COBEM 2023 27TH INTERNATIONAL CONGRESS OF MECHANICAL ENGINEERING

ABCM Brazilian Society of Mechanical Sciences and Engineering
27th ABCM International Congress of Mechanical Engineering
December 04-08, 2023, Florianópolis-SC, Brazil

COB-2023-0094
HOW TO EFFECTTIVELY USE OMA (OPERATIONAL MODAL ANALYSIS) FOR TURBOMACHINERY

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Abstract. *Challenging applications within the industry of high-performance centrifugal compressors, such CO2 reinjection Turbocompressor on "offshore" platforms, have been frequent. These machines work with high pressures and have been forcing the equipment manufacturer (OEM) as well and users to develop more sophisticated technologies in manufacturing and increasingly complex acceptance tests. The frontier of manufacturers knowledge has been challenged with high density gases, which add uncertainties in machines design, regard to their rotodynamic stability. The number of end users who require stability verification has grown. There is yet subsidiary risk of unbalance sensitivity increases on this turbomachinery. This risk is referred as: Rotodynamic Design Uncertainties. Although one machine like that can operate satisfactorily on its OEM test foundation, there is a risk that this machine will become unable to operate in an "offshore" plant. Aware of this little-recognized reality, we are developing research to present a treatment for this design anomaly: Rotodynamic Design Uncertainties. We are proposing small modifications to the ADX vibration monitoring system, developed by (LEDAV/COPPE/UFRJ), which will allow a data set project to be carried out in PETROBRAS/REFAP and subsequent analysis of the results in the LEDAV environment, using the ARTEMIS System.*
Keywords: *OMA, Turbomachinery, Compressor, applicability, effectively*.

1. INTRODUCTION

Challenging applications in the high-performance centrifugal compressor oil and gas industry, such as natural gas and CO2 reinjection Turbomachinery on offshore platforms, have become increasingly common. These machines operate at pressure levels of up to 600 atmospheres, which has pushed both the original equipment manufacturers (OEMs) and oil companies, who are the users of this equipment, to develop more sophisticated technologies in manufacturing processes and complex acceptance tests. The aim is to meet service demands and ensure smooth integration into production.

Manufacturers' knowledge frontiers (OEMs) have been challenged by using gases with increasingly high densities, such as CO2. These gases introduce significant uncertainties in the design of these machines, particularly concerning their rotodynamic stability according to API 617 Standard. As a result, the number of end users requesting stability verification tests to validate the rotodynamic model has increased. These tests aim to reduce the risks of equipment rotodynamic instability (whip) during their operational lifespan on offshore platforms.

In addition to the mentioned scenarios, there is another subsidiary risk that is not immediately apparent but further adds to the risks described above. This risk involves the sudden emergence of new natural frequencies within the operating range of these machines, accompanied by low damping factors. Such occurrences can happen during full load or partial load operations.

This risk, seemingly not well understood by users or machinery manufacturers, has resulted in significant production losses reported by several oil companies. These losses manifest as turbo machines that cannot be balanced, even in the work cradle. It should be noted that these same machines undergo thorough testing approval during factory acceptance tests conducted by the buyers and users (Running Test API 617).

While the approval obtained during factory tests is highly desirable, it does not guarantee proper functioning when the machines are sent to their operating platforms. This risk is commonly referred to as "rotodynamic design uncertainties".

To illustrate this issue, consider a real case where a multinational oil company had to decommission its semi-submersible production platform because it was incapable of producing an appropriate amount of oil. This platform operated in a non-surging oil field and relied on a gas-lift compressor since water injection was not feasible. However, this compressor failed to operate satisfactorily due to excessive vibration. The oil company hired an international consultant to address the problem.

Despite major design changes recommended by the consultant, implemented by the OEM, and subsequently subjected to an acceptance test conducted by the OEM, the compressor continued to vibrate in the same manner after reinstallation on the platform. Consequently, gas-lift operations at the field became unfeasible. The consultant hired to solve the problem later declared that the compressor was inherently "unbalanced" and difficult to rectify.

This risk is associated with a paradox: an excessive increase in rotor damping (sealing and bearings) to prevent rotodynamic instability can lead to rotor locking and an unexpected reduction in the system's damping capacity. It is important to note that the internal gas sealing of these compressors alone cannot stiffen the system and increase the rigidity of the rotor suspension by compressing the gas film, as the gas is compressible.

The increase in rotor damping mentioned earlier can diminish the machine's ability to dissipate unbalanced energy, ultimately leading the force to integrally be transmitted to support structure which has a low damping factor and consequently can greatly increase machine vibration.

These low damping factors of the support system (structure) are responsible for high levels of vibration associated with the first harmonic and have enough potential to shut-off machine operation in the field (unbalance).

We can also refer to this same misunderstood issue, making use of the point of view expressed by an important “player” in the OEM universe. In the article: Giuseppe Vannini et all, 46 th TURBOMACHINERY SYMPOSIA HOUSTON TEXAS December 11-14, 2017 a world-renowned OEM's experienced consultant suggests the possibility of operating a turbomachine above its third critical. It is important to highlight that manufacturers are always interested in building ever smaller machines, lighters, with higher rotations, with greater useful power and, therefore, with lower manufacturing costs, aiming at greater productivity in the equipment process sales.

Although a machine like the one proposed in the article referred to above, can operate satisfactorily on its test bench (with an inertial foundation), meeting the requirements of API 617 Standard (amplification factor less than 2.5 in all critics within the operating range), there is a very high risk that the same machine will become unable to operate satisfactorily in an “off-shore” plant, due to rotodynamic design uncertainties.

This uncertainty strongly resides in the fact that the roto-dynamic parameters (natural frequency and damping factor of the rotor) will be unpredictable in a work cradle that is unable to guarantee good elastic properties, making it not possible to estimate transmitted vibration a priori. The elastic characteristics of the support of this machine, which is represented by its FRF (frequency response function), and consequently its vibration performance in the field, are not defined. That is a turbomachine impregnated with a great rotodynamic design uncertainty that is many times greater than would be expected from another machine operating between the first and second crits, as is normally recommended by consultants (expertise). It is important to point out that two identical platforms, built under the same project, will have different FRFs at their anchor points.

This statement becomes clear when we consider that the rigidity of these support structures is unknown in the initial phases of the project, only being defined at the end of the execution of the physical project "as built", which will be strongly impacted by the assembly methods, tolerance of manufacture of the thicknesses of the metallic profiles used, construction welds and decisions out of project that are inherent to large structures assembly process , ... (uncertainties of rotodynamic design). We also point out that at the end of the physical design of these platforms, their machines will already have been tested and delivered to the site.

In API 617 Standard, this subject is dealt with inadequately and not very objectively, compromising and making the equipment buyer responsible if the stiffness of the support structure does not exceed 3.5 times the stiffness of the bearing, which is impossible to be defined/assured, in the design phase associated with turbomachinery purchase.

Even knowing all these particularities and uncertainties described above, we found that the number of turbo-machinery operators interested in carrying out test verification of machineries rotodynamic stability has been growing. These tests allow verifying the accuracy of the rotodynamic modeling and help to mitigate the risk of “whip”.

Aware of this reality, that is little recognized by many international players, we are developing a project in which we intend to present a treatment for this design anomaly, known as rotodynamic design uncertainties. For this purpose, small modifications are proposed in the ADX vibration monitoring system, developed by the Laboratory of Tests and Analysis of Vibration of the UFRJ (LEDAV/COPPE/UFRJ), which allow the implementation of a data storage setup project in one of PETROBRAS Refineries (REFAP) and subsequent analysis of the results in the laboratory, using the ARTEMIS code.

In case of successful experience, LEDAV will be able to offer PETROBRAS a service/equipment that will raise the roto-dynamic parameters (rotor natural frequency and its damping factor), for both factory and field tests. We know that Petrobras lacks this need and today does not have this “design facility”, therefore, it willing for this service/facility.

Negotiations for data storage at REFAP have already been carried out by Dr. Luiz Vaz, coordinator of this project, and are already in an advanced stage of implementation.

After this brief introduction, it becomes necessary to discuss which are the best options for planning and executing acceptance tests for these machines both in the factory and in the field.

2. CHOICE OF METHOD FOR TESTING TURBO MACHINERY

There are many tests that a turbo machinery needs attend to be approved before it can be considered qualified for service and incorporated into the production portfolio of an industrial plant.

The focus of this research is the effectiveness of tests related to the rotodynamic stability of these turbo machinery.

Pursuing this focus and being aware of all the afore mentioned particularities, it is necessary to find the method that best meets this reality and can be used during the rotodynamic stability tests that need to be witnessed by the buyer.

The choice is complex and will be unable to ensure, with 100% certainty, the proper functioning of this machine in its work cradle. The most frequently recommended proposition has been the EMA technique (Experimental Modal Analysis), which has some advantages and some disadvantages, such as:

. It requires the temporary installation of an electromagnetic exciter, "shaker," mounted on the opposite side of the driver to asynchronously excite the modes of interest. This creates additional costs and difficulties for the test, as well as slightly altering the rotodynamic of the rotor.

. It depicts the machine's dynamics on an effectively designed inertial support specified by the manufacturer for this purpose (factory testing at the OEM). This dynamic cannot be reproduced on the machine's target platform (workstation), which is an extremely more flexible, complex, and unpredictable structure.

The other possible choice would be the experimental technique OMA (Operational Modal Analysis) frequently used in modal analysis of structures, which also presents functional advantages and disadvantages, such as:

. It does not require an external source since the excitation will come from the aerodynamic noise produced by the gas inside the machine.

. It can easily be reproduced at field where the facilities are conveniently prepared for this experience, such as in REFAP.

. It depicts the machine's dynamics in its current support: both in the manufacturer's facilities and in its real support when the machine is undergoing to in-situ field test.

This method does not seem to be widely used currently, according to the findings from published papers in the last four years. This research aims to focus on the reasons associated with the low interest demonstrated in this method.

The main disadvantage is the uncertainty associated with the resolution and effectiveness of the test results, as will be shown in this research. This uncertainty is related to the low level of internal gas noise produced inside the compressors, which is insignificant in most cases and greatly hampers the method's effectiveness and applicability.

In the reference paper "Zag, L.O et al., IFToMM 2018, MMS 61, Vol 2pp.460,2019," the results of applying the OMA technique in a field test of rotodynamic stability of a recycle compressor are shown. This article presents the results of a test conducted by the manufacturer, which guarantees the absence of rotodynamic instability "whip" in this machine. Although this is good, it can be considered frustrating as it does not reveal any sub-harmonic vibration manifestations in its cascade diagram, as shown below.

Figure-1 shows the cascade diagram of the tests conducted at the manufacturer's facilities. The (EMA) test had revealed a natural frequency of the rotor at 7900 rpm, and therefore, some vibrational manifestation in this rotor near 7900 rpm would be expected, caused by internal compressor noise.

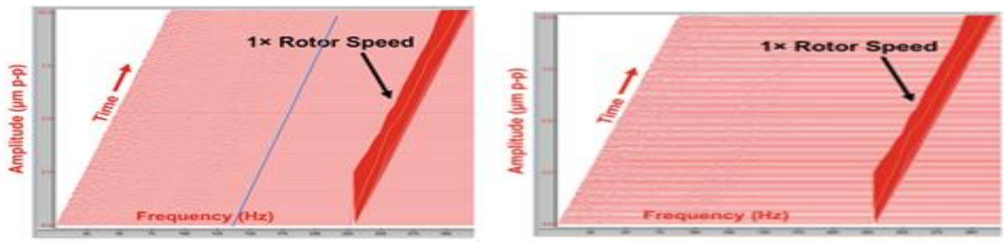

Figure 1. Cascade diagram showing the absence of instability.

This occurred because the aerodynamic gas noise inside this machine did not have enough energy to be captured by the displacement sensors used. Some articles, like this one, try to relate the aerodynamic noise inside the machines to the density of the compressed gas, and this seems to be a big misconception, as we intend to discuss later in this research.

. *Let's now interrupt previous discussion on test strategy, to take a closer look on experimental techniques: EMA/OMA*

In mechanics, structures vibrate on a time basis, and modal analysis is useful for identifying these same movements after decomposing them into their natural modes of vibration, which establishes a new coordinate system for treating the problem. The data is originally obtained in the time domain with the help of vibration sensors and then numerically processed to bring the problem into the frequency domain, where its "fingerprint" can be characterized (natural frequencies, vibration modes of the system, and damping factors).

In Figure 2, a simple structure presents an infinite number of vibration modes that become more complicated as their natural frequencies increase, requiring more and more energy to vibrate at higher critical frequencies.

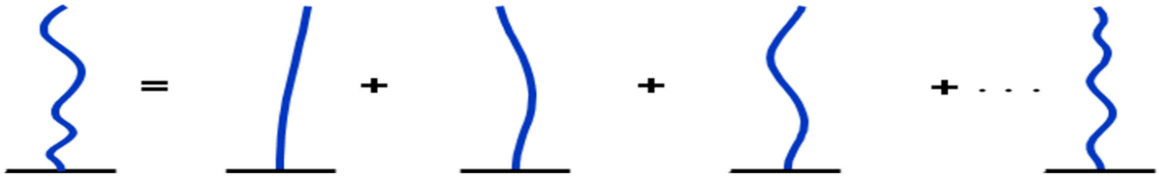

Figure 2. Vibration Mode Decomposition of an EMA Beam

At Experimental Modal Analysis EMA, is possible to estimate the vibration modes of a simple structure in laboratories by using an external excitation source and sensors (accelerometers). For example, the external excitation can be done using impact hammers or external exciters on a freely suspended structure in a laboratory, and the response of the structure is measured with the help of accelerometers. In large structures, variable frequency exciters of large size can also be used. The signals collected from the structure by the accelerometers are transformed into the frequency domain (FFT), resulting in input (force) and output (acceleration/displacement) frequency spectra. Dividing the output signal (displacement) by the input signal (force) yields the Frequency Response Function (FRF). For the FRF to be accurate, it is important that only the excitation force is acting on the structure at the time of measurement; otherwise, we will have a "polluted" FRF.

The Coherence Function can be used for evaluating the quality of the FRF. It is important to note that EMA testing requires a predominantly deterministic mathematical approach. Referring to the limitations of this method, it should be noted that in some cases, there may be difficulties in presenting the expected results, and it is interesting to list and explain the reasons that require the use of other complementary analysis tools. We can summarize the limitations as follows:

1- Inability to apply external excitation during the test (such as in the case of field turbomachinery tests). If a concrete, objective, and effective excitation is not possible, the FRF will be compromised, as well as the identification of natural vibration frequencies and structural damping factors.
2- Improper boundary conditions that do not accurately represent the real support conditions can significantly affect the results and alter the obtained modal parameters.
3- Improper excitation levels during testing can similarly compromise the results, falsifying and affecting the modal parameters.
4- In the case of large and symmetric structures, difficulties may arise in exciting the structure to reveal the modes.

As suggested in the ARTEMIS platform, the application of EMA in certain mechanical structures is often difficult or nearly impossible, such as:

1- Large structures, such as bridges, towers, platforms, etc.
2- Naval structures, including ships, submarines, offshore platforms, etc.

All these structures are subjected to uncontrollable local external forces that cannot be removed, measured, or experimentally applied, and they are responsible for the vibrations. These vibrations practically prevent the application of EMA and compromise the resulting FRF. In such cases, the only possible alternative for obtaining the rotodynamic parameters is to use the prevailing excitation forces, which are unknown and cannot be simulated.

At Operational Modal Analysis technology OMA, these readings of the prevailing excitation forces are obtained by strategically placing sensors in such a way that they can emulate the natural vibration modes that are generated and expected based on the conducted dynamic modeling studies. This procedure is often applied in a stochastic process and can identify the modal parameters of the structure (natural frequencies, natural vibration modes, and damping factors). These results can be obtained using different types of sensors.

The theoretical framework required for an OMA approach is different from the deterministic approach used in EMA, as it requires a statistical treatment where the excitation forces are unknown and will be inferred from stochastic processing.

This stochastic treatment assumes that the excitation is produced by white noise (equal energy level at all frequencies of interest), thus equally exciting all the modes of interest. In practice, this is not true because there are frequency ranges that contain more energy than others. To compensate for these excitation force modeling inconsistencies, the stochastic processing considers that the white noise is modified by the insertion of a linear filter that gives it a realistic aspect (true distribution of the applied force). Therefore, the structural response to the assumed loading is a combination of the dynamics of the structure itself with the filtered excitation, which represents a loading of unknown forces that cannot be measured. As shown in Figure 3

We can enumerate the practical implications of these considerations as follows: In practice, the application of OMA assumes a broadband frequency signal that covers all the modes of interest (especially the first mode). If there is inconsistent excitation of the modes, the result will be less representative of reality. Narrowband excitations are acceptable if they result from broadband excitations.

Another practical implication of the above approach assumes that some of the modes displayed in the system response (frequency peaks) may not originate from the structure itself but from an improperly filtered input mode of the system.

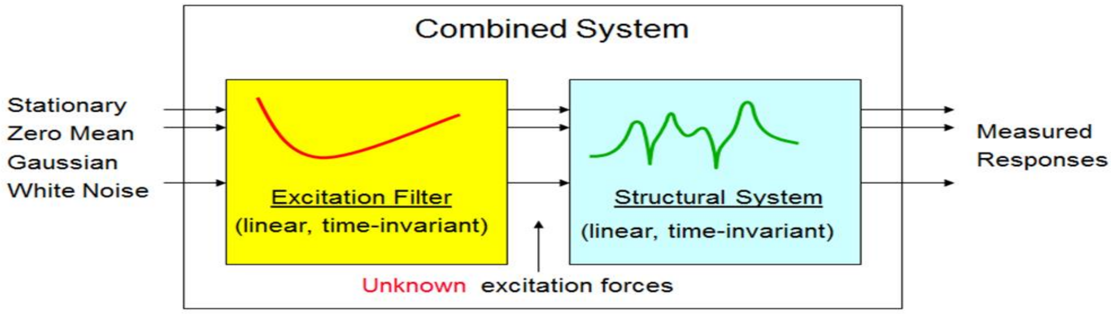

Figure 3. Operational Modal Analysis Processing Strategy

In practice, some of the peaks in the spectrum may not be related to the structure itself but rather to the input signal, which in the case of turbomachinery could be associated with its rotation and harmonics.

As a measurement procedure, sensors can be arranged in their modal form and a measurement window (time window) can be performed, or the sensor can be moved to various predefined positions and multiple measurement processes can be conducted. The positioning of accelerometers established for measurement is referred to as the "TEST SET-UP." The measurements performed with that configuration are called "DATA-SET." Each new measurement on the same set-up will be a new dataset, which can have a duration as long as desired.

In some cases, sensors are permanently installed in structures, such as continuous monitoring of turbomachinery, and in these cases, we have a unique set-up. In other cases, a set of sensors can move along a structure (a monitored vehicle crossing a road/bridge), and we have a case of multiple set-ups. It can be concluded that OMA should not be seen as a substitute for EMA but rather as an alternative technology in cases where input forces cannot be controlled or even measured (as in the case of turbomachinery).

In turbomachinery case, contrary to what is prescribed in the current technical literature, we propose that both EMA and OMA are possible tools for modal evaluation in factory acceptance tests. However, in confirmation tests in the field, after the start of plant operation, only OMA has the potential to identify the actual rotodynamic parameters of the equipment in its working environment in the field.

. *Returning to the discussion on testing strategy, we propose that:*

To achieve more robust noise levels capable of exciting the first vibration mode during tests, efforts should be made to produce consistent internal noise with a broad frequency band of excitation. This can be achieved by placing the compressors, for example, close to their "surge" condition, where aerodynamic noises consistent to the physical phenomenon associated with the gas boundary layer separation at the last impeller vanes (leading edge) of each stage (snowball effect). As presented in the reference: Baldassarre, L. et al., 44 Turbomachinery Symposium, Houston, Texas, September 14-17, 2015.

Another possible test filter window is obtained by putting the machine in the "Stonewall" condition. In this operational region of the machine, the induced noises can also be significant and consistent with gas operation near its sonic velocity inside the machine, "first rotor tip speed".

The frequently demanded factory test (EMA) allows comparing the measured modal parameters with the inertial-based, with predicted model proposed by the manufacturer, which provides credibility and confidence in the OEM's ability to predict the dynamic behavior of the machine in an inertial-based condition (completely disconnected from the field reality). It is always important to remember that according to API-617, if the dynamic stiffness of the machine foundation (support structure) is less than 3.5 times the bearing stiffness, the rotodynamic study is compromised, and in these cases, significant changes in machine's first critical parameters can occur.

Within specialized circles, highly knowledgeable international consultants in turbomachinery rotodynamic, may even consider the hypothesis of a significant reduction in vibration damping capacity of this machine caused by excessive dynamic flexibility of the platform structure.

3. OPERATIONAL MODAL ANALYSIS APPLIED ON A TURBOMACHINERY AT REFAP/PETROBRAS

The proposed implementation of this test on a refinery real existing machine, in its own working environment, aims to discuss the most critical issue of the applicability of OMA in turbomachinery. The proposal is related to finding an answer to the dilemma posed by the specialized technical literature, where a question needs to be answered:
Can OMA be robustly applied on turbomachinery? In which cases can we be confident in obtaining positive results?

This experiment aims to demonstrate that it is entirely possible to measure critical rotodynamic parameters (natural frequency and damping factor of the first mode) in a steam turbine. It also intends to explain why is this possible in a scenario where many expect OMA not to be applicable on turbomachinery, according to literature associated with the ARTEMIS platform itself.

It is also important to emphasize that the rotodynamic parameters are closely linked to mandatory issues for turbomachinery proper operation such as: rotodynamic stability and adherence to a reliable balancing, which are The two most significant failure cause (malfunction) in offshore installations, after they have already been approved in (OEM) factory acceptance tests, according to API-617 Standard.

This study second phase aims to show that OMA technology can also be applied to a centrifugal process compressor, which is driven by this steam turbine, without obtaining good results, which seems to be the most evidenced perspective in the literature, which defends the inadequacy of OMA for turbomachinery. In this opportunity, the true reasons that led OMA technology to a poor performance in determining the rotodynamic parameters will also be discussed.

In the third phase of this research, will be shown that the same compressor can yield positive results in determining the rotodynamic parameters if relevant measures are implemented before conducting the measurements.

In this work final phases (future), we will discuss how the rotodynamic parameters of an electric motor/gas turbine rotor can be determined, and new measurements will be proposed to document this last phase of the research.

. *First faze applied in steam turbine:*

The machine chosen to demonstrate this reality was the wet gas compressor turbine of the UFCC unit at REFAP (Alberto Pasqualini Refinery of Petrobras, located in Canoas, Rio Grande do Sul). This machine has been operating in a reliable way, for more than twenty years and was used as a laboratory to demonstrate the correct use of the discussed test methodology for applying OMA technique in turbomachinery. The time series was recorded in November 2022 and is presented in Figure 4, consisting of approximately 8,000,000 data points collected from turbine displacement sensors. This series was processed using the ARTEMIS code. The time series of the turbine sensors are presented in Figure 4.

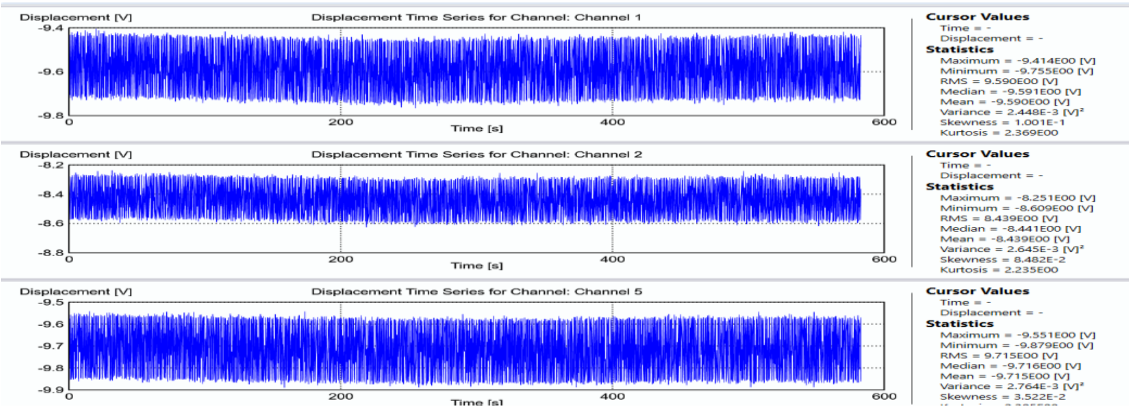

Figure 4. Time series captured by the Steam Turbine sensors

The power spectrum density of the signals from the turbine sensors is provided by ARTEMIS program and is presented below in Figure 5, where we can identify the operational parameters of the system.

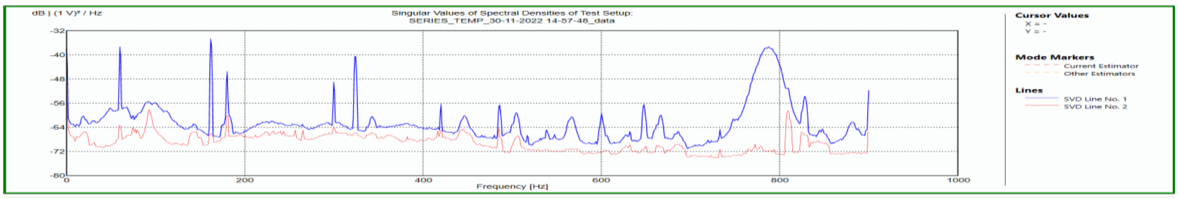

Figure 5. Power spectral density of the signal from the Steam Turbine sensors after processing in ARTEMIS

The signals processing in ARTEMIS platform allows a clear characterization of the natural frequency and damping factor of the steam turbine rotor in its actual suspension. It apply a statistical treatment to learning the real rotodynamic parameters of the machine (statistical process to learning machine properties - knowledge).

The first critical mode obtained is associated with the behavior of the equivalent suspension in its first vibration mode. It is the result of the rotor/suspension interaction, which reflects the equivalent stiffness and damping of the linear system representing the rotor support interacting with the flexible structure of the platform (rotor-structure interaction).

A relevant question arises: Is it necessary to identify the operational parameters for all rotor critical modes, or can we be satisfied only with the parameters of the first critical mode?

The obtained results (rotodynamic parameters) will be different from those obtained in the factory test. The difference in these results will be greater as the flexibility of the machine's support structure increases. This research will focus solely on the reliable detection of rotodynamic parameters in the field, which seems to be a matter of uncertainty in the literature controversy.

The same results produced here can be perfectly replicated in any steam turbine acceptance tests conducted at the manufacturer's facilities when purchasing a new machine. It should also be noted that the frequent use of EMA in these factory acceptance tests does not impede the use of OMA technology, as a parallel recording can be performed at the same time.

Figure 6 shows the appearance of the measured time signal from the steam turbine's vibration sensors when filtered with a band-pass filter tuned to the turbine's natural frequency.

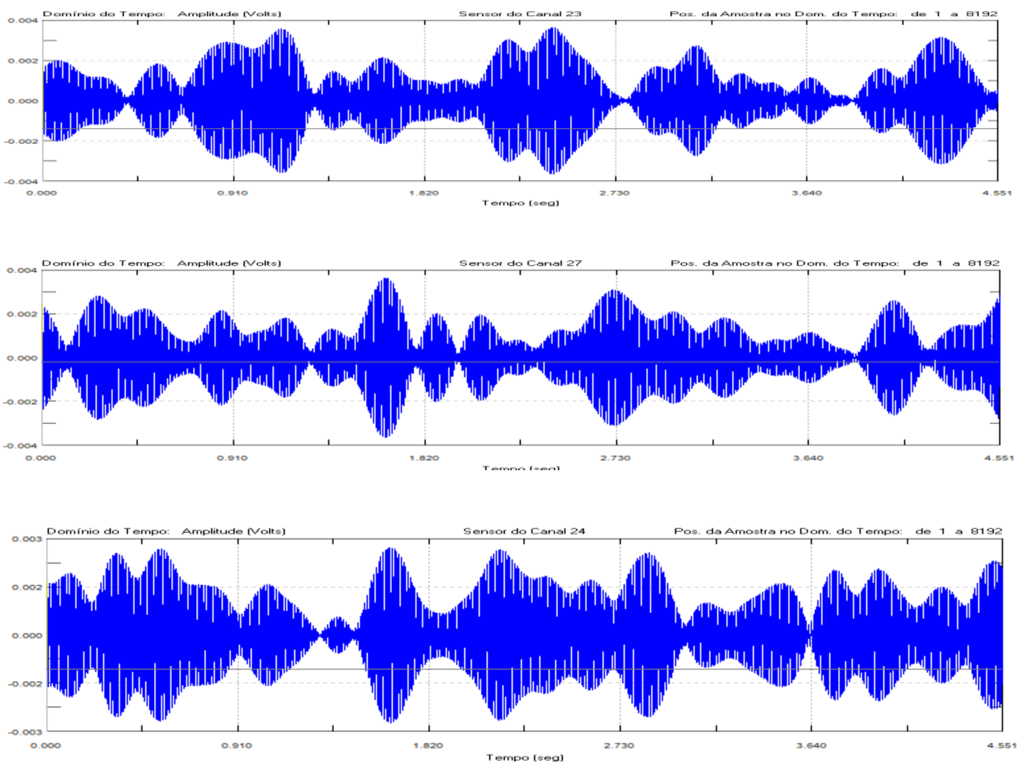

Figure 6. Turbine sensors time signals, after being filtered at band-pass filter tuned at natural frequency.

The "bubbles" captured in the signals from the turbine's vibration sensors can only be revealed in the filtered signal and are generated by the control system's action on the turbine inlet valves (partializing valves), under turbine governor control.

When partialized valves are open, the turbine steam flow reaches and exceeds its supersonic regime, generating a broadband noise with high energy that can excite the natural frequencies of the turbine rotor.

The power spectrum density from turbine sensors, processed in the ARTEMIS code, can certainly identify and confirm the values of the rotor's first natural frequency and the damping factor of the equivalent system.

- *Second faze applied in wet gas compressor:*

In the second phase of this research, the time series of the compressor was acquired on 27/08/23 and consists of data set of 4 signals, each one with approximately 2,000,000 points recorded from each of the compressor displacement sensors driven by the afore mentioned compressor. The time series from each of the compressor sensors is presented in Figure 7.

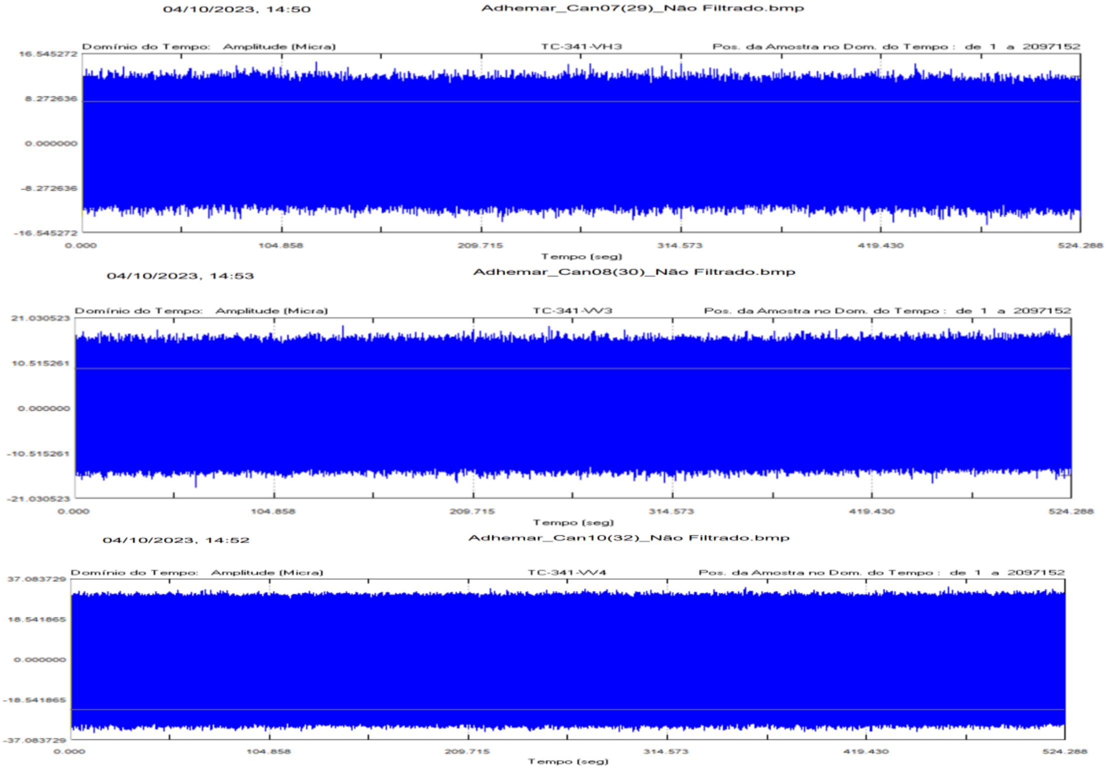

Figure 7 – Time series captured by the compressor sensors.

The deterministic processing of signals in the frequency domain, as shown in the cascade diagram in Figure 8, does not allow for the characterization of the first natural frequency of the compressor rotor in its actual suspension, as has been claimed in the specialized literature. The first critical mode of the compressor simply cannot be revealed due to the lack of noise that could sufficiently and appropriately excite the rotor with the necessary energy and frequency bandwidth. The pick observed close to natural frequency is 60 Hz electrical noise.

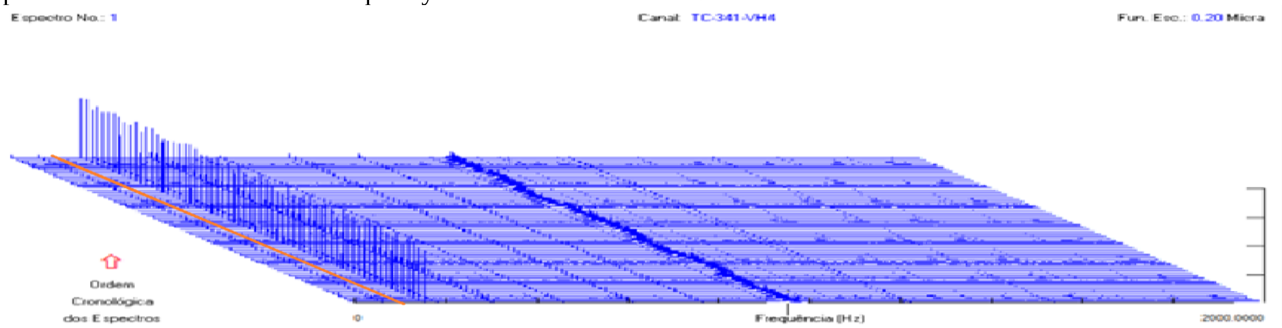

Figure 8. Cascade diagram of one of the signals collected from the gas compressor.

The normal operation of gas compressor flow within its normal operating range, generates very low aerodynamic noise despite of gas density. This noise does not have sufficient energy to provide consistent excitation with enough energy and a wide frequency bandwidth that is compatible with the expected methodology of the OMA technique.

The same time series from each of the compressor sensors are presented in Figure 9 after being filtered with a band-pass filter around the compressor's natural frequency. Natural frequency is not clear at that point once it is hidden in sensor electrical noise (by lack of energy), as discussed before. This seems to be the most evidenced perspective in the literature, which defends the inadequacy of OMA for turbomachinery.

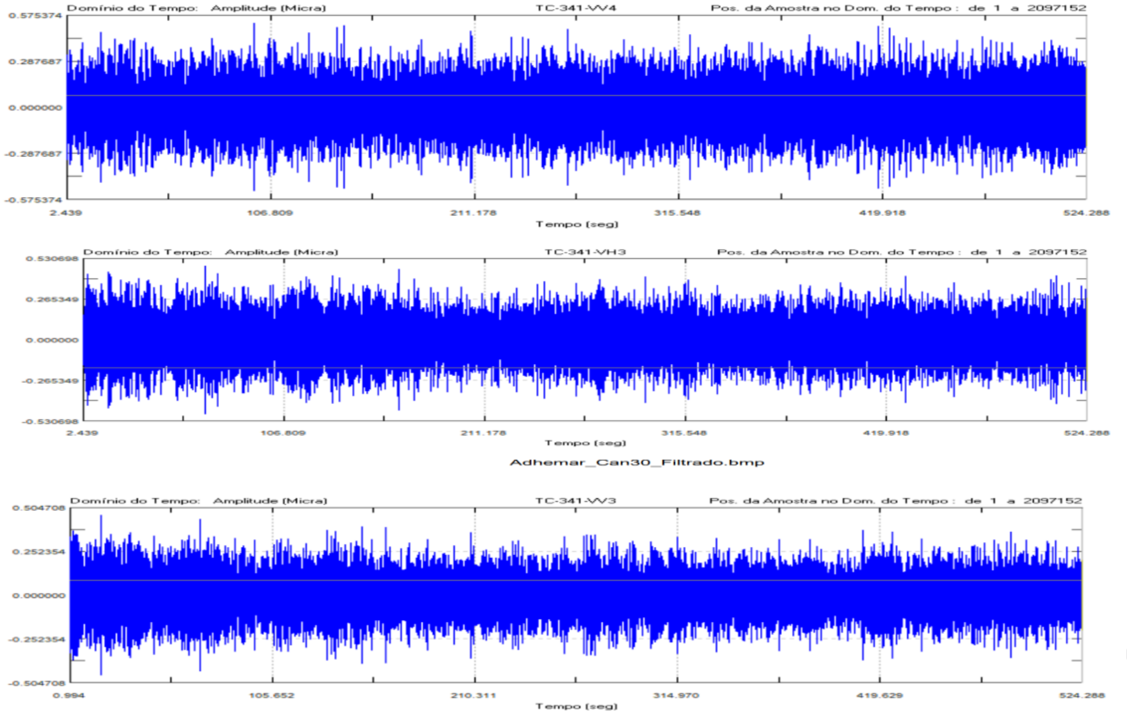

Figure 9. Gas compressor time series after being filtered with a band-pass filter at the machine's natural frequency.

To highlight the compressor behavior equivalent suspension, which is the output signal of the rotor/suspension interaction, we need to generate aerodynamic noise within the compressor that has sufficient energy to promote consistent excitation with enough energy and a wide frequency bandwidth, compatible with is expected from OMA methodology technique.

This noise can be generated in two different operational conditions:

a) By approaching the machine to its "surge" region. This operational condition establishes a pattern of highly noisy gas flow at the inlet of the last compressor impeller. It is compatible with the noise broadband frequency produced by the detachment of the gas boundary layer at the leading edge of the last impeller blades (impeller eye) in the last investigated rotor stage (snowball effect).

b) By approaching the machine to its "stonewall" operational region. This operational condition establishes a supersonic pattern at the first impeller gas flow in the first compression stage, which can be very noisily in the "stonewall" condition and is compatible with the expected noise within the OMA technology.

By driving the wet gas compressor of the UFCC unit at REFAP/Petrobras to a stonewall condition, it is possible to identify the behavior of its equivalent suspension (rotodynamic parameters) through the processing of the signals collected at the refinery, as shown in the figures below in Figures 10.

The rotodynamic parameters of the linear system, representing compressor rotor support, which is interacting with its flexible structural support, can now be identified with the help of ARTEMIS code, as shown in Figure 10 (rotor-structure interaction).

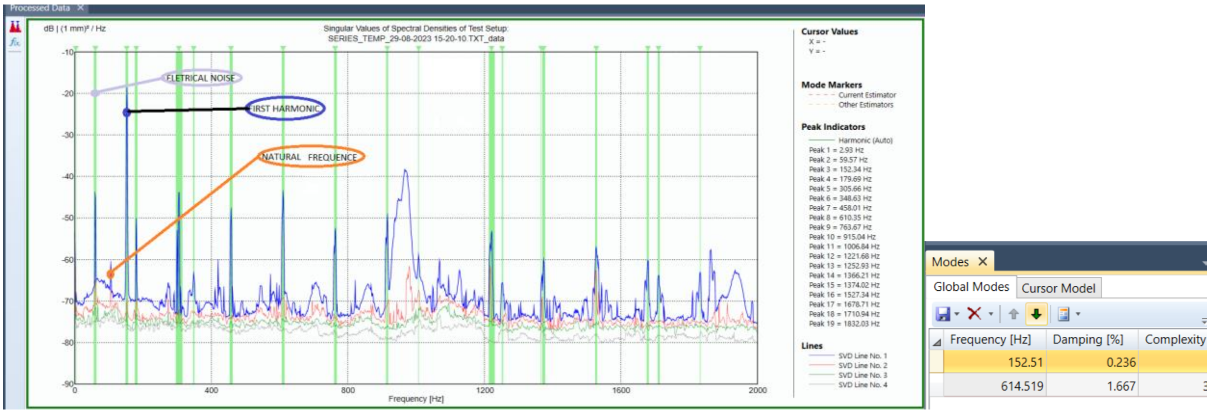

Figure 10. Power spectrum density of signal sensors (compressor near stone wall) processed by ARTEMIS

The obtained results (rotodynamic parameters) will differ from those obtained in the manufacturer running test. The difference in these results will be greater as the flexibility of the machine's support structure increases.

This research focuses solely on the reliable detection capability of the rotodynamic parameters in the field, which seems to be a subject of controversy in the literature.

The same results produced here can be perfectly generated in a machine acceptance test conducted at the manufacturer's facilities, during the purchase of a new machine. The frequent use of EMA in factory acceptance tests does not prevent the use of OMA in the same opportunity.

4. CONCLUSION

This research represents the initial part of a more complete investigation to provide a tool capable of identify the anomaly known as Rotordynamic Design Uncertainties

It can be concluded at this point that OMA technology is perfectly applicable in both factory and field tests of high performance turbo compressors, such as reinjection compressors used in offshore platforms in the pre-salt fields.

The comparison of results obtained in factory and field tests allows for a possible diagnosis of the inadequacy of the support structure for the service and will guide changes in the machine design to adjust the compressor's damping factors to its new reality (tunning excessive damping).

5. FURTURE RESEARCH PROPOSALS

In the work forth phase (future), we will discuss how the rotodynamic parameters of an electric motor and a gas turbines rotor can be determined, and a new measurement will be proposed to document this last phase of the research.

The possibility of applying OMA technology to high-power electric motors and gas turbines should be investigated, along with discussing possible necessary adaptations. Reference can also be made to a real case where the ammonia motor driven compressor at FAFEN/SERGIPE experienced a serious problem of sensitivity to unbalance caused by the flexibility of its support structure (mezzanine). Reference: ALLAIRE, P.E; ROCKWEL, D.R; CASTILHO, A; 2005, DINAME, Ouro Preto, Minas Geria Brazil, March. 28.

In this specific case, the problem was solved by modifying machine supporting mezzanine design to detune one of the natural modes of vibration at onshore support structure.

This solution would hardly be implementable in an offshore structure, and the solution would require changes in the machine design, especially in the bearing and compressor sealing designs (internal and external).

6. REFERENCES

- ARTEMIS.
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7. RESPONSIBILITY NOTICE