ASIA TURBOMACHINERY & PUMP SYMPOSIUM

ROTORDYNAMIC ANALYSIS AND TESTING ON AN INTEGRALLY-GEARED COMPRESSOR SERVING A HYDROGEN RECOVERY APPLICATION

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ABSTRACT

The paper examines the case of a rotordynamic analysis and subsequent rotodynamic testing conducted during the pre-order phase for a five-stage integrally-geared centrifugal compressor, which today is operating in a refinery in western India. Performed in addition to the standard lateral vibration analysis, a rotordynamic testing method known as unbalance-response-verification-test was performed. This was done as a further means to secure the reliability of the compressor's rotor and bearing design, and the seal configuration (requested by the end user as double dry gas seal). The compressor is deployed within a Pressure Swing Adsorption (PSA) Tail Gas application where it supports in the recovery of hydrogen.

INTRODUCTION

When examining the innerworkings of their plant operations, few industries within the hydrocarbon sector require the level of availability, reliability and safety that refineries do. For their general operation, their various refining processes, and for specific equipment deployed therein, these value parameters are crucial for the plant's operator. This pertains particularly to the handling of the many challenging gases found in various refinery processes and, therefore, the reliability of compressor equipment designed for and used in these processes – regardless of whether the process uses centrifugal compression technologies such as inline or integrally geared, or positive displacement compression methods. In this paper, the authors will center their discussion around the aspect of compressor reliability and the design of an integrally-geared gas compressor deployed in the gasification unit in one of India's largest refineries.

In July 2002, the American Petroleum Institute (API) 617 considered integrally geared compressors in their 7^{th} edition, thereby extending the original scope of chapter 3. Today, integrally-geared centrifugal compressors (IGC) are widely used across the hydrocarbon industries and are also gaining growing acceptance for refinery applications, thus, reflecting the many safe and reliable installations around the globe, API Standard 617, Chapter 3 today describes integrally-geared equipment as "*helping assure users that IG compressors are properly specified, designed and tested"* (Beaty, Eisele, Maceyka et. al. 2000; American Petroleum Institute 2009). API 617, Chapter 3 includes – among many others – the gearing within the compressor, which is an essential and integral part of this compressor type, as well as the aspect of the overhung rotors, bearing design (which supports the dynamic behavior and helps prevent instabilities), while also defining seals; typically dry face seals are used, with the aim minimizing leakages (Patel and Struck 2017). Defined in API Chapter 1-2.6 is the lateral analysis during the design phase – it is intended to safeguard against excessive vibration levels, defining maximum permitted vibration amplitudes regarding the minimum design clearances (American Petroleum Institute 2009). In the latter section of this paper, the authors will return to this point and discuss it in the specific context of the refinery installation in western India.

That said, the end user requirements for some applications span beyond the requirements and guardrails set forth by API, thus requiring additional analysis and testing methods with either individual components and/or overall compressor design. In the realm of rotordynamic analysis and testing, the unbalance-response-verification-test by API 617 7^{th} edition Chapter 1- 2.6.3 provides one suitable avenue to help ensure these additional reliability requirements beyond the standard lateral analysis: With the unbalance-response-verification-test, a defined unbalance is applied to the rotor (American Petroleum Institute 2009). During physical testing, it is measured how the rotor behaves as a result of the applied load. In addition, the same situation is simulated during the lateral vibration analysis and compared to the results achieved during the testing. This means that matching result resemble a rating for the quality of the simulation results.

GENERAL DESCRIPTION OF THE APPLICATION AND COMPRESSOR REQUIREMENTS

In order to fully place the rotodynamic analysis and testing into proper context, it is important to review the fundamental parameters of the overall application and, consequently, the requirements this poses for the compressor. In the case of this Indian refinery, the application in question is for the processing of PSA Tail Gas. In principle, in this process, valuable components of a gas mixture are extracted and recovered for further use in the refinery's downstream processes: In the particular section of the process, the focus lies on hydrogen recovery. While the majority of the hydrogen (about 70%) is recovered during the primary PSA process, a portion of the hydrogen remains in the so-called tail gas. Like many other many gases found in refining processes, tail gas is a gas mixture with a very low mol weight (in this case, between 5 and 15 kg/kmol), and consists of N_2, H_2, AR, CO_2, CH_4, H_3OH, H_2S, COS. In this process section, the plant operators need additional compression of the gas mixture at a required flow of around 80,000 Nm^3/h. The objective here is the recovery of the remaining hydrogen from the tail gas mixture and achieving purity of 99.9% in a reliable manner.

Specifically, the tail gas passes through an acid removal plant, where the hydrogen is removed and sent to the CO Shift (a technology developed by Fluor): In that step, 80% of the hydrogen is separated out using the PSA Tail Gas process. Further downstream, the integrally-geared compressor is used to boost the gas pressure up to 5.7 bar(a) in order to separate out the high calorific value tail gas to be used in the burners of the heat recovery process.

DESCRIPTION OF THE TURBOCOMPRESSOR'S DESIGN CONCEPT FOR THIS PROCESS

The additional rotordynamic analysis and testing described in the latter part of this paper was conducted in the wake of the customer's specification requirements, and, consequently, the resulting compressor design that Atlas Copco Gas and Process proposed for this installation. To comply with the end user requirements, the company's engineering team conducted a preliminary rotordynamic analysis during the pre-order phase. The aim was to verify that the rotor model of the compressor to be offered to the customer was designed in line with the customer's aerodynamic requirements. The main reason for this upfront check was that the customer specification required a double dry gas seal configuration in combination with a relatively large impeller. The design used an integrally-geared compression concept with an overhung impeller design. Fundamentally, in integrally-geared compressor design, the impellers are located at the end of the rotor, with the bearings located between the impellers and the gear – which drives the rotor – being positioned between the bearings (Patel and Struck 2017). Because the impellers are not positioned between the bearings per se, the rotor design is called overhung design (see also Figure 1).

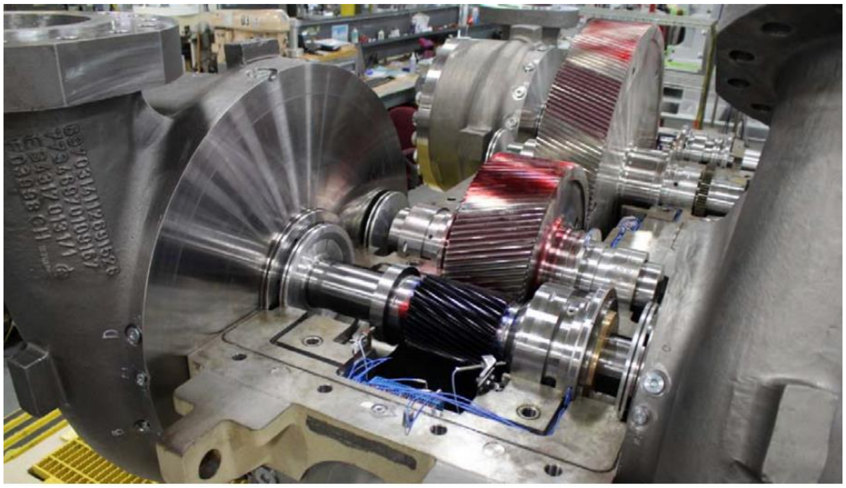

Figure 1: Overhung rotors in an integrally geared compressor during assembly.

The compressor design concept shall be described in greater detail in the following, thus setting the framework for a description of the additional rotordynamic analysis and testing performed.

For the PSA Tail Gas application, the process specification required a centrifugal compressor to compress hydrogen mix gas downstream of a Gasification-PSA with for process-related reasons a constant inlet pressure of 1,26 bara to about 5.72 bar(a) discharge pressure with operating cases accommodating different mol weights (from 5.9 to 13.7 kg/kmol), changing volume flows, as well as +/-3% frequency variations. This resulted in a comprehensive matrix of more than ~~36~~ 33 operating points (11 operating points at 48.5, 50 and 51.5 Hz, respectively).

Considering the above parameters, the aerodynamic design to accommodate these requirements would have typically called for a four-stage integrally geared centrifugal compressor. However, the specification for this project was setting mechanical boundaries and experience levels differently, thereby altering the recommended compressor design per customer amendment of API 617. In line with an API 617 customer amendment in chapter 1, impeller loads had to be within 75.6% of the impeller material's yield strength. In order to comply with this requirement, the load of the individual stages and impeller of this integrally-geared compressor had to be reduced in such a way that the impeller tip speeds translated into a yield strength below the 75.6% threshold. Thus, the outcome of this load distribution resulted in the proposal to use an integrally-geared design with five compression stages arranged on three pinions and mounted on one gear box.

Like many refinery processes, the specific application in question required double dry gas seals to meet process safety requirements. In this case, the double dry gas seal configuration was possible from a process standpoint of view because the inlet pressure of the compressor was close to atmospheric pressure and the process was able to accept additional nitrogen. Such a double dry gas seal requires more space compared to a single dry gas seal, for example, thus resulting in a larger impeller overhung.

With the relatively heavy impellers arranged in an overhung design, the engineering team at Atlas Copco Gas and Process that was supporting the bid process ultimately faced one key question it had to resolve: They needed to determine whether the three rotors could be delivered in compliance with stability criteria as specified under item 2.6 of API 617 chapter 1, and whether they could meet the requirement of a customer amendment to API 617 chapter 3 (2.7.1.3.1) calling for lower bearing metal temperature (American Petroleum Institute 2009). Being aware that API had the requirement of 100°C (212°F) maximum bearing metal temperature, the customer amendment specified that the temperature may not exceed 90/95°C (194°F/203°F) at most adverse operation with maximum oil inlet temperature of 50°C (122°F); this can be achieved through appropriate bearing design and power distribution.

With those parameters constituting the framework of requirements, the team needed to ensure that the rotor / bearing system would be technically feasible. Therefore, a rotordynamic analysis was performed even before accepting the order. The pre-order analysis performed by the engineering team rendered a positive outcome, confirming that the selected rotor configuration would comply with the requirements described before. The latter part of this paper will describe in greater detail the additional rotordynamic analysis and testing.

DESCRIPTION OF THE ROTORDYNAMIC ANALYSIS AND TESTING PERFORMED

In this specific project, the end user placed a very large emphasis on overall equipment reliability. With the deployment of a double dry face seal configuration, for example, a redundant seal system was specified. This type of seal requires significantly greater design space than single seal types, thus extending the impeller overhung. The increased rotor length causes a more filigree shape, which in turn makes the rotor more vulnerable to dynamic effects. This means that in this case the requirement for a more robust seal indirectly triggered caused the overall challenge of stable rotordynamic design.

For simulations, it is usually a model that serves as the basis, and naturally this will naturally represent a simplification of the real process conditions. For each such simulation used in industrial design processes, the goal is to simplify as much as possible in order to reduce numerical effort and simulation time. Additionally, the model shall deliver results with acceptable accuracy, and the test in this case was able to provide valuable insight into the projected rotordynamic behavior once the compressor would be in operation.

In this specific case, the customer for this compressor requested a verification of the results from the addition to the lateral vibration analysis using an unbalance-response-verification-test. In this verification test, a defined unbalance is placed on the final rotor. It is measured during a physical test how the rotor behaves as a result of the applied load. The same situation is simulated at lateral vibration analysis and compared to the results generated in the test bed. The match of the results provides a rating regarding the quality of the simulation results.

When performing standard testing procedure, the balanced rotor vibration is measured during a mechanical test run. Many compressor manufacturers (including Atlas Copco Gas and Process) perform such a mechanical test run for each compressor, and it provides a valuable assessment concerning the rotor dynamics inside the compressors that are to be supplied to the customer. In many cases, customers will tend to opt against the unbalance-response-verification-test because of the comparatively high cost compared versus standard test procedures performed during the mechanical test run. However, in this specific case, it was defined as a requirement. The basic requirements for the test procedure and limits for the accuracy of the simulation is described at API 617 7th Chapter 1 – 2.6.3 (American Petroleum Institute 2009). API still allows a certain degree of freedom for the test procedure, meaning the manufacturer can decide at which position of the rotor the additional unbalance is placed. In this case, the balancing notch at the impeller was chosen as the position for the unbalance weight. On this note, API 684 section 2.9.5, describes examples for unbalance response verification test.

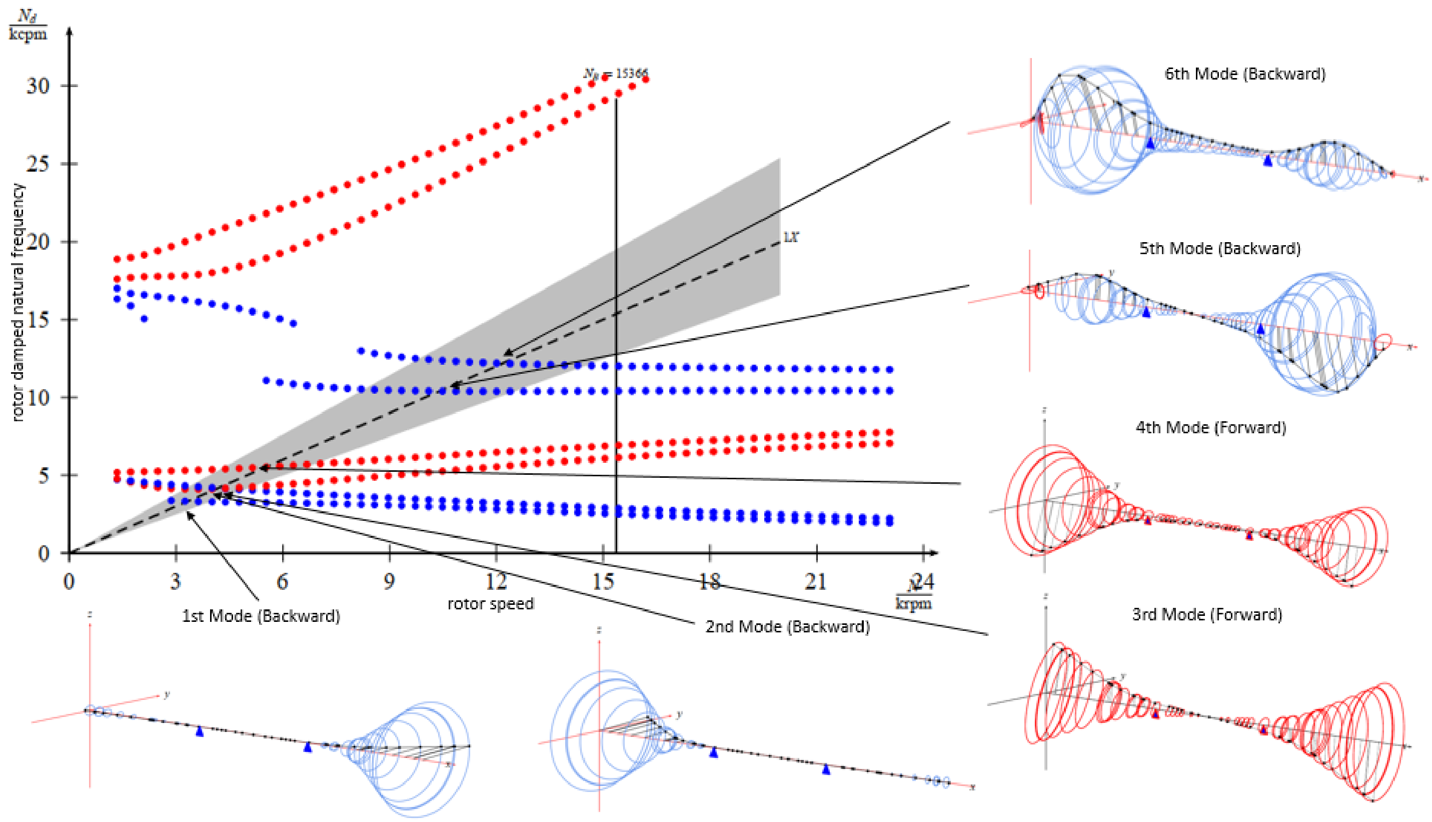

Figure 2. Campbell diagram of the second rotor and natural mode shapes

Figure 2 shows the Campbell diagram of the second rotor and the natural mode shapes which are driven through at startup. The color represents the rotational direction of the mode in relation to rotation direction. Blue dots and modes are backward modes, the red dots and modes represent forward modes. Resonances can occur if natural frequencies are excited. This can result in unacceptable vibration or even destruction of parts. An unbalanced load is typically the source of excitation on rotors during regular operation. Even a very well-balanced rotor will tend to display a residual unbalance, which can exceed natural frequencies. An excitation can occur if the frequency of the excitation matches the natural frequency. In addition, the exciting force leads to a deformation, which matches with the corresponding mode shape. If both criteria are fulfilled, the rotor can still operate close to a natural frequency if the damping of the system is high enough to dissipate the induced energy. Hydrodynamic friction-type bearings are usually a source of damping for the rotor. The backward modes shown in Figure 2 are not excitable by unbalance loads, because excitation load and mode shape do not match.

As a standard rotordynamic design procedure, compressor manufacturers perform a damped unbalance response analysis for each pinion delivered to the customer. Three load cases based on unbalance loads are assumed:

- Unbalanced load at both impellers without phase angle between loads (Figure 3)
- Unbalanced load at both impellers with phase angle of 180° (Figure 4)
- Unbalanced load at the center of the pinion (Figure 5)

Figure to Fig show the relation between excitation, mode shape and expected reply of the rotor. The colors of the amplification curves show different positions along the rotor. Only the two forward modes are excitable and generating peaks at the plots. At the third natural mode, both shaft ends are vibrating in opposite direction. This mode is much more excitable at the second load case where the unbalance load is applied with the same phase angle. The natural mode shape 4 shows an analog behavior in combination with the first load case.

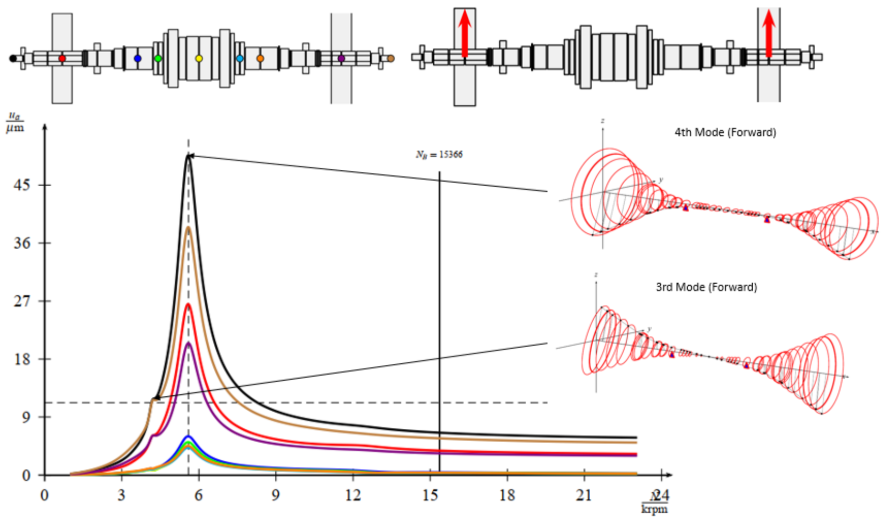

Figure 3. Damped Unbalance Response Analysis Load Case 1

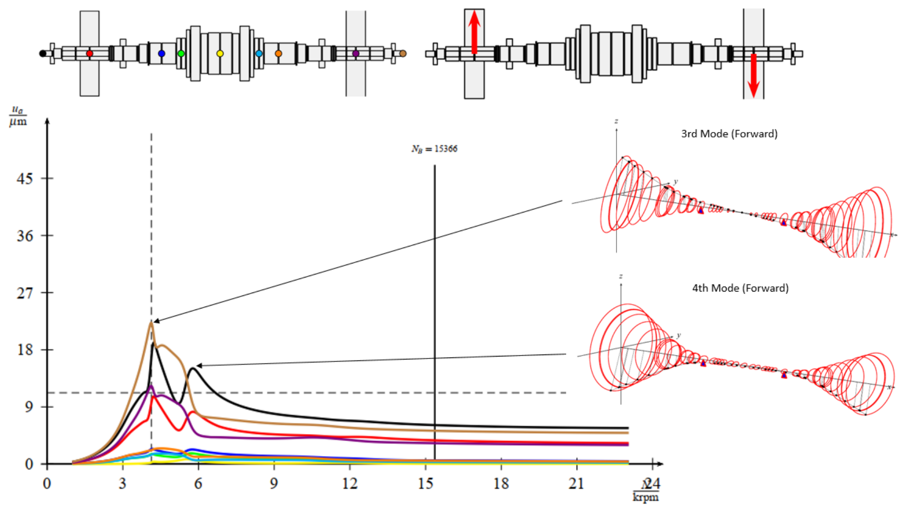

Figure 4. Damped Unbalance Response Analysis Load Case 2

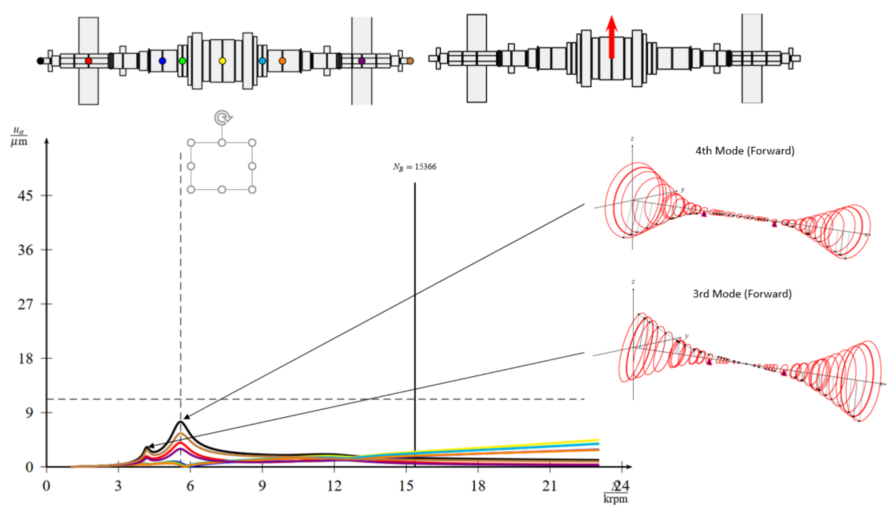

Figure 5. Damped Unbalance Response Analysis Load Case 3

In order to verify the simulation model, an unbalance of eight times API-unbalance U was applied at the back of the fourth impeller. For the test, the rotor was accelerated from standstill to maximum operating speed. Also, vibrations during startup were measured using vibration probes. The same scenario was simulated using a model identical to the one used in the lateral vibration analysis. A comparison between both results of the second rotor (shown in Figure 7).

As expected, the measurement also showed only the two peaks related to the forward modes three and four. The peaks are slightly overestimated, which proved that the simulation considered worst-case damping conditions. The natural frequency was simulated with an accuracy better than 4% for all shafts.

Figures 6-8 provide a complete overview of the unbalance response verification test for all the three rotors.

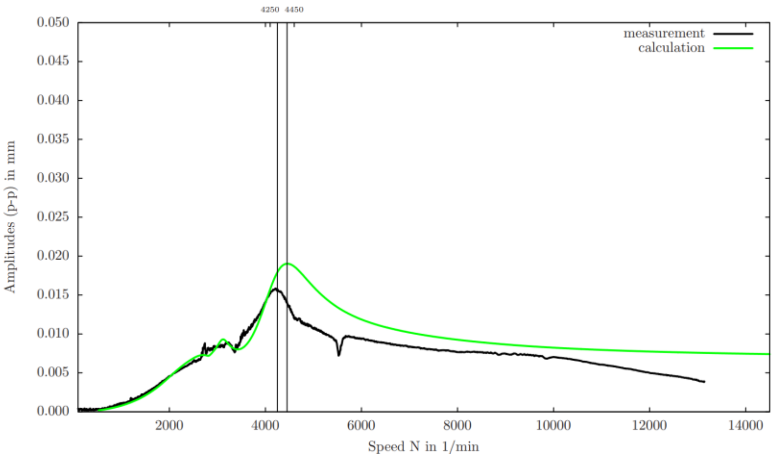

Figure 6. Results of Unbalance Response Verification Test on Rotor 1

Figure 6 shows the comparison of calculated reaction of the first rotor (green curve) with the measured reaction (black line). Both curves are showing two peaks due to two excitable natural frequencies. The maximum amplification at excitation of the fourth mode is slightly overestimated in the simulation. This represents a minor deviation between measurement and simulation, but the prediction of the simulation model is on the safe side. The rotational speed of the shaft when the maximum vibration occurs is predicted with an accuracy of 4.5%, thus fulfilling the API requirements. A comparison of the predicted amplitudes is not required by API, but even the amplitudes of measured and calculated start-up were shown to be matching very well.

The third rotor – depicted in Figure 8 – showed a similar behavior: There was a good correlation of 4% between measured and calculated natural frequency. Also, the level of amplification was calculated with satisfactory accuracy.

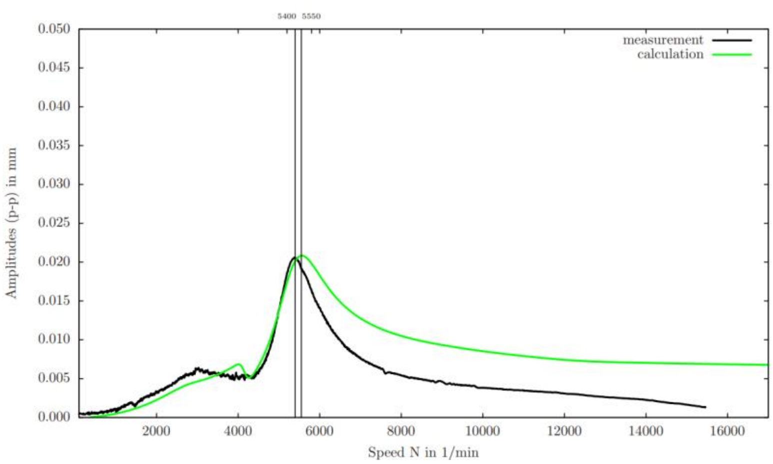

Figure 7. Results of Unbalance Response Verification Test on Rotor 2

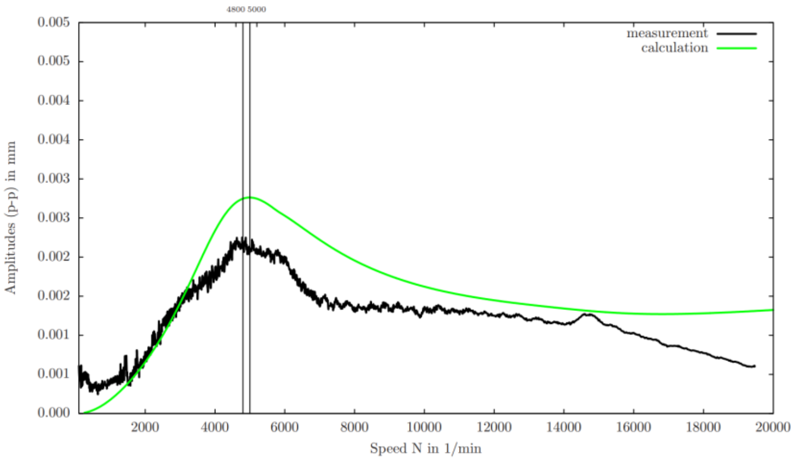

Figure 8 - Results of Unbalance Response Verification Test on Rotor 3

CONCLUSIONS

In today's refineries, thousands of often complex and intricate processes are in operation, driven and safeguarded by different types of compression methods, with reliability being a universal maxim. In this paper, the authors demonstrated how the five-stage integrally-geared compressor design concept in combination with additional rotordynamic and analysis testing allows for compliance with stringent reliability requirements by end users in the refinery industry (i.e., PSA Tail Gas applications), spanning well beyond the scope of API 617. As shown in the paper, advances in dry gas seal technology, increased knowledge on aerodynamic and rotor dynamic design and performance prediction, and more advanced manufacturing methods in producing both accurate rotating parts and complex components have widened integrally-geared compression technology's role as a feasible choice for the refinery and petrochemical world.

NOMENCLATURE

API = American Petroleum Institute
IGC = Integrally-geared compressor
PSA = Process Swing Adsorption

REFERENCES

American Petroleum Institute, 2009, "API 617, Axial and Centrifugal Compressors and Expandercompressors, 7th edition.

American Petroleum Institute, 2010, "API 684, Standard Paragraphs Rotodynamic Tutorial: Lateral
Critical Speeds, Unbalance Response, Stability, Train Torsionals and Rotor Balancing, volume second edition.

Beaty, P., Eisele, K., Maceyka, T., et. al. 2000, "Integrally-geared API 617 process gas compressors," *29th Turbomachinery Symposium Houston.*

Patel, T. and Struck, H. 2017, "Ensuring integrally-geared compressor reliability wit API 617," *10° Fórum de Turbomáquinas*, Rio de Janeiro, November 28-30, 2017.