra, the next step is the simulation of unbalance. Prior to fault simulation the teacher discusses with the students elements of rotor dynamics theory and describes mechanical and turbomachinery components faults that may result to unbalance. Such kinds of faults are fouling or erode during operation, foreign object damage and thermal effects, shaft sag, and rotor stator rubs [4].

Following the theoretical discussion, unbalance of the turbine and of the compressor is simulated. Firstly a mass of 24 gr is added at the compressor plane, at the position of 135^{o} counterclockwise and the power spectra difference from the reference ones for the two accelerometers are acquired. The power spectra differences, as seen in the GUI of the data acquisition program are presented in Figure 10.



(a) (b)

FIGURE 10. Power spectra differences (fault signature) for compressor unbalance calculated from (a) the compressor plane accelerometer and (b) the turbine plane accelerometer



(a) (b)

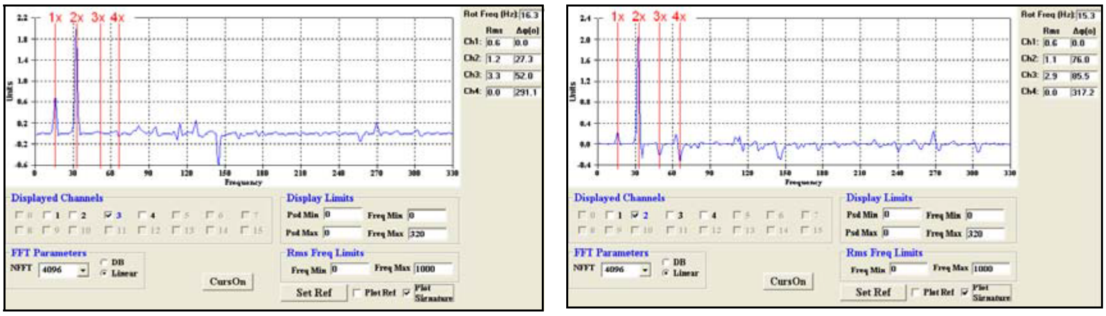
FIGURE 11. Power spectra differences (fault signature) for turbine unbalance calculated from (a) the compressor plane accelerometer and (b) the turbine plane accelerometer

The same routine (addition of a 24 gr mass at the position of 135^o counterclockwise) is repeated for simulating turbine unbalance and the corresponding power spectra differences are acquired (Figure 11).

From the results presented it is evident that the expected signature of unbalance is produced by the test rig. Specifically according to the literature [1,5] unbalance produce an increase of the vibration level at 1x of the shaft rotational speed frequency. In this way the students verify the theoretical knowledge through observation.

The test rig is stopped and the simulation of misalignment starts. Using the test rig as an example the teacher discuss about real word cases of misalignment and the operational problems that may arise such as excessive vibrations which will trip the engine and bearing failures. Also the most probable causes for misalignment such as improper mounting and expansion of the gear housing in case of power production gas turbines [1] are discussed, along with the parameters that define the misalignment tolerance, such as coupling length and type of bearings.

In order to simulate misalignment the shaft end connected to the motor via the coupling is moved vertically via the seating's mechanism described. The power spectra and the faults signatures are acquired and promptly analyzed. The results can be seen in Figure 12.



(a) (b)

FIGURE 12. Power spectra differences (fault signature) for misalignment calculated from (a) the compressor plane accelerometer and (b) the turbine plane accelerometer

From the results presented it is evident that the expected signature of misalignment is produced by the test rig. Specifically according to the literature [1,5] misalignment produce an increase of the vibration level at 2x of the shaft rotational speed frequency. In this way the students verify the theoretical knowledge through observation.

In order to further increase the insight of the students, homework is assigned based on the measurements taken during the unbalance simulation. The objective of the homework is to find the balancing masses and their radial position that should to be added at the compressor and turbine plane in order to balance the shaft. The method that has to be implemented is the standard two-plane influence coefficient method [6].

EDUCATIONAL ASPECTS

The test rig developed during a diploma thesis by an Engineering student is currently used for supporting courses on gas turbine diagnostic methods. In this context it has educational value for both the student that designed and constructed the test rig and for the students that use it.

As a laboratory exercise, as discussed above, it teaches in an experiential way the basic principle of diagnosis. It enhances the understanding of concepts, such as vibration level, power spectrum and faults signature, which are of paramount importance for applying diagnostic methods. At the same time the students learn how to set up a measuring chain for monitoring vibrations of a rotating system and they use the knowledge gained to answer to a real life problem (shaft balancing). Another important aspect of the exercise is the verification of the theoretical knowledge through observation in real time and in a controllable environment.

Concerning the student that designed and participated in the construction of the test rig, the procedure has several educational aspects. Execution of a project of this type requires the continual avocation with the subject and the student works within the Laboratory facilities cooperating with technicians and with the supervisor, thus emulating real working environment conditions. The student worked in the workshop and used the lab infrastructure, including tools, materials, electronic equipment and instrumentation, thus gaining experience in many engineering disciplines. The student was encouraged to take initiatives for solving the engineering problems that arose during the project, while he was responsible for contacting vendors and manufacturers, gaining experience to the field of project planning and enhancing his decision making abilities.

The fact that the test rig has been developed by a diploma thesis project has its unique merit in the educational process. Specifically it gives students confidence for their knowledge and motivate them to pursuit similar multidisciplinary projects, as for example the building of a contra-rotating compressor facility that has been assigned to two students the years following this project and was concluded by them [7,8] .

SUMMARY

The process for designing and setting up a mechanical fault simulation test rig has been described. The design process has been described, starting with the initial requirements and going through specific choices for the final layout. The way choices were made was presented, on the basis of serving in the best way the educational purpose that the test rig is mainly destined for. The test procedure and sample results have been presented, verifying the successful operation of the test rig and confirming the effectiveness of the design choices. The education aspects and expected benefits, for either students involved projects for setting up an installation, or students using the test rig as a support to the diagnostic related courses were highlighted.

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