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Development of a research test rig for advanced analyses in centrifugal compressors

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Abstract

In this study, the design process of a new research test rig for centrifugal compressor stages is presented.
The rig has been specifically conceived for advanced analyses, with particular focus on rotating stall and in general on the operating conditions close to the minimum flow limit, which represent the research frontier in view of an extension of the stages rangeability.The new rig will be able to test industrial impellers at peripheral Mach numbers up to 0.7, operating in open-loop with ambient inlet conditions. A modular design will allow to test different stage configurations and then to carry out systematic optimization campaigns on a single specific component.
The conceptual design of the rig is here described and explained, including the selection of the best architecture and layout, the drivetrain assessment and the rotordynamic analysis.

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Keywords: centrifugal compressor; test rig; rotating stall; experimental analysis; rotordynamics.

1. Introduction

Centrifugal compressors are a key technology in the present energy scenario, particularly in the oil & gas compartment. Due to the huge amount of power connected to their industrial applications, even small efficiency and/or rangeability increases would in fact provide significant energy and money savings. In view of an extension of the minimum flow limit, in particular, manufacturers pay lot of interest in an inner comprehension of the phenomena which precede the surge, with particular focus on rotating stall.

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Due to the short time-to-market of industrial products, however, industry very often decided to demand this type of analyses to the academic world. After the fundamental studies on stall mechanisms which date back to more than fifty years ago(e.g. [1]), academia has undertaken different ways to analyze rotating stall, either with numerical approaches (e.g.[2]) or with experimental studies (e.g. [3-4]), mainly carried out in technical partnership with the industry itself, which was the only one able to provide realistic test cases and the adequate apparatus to test them. Academic test rigs fully dedicated to stall analysis are very few (e.g. [5]) and often scaled in dimensions and operating capabilities (flow coefficients and peripheral Mach numbers), giving results mainly referred to a single aspect (e.g. influence of a specific geometry).Moving from this technical background, the University of Firenze, in cooperation with *GE Oil&Gas*, decided to develop a new test rig fully dedicated to advanced flow analyses in centrifugal compressors. The rig is thought not to reproduce a typical industrial apparatus. Conversely, it is conceived as a flexible layout, able to test different stage configurations and to house several advanced measurement systems, in order to enable all those analyses which are not compatible with the standard industrial practice.

2. Rig concept design and layout

The main requisites of the test rig were first defined. The new facility was in fact intended to:

- ENABLENON-CONVENTIONAL ANALYSES, which often are not allowed by industrial rigs due to space or complexity issues, both in terms of test methodology and of measurement systems (e.g. PIV and other detailed flow analyses).
- BE FLEXIBLE, easily allowing the use of multiple stage configurations and/or new devices.
- SUPPORT THE ADVANCED DESIGN OF REAL INDUSTRIAL MACHINES, in order to transform each scientific discovery or improvement into tangible performance increases of industrial products.

In order to accomplish this latter goal, a technical survey on the *GE Oil&Gas*stall analysis database was first carried out [6]. Starting from preliminary limitations on the peripheral Mach numbers (up to 0.7) and on the geometrical proportions of the impellers (diameters up to 0.4 m), due to mechanical issues (discussed later on in the paper), industrial stages were catalogued in terms flow coefficient and power (e.g. Fig. 1a). By doing so, the rig capability was defined (Fig. 1b) in order to comprehend more than 60% of the industrial test cases with reasonable costs and installed power. It is worth remarking that the present rig cannot fulfill similitude conditions for the Reynolds number, as it operates with air at ambient pressure and temperature at the inlet. This technical drawback could indeed be overcome by making use of closed-loop configurations and/or working gasses with a higher molecular weight (e.g. CO_2). In the present application, however, the need of a versatile tool oriented the design to an open-loop configuration, also having the possibility of a diffuser freely discharging in open field (very useful in case of vaneless diffuser rotating stall analyses). In addition, Senoo et al. [1]. demonstrated that stall behavior is Reynold number independent for small ratios between the diffuser's width and the impeller radius.

2.1. Rig layout

When approaching the design of a non-conventional rig, the design possibilities were indeed very many.A literature survey revealed that some examples of research test rigs for these analyses did exist (e.g. [5,7]), even if characterized by different degrees of simplification.In particular, it was noticed that some of these rigs were constructed with a vertical layout, with the impeller discharging parallel to the floor. This layout provides some advantages (reduction of overall dimensions, easy access for optical instrumentation), but is unfortunately compatible only with reduced installed power and impeller peripheral speeds due to the problems induced by vibrations and structural stresses during operation. For these reasons, in the present application the conventional design layout having the axis of rotation parallel to the floor was adopted.Figure 1c reports a side view of the new test rig with its overall dimensions.

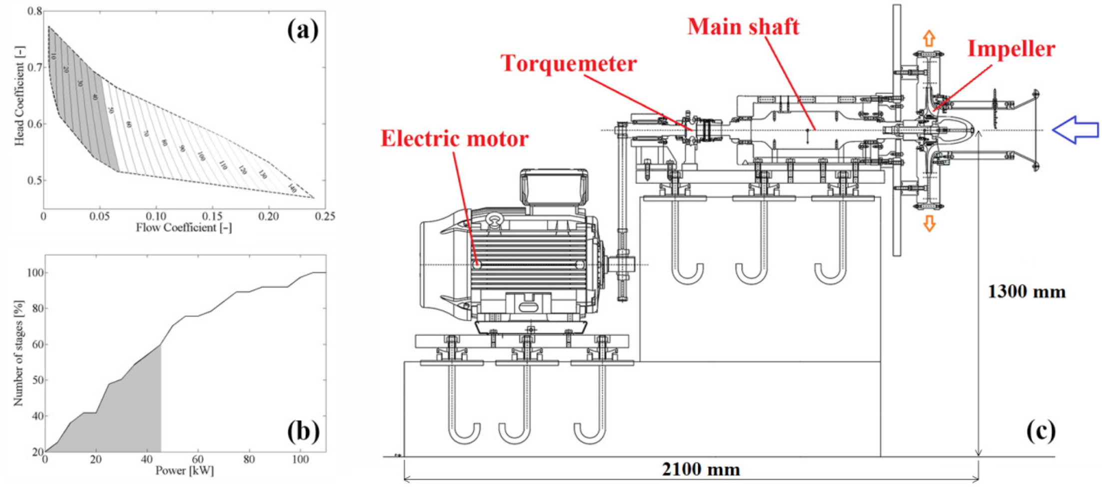

Fig. 1. (a) Rig power sizing based on the industrial database; (b) Testable stages as a function of power; (c) Test rig layout.

Upon examination of Fig. 1c some specific characteristics of the new rig are apparent. The impeller is fed by an axial duct (whose real length will be evaluated depending on the final geometry and operating conditions of the stage). In the simplest version (e.g. for vaneless diffuser studies), the stage will discharge in open field inside a shielded chamber, for which a proper air recirculation system will be provided. Additional configurations will also be available (see Sect. 2.2).The impeller is driven by a 45 kW electric motor through a high-velocity belt and pulleys system. This configuration, which represents a very common solution in industrial machines, is quite unusual for centrifugal impellers rigs, were the revolution speeds and the rated power are high. In view of a reduction of the system complexity and cost, however, it was here explored as it allows both a more rational use of space in the rig (the rotor train is split in two) and particularly the removal of a lubricated gear between the motor and the impeller. This device, often used in industrial rigs, is needed to match the revolution speed of the electric motor (max 3000 rpm) with the needed speed of the shaft (approximately 13500 rpm in the present rig). To do so, it absorbs a lot of additional power and also implies the presence of a dedicated lubricating system.

The present layout, conversely, ensures the required speed factor of 4.5 simply using pulleys, which, however, were custom realized (in cooperation with the producer) using high-characteristics steel and dynamically equilibrated. Moreover, the proposed layout, in which the electric motor is placed into a dedicated slide at a different height with respect to the main shaft, allows the operator to simply impose the proper tension to the belt by moving the motor orthogonally to the shaft.The main rotor train is again split into two shafts. The secondary shaft (on the left in Fig. 1c) connects the smaller pulley with a high-frequency torquemeter. A mechanical joint then connects the exit of the torquemeter with the main shaft, which was inserted into a special steel cartridge, hosting the bearings and several control sensors (thermocouples, vibration probes, tachometers). With the primary aim of ensuring the maximum available simplicity and cost-reduction, both shafts are supported by high-precision sealed ceramic bearings by SKF®. Ceramic rolling elements, much harder and stiffer than conventional steel elements, ensure good performance even at extremely high revolution speeds, avoiding however any kind of lubricating system. As shown in Sect. 3, the bearings were selected after a cross-analysis with the rotordynamic model of the system, ensuring a safe operation up to the maximum revolution speed of 13500 rpm.

Vibrations and mechanical stresses represented a major issue for this compact layout. Each component of the system has undergone a very accurate design process, comprehending both assessed design rules of the industrial partner and additional numerical analyses. In particular, the main shaft has undergone an iterative design process, in which structural FEM analyses made with ANSYS™ and rotordynamic simulations (see Sect. 3) have been carried out at the heaviest operating conditions to define the optimal mass distribution and bearing distance along the axis. The real rigidities (provided by the manufacturers) of both the torquemeter and the bearings themselves have been considered. The analysis ensured that, even if with the heaviest overhung impeller and the maximum revolution speed, the deformation of the shaft was small enough not to threaten the required tolerance of the bearings (ISO P4A).

For the same reasons, also the concrete platform on which both the motor and the rotor train lay on has been specifically designed by a civil engineer, both in terms of scaffold geometry and of concrete mixture. In detail, in order to make it resistant to the vibrations at multiple expected frequencies, special additives for the concrete have been added, i.e. a super-fluidizing powder (5%) made of submicronic particles of amorphous silica, a liquid admixture for reducing the hydraulic shrinkage (2%) and mono-filament polypropylene fibers to avoid cracks due to hygrometric shrinkage at both fresh and hardened stages. In addition, the whole platform has been set down on four supports in a special plastic material able to isolate the rig from the laboratory ground.Finally, to ensure a perfect alignment of the components, both the motor and the rotors were placed on steel plates, directly connected to the platform by means several tie-rods drowned in the concrete and each one laying on six semi-spherical supports to restore the planarity of the system, which was ensured within a tolerance of less than 0.01 mm/m.

2.2. Stage configurations

The base configuration of the system (Conf. A) is the one represented in Fig. 1, with the impeller overhanging the steel main plate, which will house the majority of fixed sensors along the flowpath, and discharging freely in the environment. The proposed solution was thought to represent the best layout for a variety of experimental measurements, with particular reference to optical systems (e.g. PIV, LDV, etc.), which can benefit from the free space in front of the stage to be placed and have proper optical accesses, which will be defined each time depending on the specific analysis and the stage characteristics.

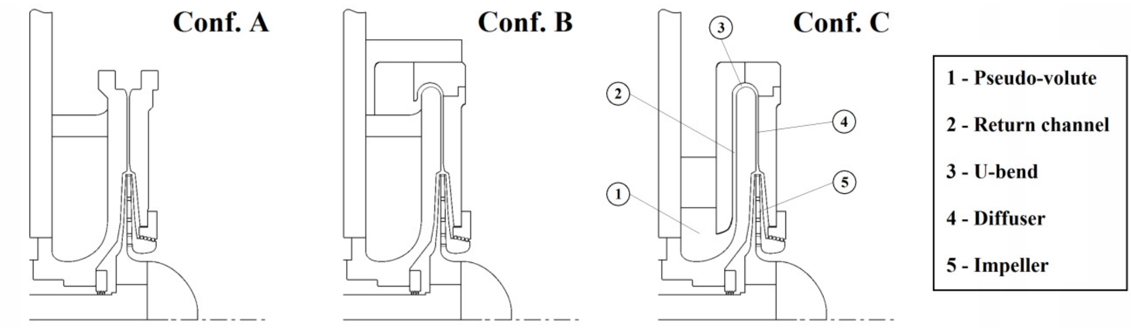

Fig. 2. Different stage configurations (simplified visualization to improve readability): (a) Open-field discharge of the diffuser; (b) Pseudo-volute after the U-bend; (c) Return channel and pseudo-volute.

Configuration A is quite unusual in industrial tests, but was here considered of major interest as it allows very significant test on vaneless diffuser rotating stall discarding any possible influence of the static parts downstream the diffuser. The open-field discharge in fact represents a perfect boundary conditions to be used in the tests, also in view of future comparisons between experiments and CFD simulations. In view of specific test campaigns devoted at investigating the stall mechanism and the influence of the stage geometry, interchangeable additional configurations were further designed (Fig. 2). In Conf. B the flow is discharged into a pseudo-volute having a squared section and positioned just after the U-bend. This particular solution puts focus on the effects of the U-bend on rotating stall. This component has been recently thought by the industrial partner to be somehow responsible of inducing perturbations affecting the inception and development of stall.

Finally, Conf. C represents a more conventional test flowpath also having a bladed return channel and discharging into a pseudo-volute. The use of Conf. C will be recommended in tests in which a closer match between laboratory tests and on-field performance is requested.

As a general remark, it is worth noticing that the proposed layout is able to modify the three configurations by changing only few components and with no detriment of the rotordynamic stability of the system, then obtaining that flexibility which represented one of the main goals of the design process.

2.3. Instrumentation and measurement systems

In addition to the torque and vibrations measurements made on the rotor, in any test different stations will be instrumented along the flowpath with a variety of sensors, summarized in Table 1.In detail, dynamic pressure measurements will be obtained with miniaturized piezoresistive sensors (Kulite® XTL-190M-5D), while small Pitot tubes, miniaturized three-holes probes, shielded Kiel probes and thermocouples will be used to define all the other parameters described in Table 1. Depending on the test, the rig will be also able to host additional advanced measurements systems, among which fast-response pressure probes, a PIV at the impeller outlet, a LDV system focused on the diffuser inlet.

Table 1. Measurements (number of sensors) at different sections along the flowpath and their circumferential positioning (brackets in degrees): h=hub side - s=shroud side - t=traversing - r=rake of multiple sensors (different radial positioning).

Section positioning	Static pressure	Total pressure	Total temperature	Flow Angle	Flow velocity	Dynamic pressure
Stage inlet	4x h+s[30 120 210 300]	3x r[75 195 315]	4x r[0 60 90 180]	3x t[75 195 315]	2x t[15 135]	2x s[0 90]
Impeller outlet			4x s[0 60 90 180]			
Diffuser inlet (parallel walls)	4x h+s[30 120 210 300]	3x t[75 195 315]	4x s[0 60 90 180]	3x t[75 195 315]	3x t[15 135 255]	4x s[0 60 90 180]
Mid diffuser						4x s[0 60 90 180]
Diffuser outlet	4x h+s[30 120 210 300]	3x t[75 195 315]	4x s[0 60 90 180]	3x t[75 195 315]	3x t[15 135 255]	4x s[0 60 90 180]
U-bend exit (Conf. B & C)	4x h+s[30 120 210 300]	3x t[75 195 315]	3x s[0 90 180]	3x t[75 195 315]	2x t[15 135]	2x s[0 90]
Return channel outlet (Conf. C)	4x h+s[30 120 210 300]	3x r[75 195 315]	3x r[0 90 180]	3x t[75 195 315]	2x t[15 135]	2x s[0 90]

2.4. Rotordynamic analyses

Rotordynamic analyses have been constantly carried out during the whole design process of the rig, both to dimension the components and to verify the correct selection of bearings and mechanical joints.

In particular, due to the decision of adopting high-precision ceramic bearings with no oil lubrication, the system became more rigid and therefore stricter requirements were needed to ensure the almost null plays required.Moreover, based on the fact the rig will be hosted in a research laboratory, safety reasons suggested to design the system so that the first critical frequency falls out of the operating range of the rotor. In industrial tests, this precaution is often bypassed as the system can quite smoothly pass through the first critical when accelerating. In the present case, however, in order to prevent any possible anomalous vibrations on the system, the system is expected to work within the stable range. To this purpose, however, many elements of the drivetrain were oversized with respect to a more conventional design.Once the design process was completed, a final check on the rotordynamic behavior was carried out using a numerical code [8] based on a finite elements approach, commonly used by the industrial partner for this type of studies. According to [6], the overhanging impeller wasmodelled as a lumped mass with no radial inertia considered (i.e. no gyroscopic stiffening). Both the torquemeter and the seals' effect were included in the simulations. For brevity reasons, additional details on the rotordynamic model and its assumptions can be found in Refs. [4] and [6]. The rotordynamic model confirmed that the rotor is subcritical for all the possible stage configurations, having a safe margin of 28% at the maximum revolution speed, fully compatible with both the industrial common practice and the API617 standard.

3. Conclusions

In the paper, the design of a new research test rig for centrifugal compressors is presented. The new rig is able to reproduce peripheral Mach numbers up to 0.7 and ensure the real flow coefficients of the testing stages. Based on the innovative design here proposed, different stage configurations are allowed by the rig with only few modifications in the experimental apparatus. The rig is free of any lubricating system and presents a drivetrain architecture extremely simple and low cost.In the near future, the rig will enter into operation and will hopefully represent a valuable new tool to achieve a better comprehension of several complex flow phenomena, including the inception and development of diffuser rotating stall.

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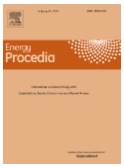

Biography

Lorenzo Ferrari received the MSc in Mechanical Engineering in 1999 and the PhD diploma in “Energy Engineering and Innovative Industrial Technologies” in 2003.Since 2011, he has been Researcher at ICCOM-CNR (National Research Council of Italy). His research fields are energy systems, renewable energies and measurement techniques applied to turbomachinery.