Test Rig Configuration for the Investigation of an Industrial-type Centrifugal Compressor Stage

Centrifugal compressors are an essential part of the industrial environment and, as such, are subject to the most stringent requirements in respect of operational efficiency in large parts of the compressor map. As part of FVV research project no. 1279 "Design and implementation of the FVV industrial compressor", a new compressor stage was designed as a research platform at RWTH Aachen University to provide the basis for investigating the stability limit, as well as interaction phenomena, in part-load operation.

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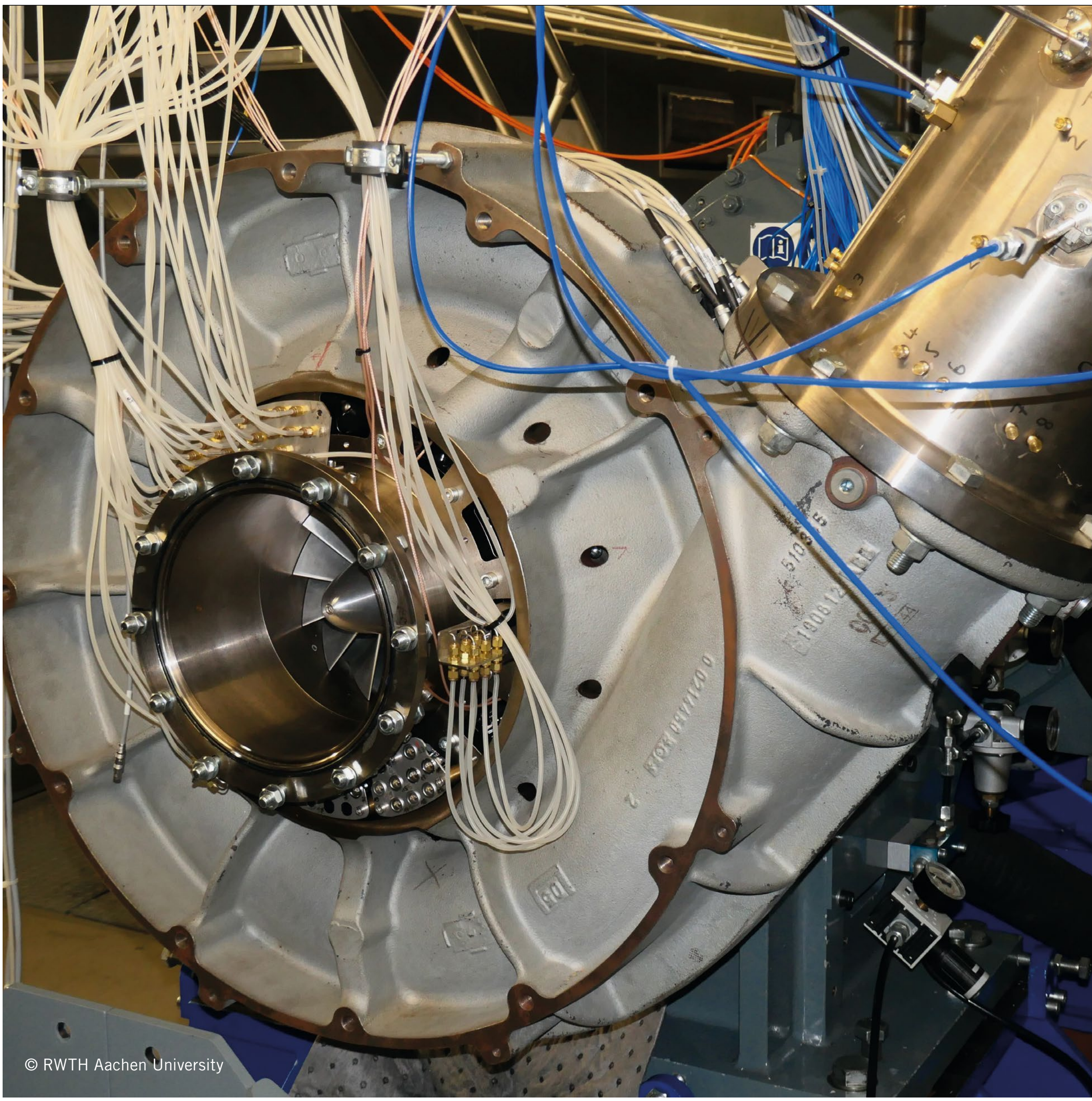

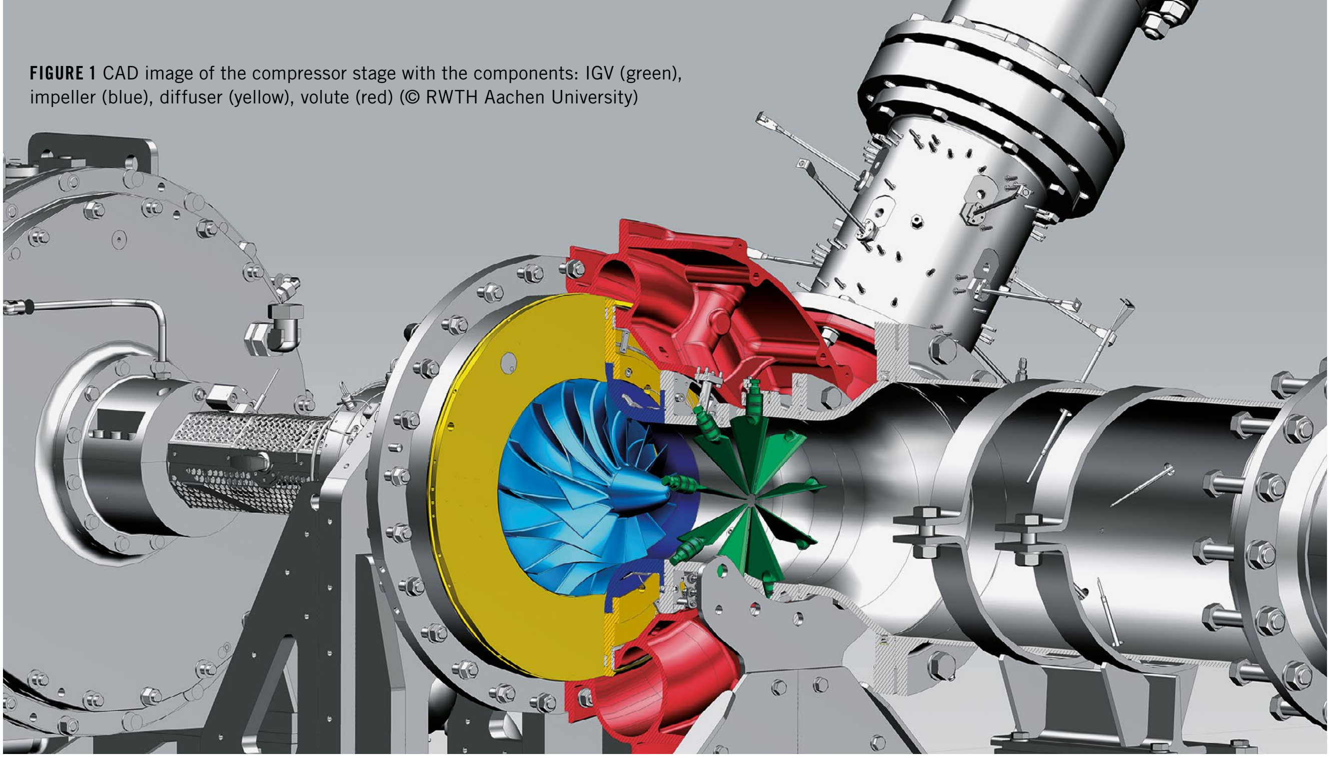

FIGURE 1 CAD image of the compressor stage with the components: IGV (green), impeller (blue), diffuser (yellow), volute (red) (© RWTH Aachen University)

1 MOTIVATION

At the Institute of Jet Propulsion and Turbomachinery (IST) of RWTH Aachen University, various centrifugal compressor stages have been investigated on a test rig which has been in operation since 2011. Centrifugal compressors are widely used in industrial applications and account for a not insignificant part of the energy demand. Efficiency can be achieved, among other things, by flexible operation throughout the whole compressor map, which requires a good understanding of part-load behavior. Industry-like centrifugal compressor stages can be examined with the test rig configuration described below.

2 DESIGN PROCESS

As with all compressor stages, the design process is interdisciplinary in nature for large engines, with a top-down approach. Using simple analytical 0-D models, the requirements are transferred into a specification sheet. The analytical procedures are supplemented by empirical models for estimating aerodynamic losses. After that, the essential variables of the compressor stage will become known. Based on this, the geometry is worked out in further detail before compressor maps are calculated by means of steady-state flow simulations. With their help, the most promising designs are selected and then subjected to a structural-mechanical analysis. The Finite Element Method (FEM) is used to check two essential criteria: First, the stresses must not exceed the material-specific limits; second, a check is made to ensure that the natural frequencies are outside the range of potential resonances, since disturbances or interactions with other components can lead to the excitation of blade vibrations. These calculations may necessitate modifications to the geometry, whereby the blade thickness can be used to influence this. Thus, it also changes the aerodynamic properties of the impeller blade, so that the maps must be recalculated. This iterative procedure is repeated until all specifications are met, which will be followed by integration into the overall test rig system, which then requires further safety-related investigations.

FIGURE 1 shows the final design consisting of the impeller, the vaneless diffuser and the volute; its design parameters are listed in **TABLE 1**. Furthermore, an Inlet Guide Vane (IGV) is installed in the suction line, which offers a further control variable for the compressor stage, in addition to the rotational speed of the motor and throttle position.

3 MEASUREMENT CONCEPT

The aim of this project is to design a new compressor stage, of which the prototype will then be examined in detail in a highly instrumented test rig. For the measurement of the compressor map, the stage must be balanced. On the one hand, the air mass flow is determined via an ISA 1932 inlet nozzle. On the other hand, total pressure and total temperature probes are installed at the compressor inlet (CI) and outlet (CO), which are supplemented by static wall pressure holes. This means that the thermodynamic condition is fully defined by the variables measured so that it is possible to calculate parameters, such as efficiency, pressure ratio and flow characteristics.

FIGURE 2 shows the positions within the compressor stage, which allow for a detailed measurement of the stage. Lines represent a distribution of measurement positions in meridional direction, whereas symbols represent a distribution in circumferential direction. For example, position (a) in **FIGURE 2** represents the measurement of the circumferential pressure distribution at different positions. Here, it can be recorded to what extent the volute influences the pressure field over the circumference and how this propagates upstream. The meridional pressure build-up across the compressor stage, **FIGURE 2** (b), can be determined with the use of wall pressure holes opposite the volute tongue. Access points for pneumatic probes allow intrusive measurements of the flow: Both the impeller inflow, **FIGURE 2** (c), as a function of the

Geometric boundary conditions	
Impeller outlet diameter	$D_2 = 330\ mm$
Diffuser outlet diameter	$D_4 = 570\ mm\ \left(\frac{D_4}{D_2} = 1.73\right)$
Diffuser width	$b_4 = 21\ mm$
Impeller outlet blade metal angle	$\beta_{2,S} = 120°$
Blade count	$Z = 14$
Aerodynamic boundary conditions	
Design rotational speed	$N_{AP} = 21{,}992$ rpm
Design mass flow	$\dot{m}_{AP} = 5.1\ \frac{kg}{s}$
Design flow coefficient	$\varphi_{AP} = \frac{4\dot{V}}{\pi D_2^2 u_2} \approx 0.13$

TABLE 1 Geometrical and aerodynamic design parameters of the compressor stage (© RWTH Aachen University)

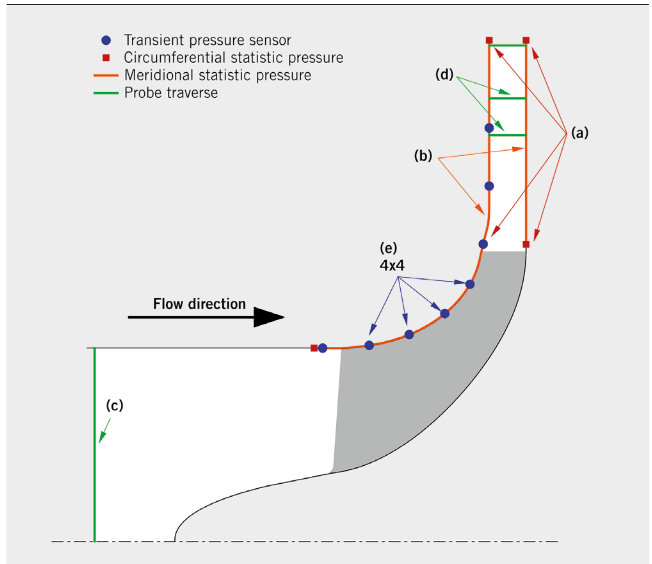

FIGURE 2 Schematic representation of the measurement concept for detailed examination of the test rig (© RWTH Aachen University)

IGV angle as well as the diffuser flow, **FIGURE 2** (d), in the channel height direction can be measured. Transient piezo-resistive pressure sensors are installed, **FIGURE 2** (e), so that the stability limit of the compressor stage can be investigated. In particular, the measuring field at four circumferential positions each with four sensors distributed equidistantly over the meridional direction should be mentioned here. This arrangement makes it possible to detect and clearly identify rotating stall cells. This flow phenomenon is characteristic for the stability boundary.

–	Unit	EXP	CFD	|Δ|	Measurement uncertainty
Efficiency $\eta_{s,tt,CO}$	%	82.0154	–	–	0.7917
Pressure ratio $\pi_{tt,CO}$	-	2.3685	–	–	0.0150
Efficiency $\eta_{s,ts,DO}$	%	73.9243	73.9952	0.0709	0.4896
Pressure ratio $\pi_{ts,DO}$	-	2.1945	2.1878	0.0068	0.0068

TABLE 2 Experimental-numerical comparison of the design point at N_{red} = 21,992 rpm and $\dot{m}_{red}$ = 5.1 kg/s (© RWTH Aachen University)

4 MEASUREMENT RESULTS

At the design point, the compressor stage achieves a total-to-total efficiency of $\eta_{tt,\,CO}$ = 82.02 % with a total-to-total pressure ratio of $\pi_{tt,\,CO}$ = 2.37. Since the volute was not taken into account during the design process and so the extent of the losses of this

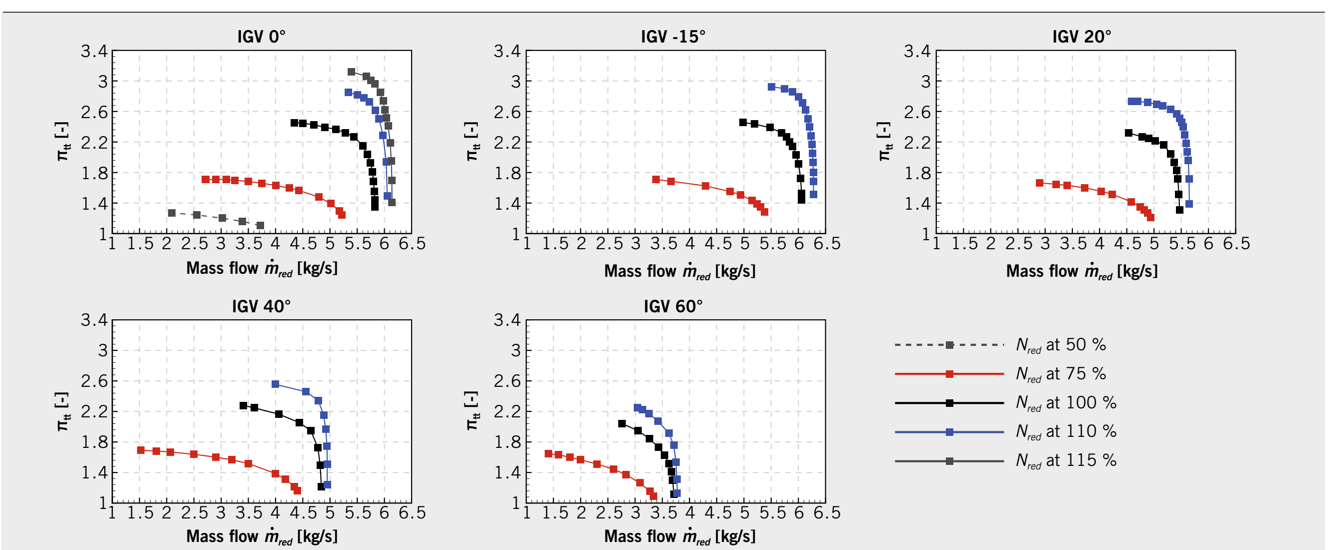

FIGURE 3 Total rotational speed and pre-swirl dependent performance map showing the pressure ratio $\pi_{tt,\,CO}$ as function of the reduced mass flow $\dot{m}_{red}$ (© RWTH Aachen University)

component are not known, a comparison of the efficiency at the Diffuser Outlet (DO) is useful. In this plane, only total-to-static performance values are available from the experiment. For these total-to-static values there is a good agreement between the numerical simulation (Computational Fluid Dynamics, CFD) and the experiment (EXP), with the deviations only slightly exceeding the measurement uncertainties in the experiment, **TABLE 2**. A comparison of the meridional pressure buildup also shows a good agreement between the simulations and the experiment over large parts of the compressor map.

The performance maps shown in **FIGURE 3** depict the expected behavior for a centrifugal compressor. As the rotational speed rises, an increasingly pronounced choke limit develops, which is also predicted very well by the CFD. At lower rotational speeds, the map is limited by the system resistance even before the choke limit. By setting a negative pre-swirl in the impeller inflow, the mass flow, the pressure ratio and thus the required power increases. This is determined by Euler's turbine equation, which can be used to estimate the required work input P_{therm} in a stage, Eq. 1:

Eq. 1 $$P_{therm} = \dot{m}\Delta h_t = \dot{m}\omega(c_{u2}r_2 - c_{u1}r_1)$$

If the test rig is operated without pre-swirl (c_{u1} = 0 m/s), the second term vanishes. On the other hand, with negative pre-swirl ($sgn(c_{u2}) \neq sgn(c_{u1})$), the second term increases the work input, such that the required power increases. The maximum measured mechanical power consumption of the test rig is 821 kW, whereas the measured mechanical drive power at the design point is 551 kW. Overall, this results in a performance map which has a maximum pressure ratio of $\pi_{(tt,max)}$ = 3.12 and a mass flow ranging between a minimum of 1.41 kg/s and a maximum of 6.28 kg/s.

5 UPCOMING RESEARCH PROJECTS

With the completion of the project, a research platform is available on which aerodynamic and acoustic investigations on an industrial compressor can be carried out. In one upcoming project the stability limit will be investigated. To this end, the measurement concept offers excellent possibilities for validating existing prediction models, such as those of Senoo et al. [1], and for investigating the corresponding flow phenomena in detail. In another project, it will be possible to investigate the component interaction. For example, the inflow into a volute matches optimally to the rest of the stage at only one operating point per speed line; at every other operating point, a circumferentially inhomogeneous pressure field is induced by the volute, which propagates into the impeller. Each individual passage will therefore be subject to a different backpressure, depending on the circumferential position, resulting in additional transient losses. This loss mechanism is illustrated in **FIGURE 4**. With a vaned diffuser already designed and still to be investigated in another project, a direct comparison can be made to the configuration with a vaneless diffuser. Furthermore, an exchange of the volute is planned, so that the test rig setup in this configuration will include only self-developed components – the FVV member companies thus have all measurement data and geometries available as an open test case for the validation of their own prediction tools. A novel acoustic measurement concept has already been established on a previous test rig setup to determine the acoustic power emitted into the discharge line and this is now being used on the industrial compressor [2]. In this way, the influence of the individual components on both acoustics and aerodynamics can be examined.

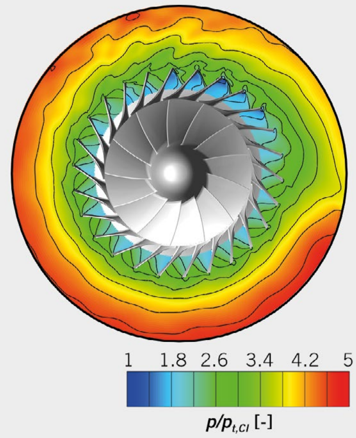

FIGURE 4 Volute-induced circumferentially inhomogeneous pressure field (© RWTH Aachen University)

6 SUMMARY

The performance map of the industrial centrifugal compressor stage designed and commissioned shows good agreement with the prediction, so that the test rig set up is suitable as a research platform for future projects. A follow-up project will aim to investigate the stability limit of the compressor stage and another one the component interactions both aerodynamically and acoustically, so that a better matching of the components can be achieved.

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