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EXPERIMENTAL INVESTIGATION OF VIBRATIONAL AND ACOUSTIC PHENOMENA FOR DETECTING THE STALL AND SURGE OF A MULTISTAGE COMPRESSOR

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ABSTRACT

Nowadays, the operative range limit of compressors is still a key aspect of the research into turbomachinery. In particular, the study of the mass flow rate lower limit represents a significant factor in order to predict and avoid the inception of critical working conditions and instabilities such as stall and surge. The importance of predicting and preventing these dangerous phenomena is vital since they lead to a loss of performance and severe damage to the compression system and the compressor components. The identification of the typical precursors of these two types of compressor unstable behaviors can imply many advantages, in both stationary and aeronautic applications, such as i) avoiding the loss of production and effectiveness of systems and ii) reducing the cost of maintenance and repairing. Many approaches can be adopted to achieve this target, but one of the most fascinating is the vibro-acoustic analysis of the compressor response during operation. At the Engineering Department of the University of Ferrara, a test bench, dedicated to the study of the performance of an aeronautic turboshaft engine multistage compressor, has been equipped with a high frequency data acquisition system. A set of triaxle accelerometers and microphones, suitable for capturing broad-band vibration and acoustic phenomena, were installed in strategic positions along the compressor and the test rig. Tests were carried out at different rotational speeds, and with two different piping system layouts, by varying the discharge volume and the position of the electric control valve. Moreover, two different methodologies were adopted to lead the compressor towards instability. This experimental campaign allowed the inception of compressor stall and surge phenomena and the acquisition of a great amount of vibro-acoustic data which were firstly elaborated through a cyclostationary domain analysis, and then correlated to the thermodynamic data recorded. Successively, the precursor signals of stall and surge were detected and identified. The results of this paper can provide a significant contribution to the knowledge of the inception mechanisms of these instabilities. In particular, the experimental data can offer a valid support to the improvement of surge and stall avoidance (or control) techniques.

INTRODUCTION

Nowadays, unstable compressor behavior is not yet fully understood and is still an attractive topic for many researchers since it strongly affects the operability and functionality of these turbomachines and thus the systems, such as gas turbines, of which they are fundamental components.

In particular, unstable operation can arise in process industries due to startups or unpredictable and rapid transients during operation. These types of events can lead the compressor to rotating stall or surge.

These two phenomena are related to each other since stall very often precedes surge [1, 2]. The degree of danger of these two phenomena is not only due to the severe loss of aerodynamic nominal conditions, which bring about decreases in power and efficiency, but also to the mechanical damage that they can produce inside the compressor and mechanical system components [3]. These effects can, in some cases, lead to the destruction of the compressor or some component of the piping system.

The vibrations and noise can contribute to the deterioration of the compressor components.

Vibrations, intended as casing vibrations and blade vibration, are important since the first instantaneously modifies the tolerance of the compressor while the second, if hard, can lead to the partial or total breaking off of the blades [4] [5].

Noise is a less complex and dangerous threat than vibrations but recently the study of aeroelasticity is becoming more popular in order to examine some aerodynamic and aeroacoustic aspects [6], and to design acoustic treatments. Interesting examples, with application to stall and surge conditions, are provided by the work of Vahdati et al. [7] and that of Schoenenborn and Breuer [8].

It is clear that in unsteady processes these quantities represent a serious complication and can often be related since both of them can reveal important clues regarding the operating condition of the compressor.

Stall and especially surge are often preceded by typical audible pulsating sounds and vibrations which help to understand that the operating limit of the compressor is about to be reached [9]. In fact, surge and near surge conditions are accompanied not only by outlet temperature excursion and fluctuating differential pressure (and mass flow rate), but also by the resulting pulsating noise and the increasing of vibrations (especially in the low frequency range). However, these precursors are not always detected, so the threat of failure cannot always be recognized.

On the other hand, stall can also be dangerous even without degenerating into surge. Stall cells can, in fact, introduce disturbances which can excite some blade passing frequency (or their harmonics), as well as the natural frequencies of the blades, impeller shroud or casing.

Some compressors (typically small scale) can operate for a long time in mild surge conditions, resisting pressure and mass flow oscillations without registering mechanical harm; and even recovering by themselves from the instability (if the instability is light and depending on the operating conditions and layout of the system). However, the dangerous effects of stall and surge should be avoided and somehow predicted or detected in their inception phase. This is the context in which this work is focused.

In this paper an experimental campaign on a multistage axial-centrifugal compressor is carried out. The facility is the same as that described in [10] but, in this case, additional sensors (accelerometers and microphones) were installed along with their specific high frequency data acquisition system (see section "Experimental Test Rig").

The aim of the paper is to compare the dynamic vibro-acoustic response to the thermodynamic data recorded during tests, demonstrating the evidence of stall and surge by using the aforementioned new instrumentation and obtaining supplementary information about these two phenomena.

NOMENCLATURE

BPF	blade passing frequency
f	spectral frequency
g	acceleration
$\dot{m}$	mass flow rate
N	rotational speed
p	pressure
$P_x(t)$	mean instantaneous power
P_x^{α}	cyclic power at cyclic frequency α
$P_x^{\alpha}(f;\Delta f)$	cyclic modulation spectrum
RH	relative humidity
T	temperature
t	time
VPF	vane passing frequency
$x(t)$	time signal
$x_{\Delta f}(t;f)$	filtered version of the time signal through a frequency band of with Δf centered at frequency f .
z	number of blades/vanes
α	cyclic frequency
Δf	frequency band
Subscripts and superscripts	
0	Stagnation physical quantity
1, 2	test rig significant sections
1a, 3a, 6a	first, third and sixth axial stage
1c	radial stage
amb	ambient
f	frontal
mot	electric motor
r	rotor
s	stator
t	top
x	axial direction
y	lateral direction
z	vertical direction

LITERATURE REVIEW

The relevance of compressor stall and surge has led to many experimental campaigns, or theoretical studies, in order to increase the knowledge of these two phenomena and their inception process [11-14]. The research advancements on stall and surge knowledge were reported by Day [15], who wrote an exhaustive review of the main works regarding this topic.

In literature, many experimental studies on compressors are present and all of them use different instrumentation, methodologies and signal analyses.

A recent work [16] was conducted by Li et al. They have analyzed test rig data, by means of the short-time-Fourier-Transform, from a seven-stage compressor by using both casing static pressure and outlet total pressure signals. Their analysis revealed that their methodology was capable of identifying stall precursors.

An interesting paper was also presented by Gallus and Hoenen [17], who have studied the airfoil and endwall boundary layers in a subsonic compressor stator. By analyzing hot-wire probe results, they demonstrated that when the compressor was approaching stall, the separated region of the blade boundary layers was evident. In that analysis the periodic signal of the blade could not be identified in the separated region, and therefore, the blade passing signal deteriorated towards stall. A few years later, the same authors carried out a further investigation presenting a method for monitoring aerodynamic load [18]. Based on the experimental results, the spectra analysis showed a different pattern with respect to nominal conditions when the compressor was approaching the surge line. They measured the unsteady pressure distribution in 8 stages of the multistage axial compressor of a LM5000 gas turbine and found a reduced pressure rise in the 13^{th} stage, where periodic fluctuations (blade passing frequency) disappeared. They correlated this event to the rotating stall phenomenon occurring on that stage.

Bright et al. [19] explored various techniques for identifying the stall precursors in several configurations of a high speed compressor stage, exposing the potential advantages of the correlation integral method.

In [20] an analysis of compressor aerodynamic instability precursors was carried out by installing pressure sensors on the compressor casing located over the rotor of three different compressors. The authors used a modified form of the auto-correlation based approach to quantify the repetitiveness of the blade signature, finding that the correlation parameter reduced when the compressor was operating close to the stability range limit. This phenomenon was interpreted as a stall and surge precursor. The same phenomenon and the same conclusions were also reported in [21] where the deterioration of the blade passing periodicity signal was encountered and recognized as a stall precursor.

A similar work was also conducted by Young et al. [22], who focused their study on the relationship between the blade passing non-repetitiveness near stall, rotor eccentricity and tip-clearance. This work demonstrated that i) when approaching stall, the blade passing irregularity changes with the tip-clearance geometry and ii) eccentricity also affects the signal which always achieved its maximum non-repetitiveness value in the proximity of the maximum clearance. They concluded that the use of the blade passing signature analysis would not be a reliable stall warning method by itself, but at least another parameter between tip clearance size or eccentricity should be measured (obviously taking into consideration other sources of irregularity).

Their result is partially in contrast with the work of Christensen et al. [23] who developed a real-time algorithm based control software for stall management. They used over-the-rotor dynamic pressure sensors and adopted a correlation method, similar to that of Dhingra et al. [20], suggesting that this system can potentially be used in routine engine testing, but also for military and commercial engines.

Nowadays, other novel methods and instruments are being proposed in order to monitor the condition of gas turbines online, for identifying compressor stall and surge inception, but also regarding the status of the gas turbine components (in particular the blades).

One of these methods is vibro-acoustic analysis. This approach could also potentially give important information only by measuring the casing vibrations [24, 25] and therefore using a non-intrusive technique; which represents a relevant aid to the maintenance and continued operation of these machines.

In fact, the vibration and acoustic measurements could enable a deeper comprehension of unsteady phenomena effects [26].

An experimental campaign conducted by Lawless and Fleeter [27, 28] was based on the Fourier analysis of simultaneously sampled data regarding sensitive electret microphones. The microphones were uniformly located around the inlet and three different diffuser geometries of a compressor. The authors also demonstrated the reliability of their methodology in the detection of stall precursor by finding different typologies of stall pattern depending on the diffuser mounted.

In [29] a preliminary analysis of the incoming flow instabilities near surge, based on vibro-acoustic measurements, allowed the identification of the unstable behavior precursors. In [30] a similar investigation was also made by Aretakis et al. who continued the research of their previous works [31, 32]. They found that an acoustic signature, exhibiting a good correlation with the operating point of the compressor curve, can be obtained. They also found that the most suitable parameter for the determination of the compressor condition (unstalled, stalled or in surge) is the acoustic pressure level RMS value corresponding to a sub harmonic frequency range.

All these examples testify the relevance and the potential of the various approaches for studying stall and surge in a compressor. In particular, the vibro-acoustic analysis has demonstrated great potential which should be explored by the further research.

CYCLOSTATIONARY ANALYISIS

Rotating machine vibro-acoustic signals can be naturally divided into periodic and random components [33]. Excitation forces induced by machinery rotating mechanisms result in the periodic counterpart of the vibro-acoustic signal. On the other hand, random contribution is mainly related to the non-periodic mechanisms such as: turbulence around fan blades, admission and exhaust of fluids in pumps, and so on.

By studying the energy flow inside the random contribution, pivotal information on the operating condition of the machine can be obtained. Signals which exhibit hidden periodicity of their energy flow are said to be cyclostationary. To the knowledge of the authors, there are no previous applications of the cyclostationary analysis of vibro-acoustic signals for the detection of both stall and surge phenomena in compressors.

The rationale of cyclostationarity was firstly formalized by Gardner [34]. However, an approach based on the concept of the "energy conveyed by the signal" was introduced by Antoni in [35]. This approach details a new definition and framework for the analysis of such signals. Several orders of cyclostationarity can be defined. Generally speaking, any cyclostationary behavior that can be detected by a non-linear transformation of degree n is referred to as nth order cyclostationarity. De facto, when a non-linear transformation of second degree highlights hidden periodicity inside the signal (second order cyclostationarity), the energy is conveyed in a periodic fashion. The central idea consists of decomposing the energy flow not only into a constant trend, but also into periodic components which depict how the energy is travelling with time.

This decomposition is performed by the mean instantaneous power (MIP), which reads as the Fourier series of the energy flow per unit of time at each instant t [35]:

$$P_x(t) = \sum_{\alpha \in \mathrm{A}} P_x^{\alpha} e^{j2\pi\alpha t} \tag{1}$$

Where

$$P_x^{\alpha} = \lim_{T\to\infty} \frac{1}{T} \int_T |x|^2 e^{-j2\pi\alpha t} dt \tag{2}$$

is the cyclic power of the signal at cyclic frequency α and A is the set of all the cyclic frequencies associated with non-zero periodic components. The MIP provides a global vision as to how the signal energy is slowing over time.

Among all the descriptors used for analyzing cyclostationary signals, the Cyclic Modulation Spectrum (CMS) has the ability to display the cyclic frequency α along with the spectral frequency f, and is defined as [35]:

$$P_x^{\alpha}\left(f;\Delta f\right) = \lim_{T\to\infty} \frac{1}{T} \int_T \left|x_{\Delta f}(t;f)\right|^2 e^{-j2\pi\alpha t} dt \tag{3}$$

Where $x_{\Delta f}(t;f)$ is the filtered version of the signal through a frequency band of width Δf centered at frequency f. Therefore, the CMS is simply the Fourier Transform of the well-known Spectrogram. The spectral and cyclic frequencies can be interpreted as the carrier frequency of the wave that transports the energy and its modulation frequency. In fact, the signal envelope represents the energy fluctuation as a function of time, which is completely reminiscent of the MIP definition. Consequently, the CMS represents the envelope frequencies α as a function of the carrier frequency f, computed in the frequency band of width Δf.

EXPERIMENTAL TEST RIG

The experimental test rig is located at the Engineering Department of the University of Ferrara. The entire facility as well as the data acquisition system of thermodynamic sensors has been exhaustively described in [10], thus in this section only a brief summary is reported.

The object of this study is a multistage axial-centrifugal compressor, see Fig. 1, that has six axial stages and one centrifugal. The number of blades for each stage of the compressor are illustrated in Tab.1 together with the blade passing frequencies (BPFs) and the vane passing frequencies (VPFs) at N=10,000 rpm (only these values are shown as the results of this paper are focused on that velocity). The BPFs and VPFs are calculated as:

$$\mathrm{BPF} = \frac{N\, z_r}{60} \tag{4}$$

$$\mathrm{VPF} = \frac{N\, z_s}{60} \tag{5}$$

The piping system (a scheme is proposed in Fig. 2) is thought to be modular so that two layouts can be obtained: Layout #1, which is equipped with a small discharge volume and Layout #2, which instead consists of a large discharge volume and is dedicated to surge analysis. The mass flow rate is regulated by controlling Valve 1 in Layout #1, and Valve 2 in Layout #2. The inlet configuration is the same for both layouts and consists of an inlet duct where an orifice plate is installed.

The compressor is driven by an electric motor which is remotely regulated by a control system software developed in LAbVIEW. In addition to the thermodynamic sensors already installed [10], see Fig. 2a, the compressor was equipped with additional instrumentation with its specific high frequency data acquisition system.

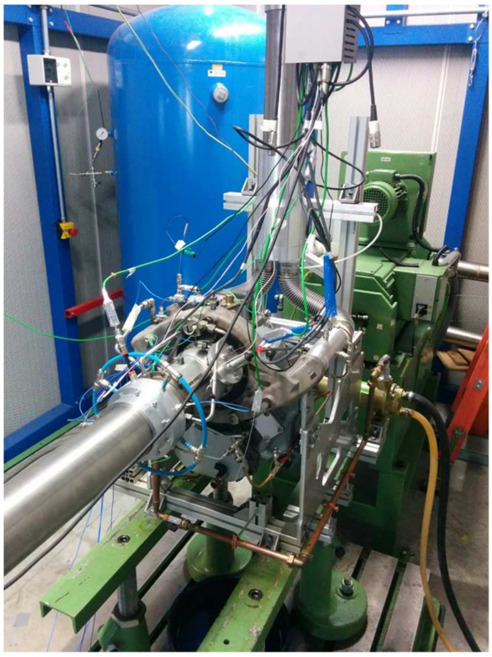

Fig. 1. The tested compressor

Tab. 1. Number of stator and rotor blades of each stage

Stage	Type	Rotor (z_r)	Stator (z_s)	@ 10,000 rpm BPF [Hz]	@ 10,000 rpm VPF [Hz]
1st	Axial	16	14	2666	2333
2nd		20	26	3333	4333
3rd		16	28	2666	4666
4th		25	32	4166	5333
5th		28	36	4666	6000
6th		25	30	4166	5000
7th	Centrifugal	28	12	4666	2000

In particular, a set of triaxial accelerometers and microphones were adopted to record vibrational and acoustic signals during compressor operation. The installed accelerometers have a linear frequency range of 1 Hz to 10 kHz. Vibro-acoustic signals were acquired by means of LMS

Scadas III front-end allowing a maximum sample frequency of 204,800 Hz.

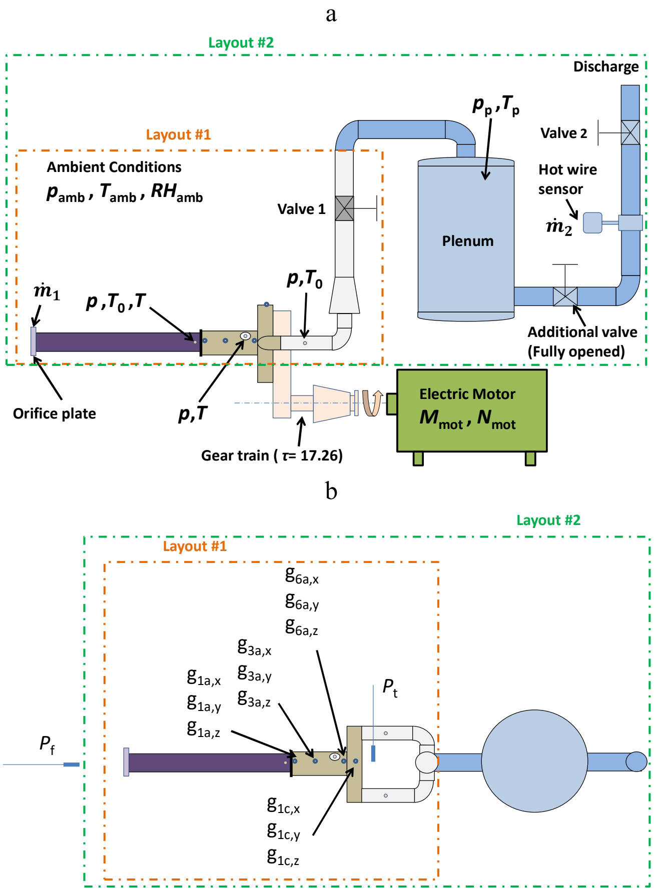

Fig. 2. Piping system configuration. Layout #1 and Layout #2 with the (a) thermodynamic and (b) vibro-acoustic sensors.

The accelerometers and microphones, Fig. 2b, like the sensors for thermodynamic parameters, Fig. 2a, were installed in strategic locations along the compressor, the piping system and the test rig zone. In particular, three accelerometers were installed on the axial part of the compressor casing and another on the centrifugal stage casing as shown in Fig. 3.

In the axial compressor the accelerometers, Fig.3a, were placed in the vicinity of the stator/rotor boundary of the first and the sixth stages, and on the rotor of the third stage. In the radial compressor, an accelerometer (Fig. 3b) was positioned on the centrifugal stage casing. The images of the microphones are not shown in this paper.

Tab. 2. Type of tests carried out

Piping system	Controlled valve	Closing mode of the valve	Type of test
Layout #1	Valve 1	Step-by-step	Steady-state
		Continuous	Transient
Layout #2	Valve 2	Step-by-step	Steady-state
		Continuous	Transient

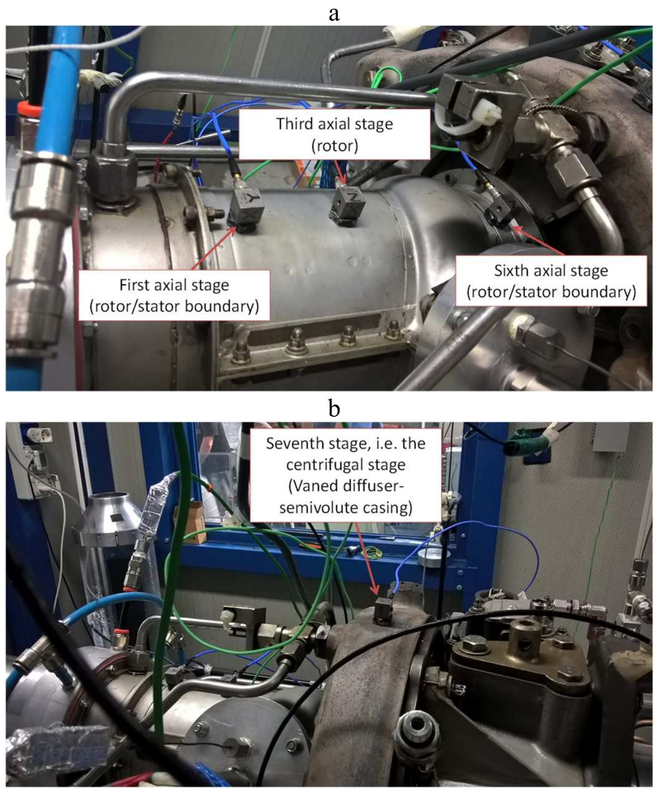

Fig. 3. Installation of the accelerometer on the axial part (a) and the radial part (b) of the compressor

METHODOLOGY AND DATA ACQUISITION

The test methodology is close to that of [10] so the compressor is tested in both Layout #1 and Layout #2, and it is driven to the instable regime by closing the control valve (Valve 1 or Valve 2). The first configuration is suitable for stall analysis, since the limited compressor downstream volume, also facilitates the stability of the operating points beyond the characteristic curve peak of the compressor. On the other hand, the second configuration is suitable for surge analysis since the accumulating mass in the plenum generates the compressor surge, once Valve 2 has achieved a certain degree of closure.

The compressor was tested at different rotational speeds: 7500, 10,000, 15,000 20,000 rpm. However, in this paper only the results at 10,000 rpm are shown since they are representative of the phenomena which were also detected at the other velocities. Both the continuous and the step-by-step closure (with steps of 5-10 degrees, depending on the proximity to the surge line) of the valves were carried out, in Layout #1 and Layout #2. Concerning the step-to-step closing valve mode test, each measurement was made after waiting a certain time which allowed the complete stabilization of the compressor regime. Thus, in this paper, this test will be called steady-state. On the other hand, transient tests are conducted by closing the control valve, with a closing velocity of 1.5 °/s, until surge occurs while the data are recorded throughout the experiment. Table 2 briefly illustrates the type of tests performed, specifying the layout, the control valve used and the closing mode of the valve. The vibro-acoustic sensors were recorded simultaneously with a sample frequency of 51,200 Hz for an extent of 60 s in the case of steady-state measurement, while during transient tests, signals were acquired for a long enough time to encompass all the valve closure. The aim is the identification of the fluid-dynamic compressor instabilities, i.e. stall and surge, through the analysis of the vibro-acoustic compressor signature.

RESULTS

The experimental data were recorded by using all the sensors shown in Fig. 2. In this paragraph only the most significant results are depicted. In particular, they concern the accelerometers positioned at the third stage as well as the intake duct microphone.

Firstly, the surge phenomenon is outlined by the analysis of the microphone P_f signals. Due to its type and location, this sensor is responsive to the air stream coming out of the compressor. Figure 4 highlights the microphone P_f time signal with its MIP in the case of Layout #1 in a steady-state test for two valve positions, i.e. completely open and completely closed. The randomness of the time signal is noticeable for the completely opened valve (Fig. 4 (a)), while as the valve closes (Fig. 4 (b)), a periodic component dominates the signal. Moreover, the periodicity of the MIP highlights that the signal energy is flowing in a periodic fashion. This is related to the air stream hitting the sensor during the instability of the compressor operating condition, i.e. the surge. The cyclic power of such a signal can then be evaluated in order to extract the cyclic frequencies of the energy flow.

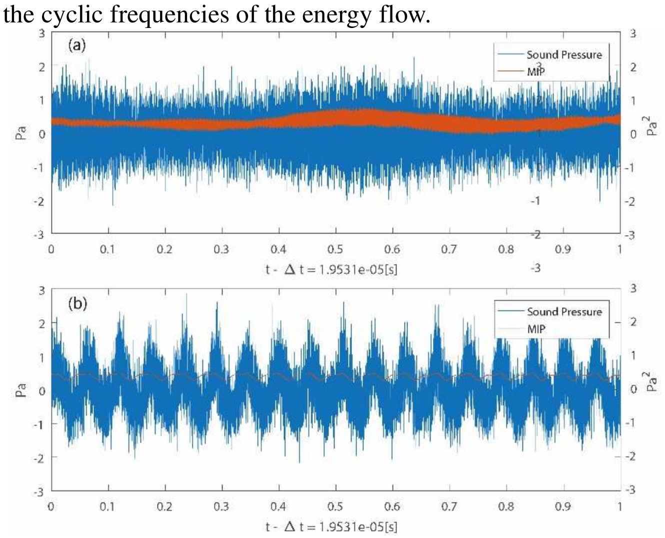

Fig. 4. Layout #1 steady-state test microphone *P*f time signal (left) with its MIP (right): (a) valve completely opened, (b) valve completely closed

The evolution of the surge phenomenon can be highlighted by the transient test in which the valve is continuously closed until surge occurs. In order to extract the signal cyclic power during the compressor dynamic operating condition, a Hanning window function of a two second width [35] is slid over the time signal and the cyclic power of the windowed signal is evaluated. Figure 5 depicts the result of this analysis. In particular, as the time increases (the valve is completely opened @ t=0 and then it starts to be closed), the cyclic frequency at 18Hz arises with its main harmonics (35.5 Hz and 53.5 Hz). De facto, this cyclic frequency is related to the energy flow during the surge phenomenon. This sensor does not perceive the modulation due to the stall cells during dynamic operation, but it only registers the characteristics of the deepest instability, i.e. the surge.

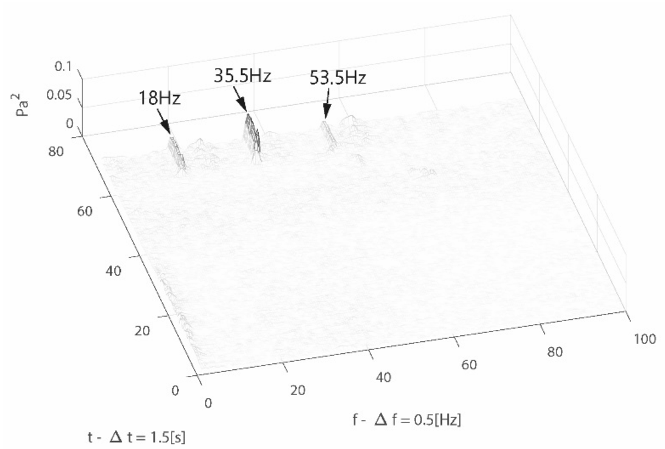

Fig. 5. Layout #1 transient test: cyclic power of microphone *P*f

Pivotal information concerning the compressor stall phenomenon can be obtained by the analysis of the acceleration signals. The CMS of the vibration signals is evaluated during the steady-state test in Layout #1. This technique can emphasize the modulation frequencies related to the signal energy flow with respect to its carrier frequencies. Figure 6 depicts the results of this analysis. It is possible to see that strong modulation phenomena are visible around 4000 Hz, where the main BPFs of the system are located (see Tab. 1). The main modulation frequencies involved are at 20.17 Hz and 49.08 Hz in the case of the completely opened valve (Fig. 6 (a)). This result changes slightly when the valve is completely closed, Fig. 6 (b). Firstly, the modulating component at 20.17 Hz decreases its frequency to 20.03 Hz, and the component at 49.08 Hz decreases its amplitude. Moreover, it is possible to see the emergence of a new cyclic frequency at 17.94 Hz which involves a greater range of spectral frequencies. As explained before, the spectral and cyclic frequencies are respectively the carrier and the modulating frequency of the energy flow inside the signal. Through analysis of the P_f microphone signals, it is possible to relate the component at 17.94 Hz to the surge phenomenon, while the other modulating frequencies could be related to other fluid-dynamic phenomena, such as stall cells. Therefore, in order to depict this behavior, an analysis of the acceleration signals during the transient test in Layout #1 is carried out. A Hanning window function is slid over the acceleration time signal and the cyclic power for each windowed portion is evaluated and depicted in Fig. 7. Two main cyclic frequencies are visible at the beginning of the test, i.e. 20 Hz and 49 Hz, when the valve is completely opened. As previously shown, this phenomenon is related to fluid-dynamic perturbations which modulate the energy flow carried by the main system BPFs. Therefore, they can be easily related to stall cells.

This is in agreement with what was found in [10], in which two high frequency dynamic pressure sensors installed at the inlet annulus of the compressor registered a stall cell rotating with a frequency around 20 ± 0.5 Hz. The analysis of the acceleration signals confirms the presence of a stall cell located in the first axial stages of the compressor (the exact position and size is difficult to define due to the absence of inside pressure measurements). Moreover, as mentioned above, at least another stall cell is present in the successive stages, and this cell/s rotates with a frequency of 49 Hz. Unfortunately, it is not possible to establish the number of stall cells without using other types of intrusive sensors.

This phenomenon is due to the operating conditions. De facto, the compressor is operating at a very low rotational speed, compared to its nominal speed; which is the reason why the stall cells generate at the first stages.

An interesting phenomenon successively occurs by further closing the valve. Figure 7 shows that at a certain closing angle of Valve 1 (approximately 25-30 °, around t=60 s), a new cyclic frequency of about 60 Hz generates. Immediately before, the stall cell rotating at 49 Hz disappears.

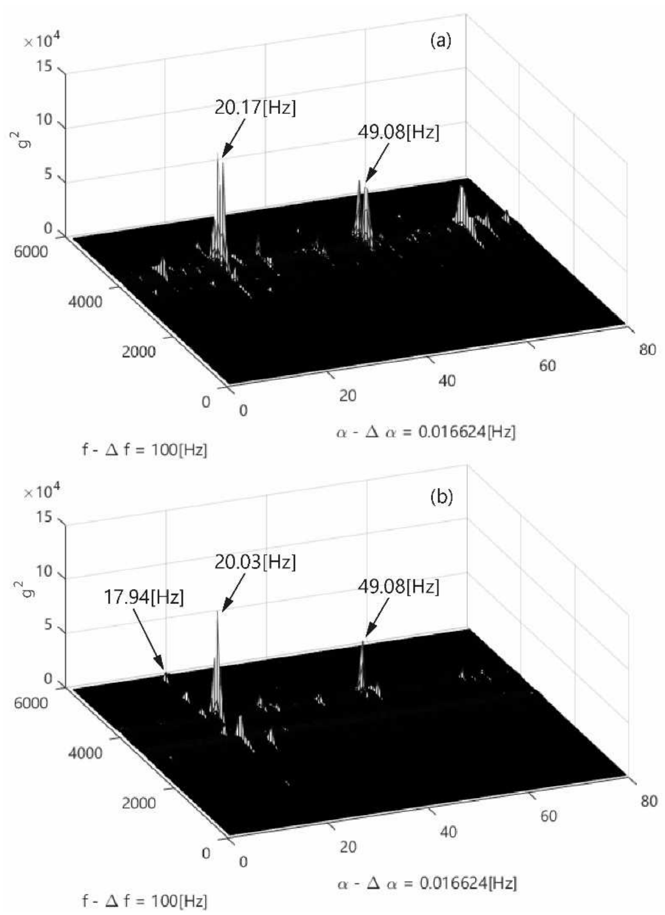

Fig. 6. Layout #1 steady-state test CMS of the $g_{3a,x}$ accelerometer: (a) valve completely open, (b) valve completely closed

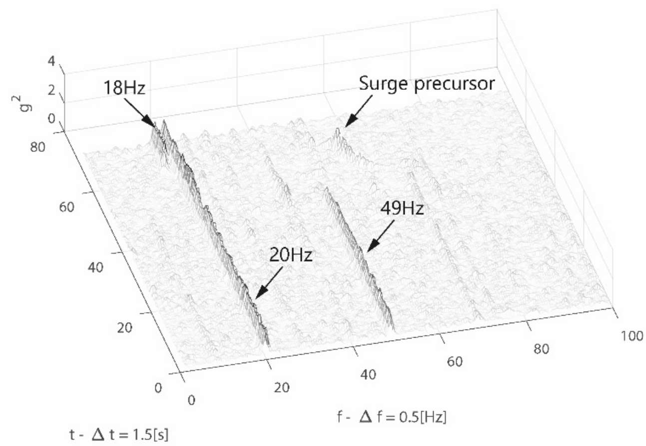

Fig. 7. Layout #1 transient test: cyclic power of accelerometer $g_{3a,x}$

This highlights the evolution of the compressor stall along with the changing size of the existing stall cell/s, or its interaction with potential new stall cell/s generated. The new modulating frequency, which represents the new stall cell/s, is obviously a surge precursor and decreases with the closure of the valve. Successively, when the valve is completely closed, surge occurs with a frequency of 18 Hz. This type of surge can be described as an instability which evidently develops because of the total absence of mass flow rate. Therefore, its high frequency is due to the small downstream volume, and thus to the small accumulation of energy that this volume allows. Another important fact is that, in surge, the original cyclic frequencies are still present, although with a lower amplitude. This confirms that in this condition, the compressor alternates surge and rotating stall of the first stages depending on the direction of the flow.

The same analysis has been carried out for the Layout #2 (with the plenum circuit). Figure 8 depicts the cyclic power of the acceleration signal evaluated in the same way as for Layout #1 (Fig. 7). The main cyclic frequencies (20 Hz and 49 Hz) related to stall phenomenon are still visible in Layout #2. As expected, the cyclic frequency related to the surge event (1 Hz) has a lower value compared to Layout #1, due to the bigger compressor downstream volume. This consideration is supported by what has been shown in [10]. However, in the case of Layout #2 it not possible to see the surge precursor (see Fig. 8 compared to Fig. 7). This behavior is mainly due to the influence of the piping system layout on the compressor dynamic characteristic behavior. In fact, in Layout #1 the fluid-dynamic instability occurs beyond the characteristic curve peak, while in Layout #2 the surge occurs in the vicinity of the peak, see [10]. Therefore, the evolution of the surge precursor cannot be clearly identified due to the rapid change in condition from stall to surge.

The results showed above highlight the possibility of detecting stall cell/s perturbation, surge precursors, as well as surge phenomena, with non-intrusive sensors such as accelerometers.

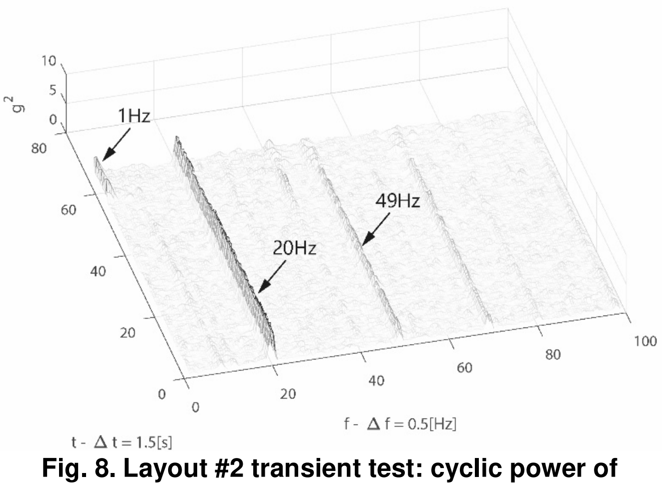

Fig. 8. Layout #2 transient test: cyclic power of accelerometer $g_{3a,x}$

CONCLUSIONS

This paper deals with a vibro-acoustic experimental study of the instability phenomena, such as stall and surge, occurring in a multi stage compressor.

Accelerometers were installed on the compressor casing in three different positions, and two microphones were positioned immediately above the compressor, and at a certain distance from the intake duct.

Two types of test have been carried out with two different piping system layouts, namely Layout #1 and #2.

Cyclostationary analysis, which is a novel methodology for turbomachine data analysis, has been carried out on the vibro-acoustic signals in order to study energy flow related to fluid-dynamic instability phenomena. This analysis has shown effectiveness in detecting stall cell/s perturbation, surge precursors, and surge phenomena. In particular, the cyclostationary analysis carried out on the acceleration signals showed the presence of two cyclic frequencies at 20 Hz and 49 Hz, which are related to rotating stall cell/s. Moreover, in Layout #1, the evolution of stall to surge is clearly visible as well as the surge frequency. Similar results have been obtained in Layout #2 in which the surge precursor was not clearly detected. This is mainly due to the influence of the piping system layout on the compressor dynamic characteristic behavior.

This work shows how an appropriate signal processing analysis of the vibro-acoustic signals could be used for the identification of compressor fluid-dynamic instabilities, i.e. stall and surge.

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