DESIGN AND TESTING OF AN ISO 5801 INLET CHAMBER TEST RIG AND RELATED ISSUES WITH THE STANDARD

Stefano CASTEGNARO[1], Massimo MASI[2],
Andrea LAZZARETTO[1]

[1] *University of Padova, Department of Industrial Engineering,
via Venezia 1, 35131 Padova, Italy*

[2] *University of Padova, Department of Management and Engineering,
stradella San Nicola 3, 36100 Vicenza, Italy*

SUMMARY

A rig has been designed and constructed according to the ISO 5801 Standard, to test fans with a maximum inlet diameter equal to 0.8 m. The rig design procedure, that turned into an effective manufacturing and installation, is described. Issues that emerged from the preliminary study of the 5801 Standard are highlighted and discussed. Suggestions to improve the Standard are proposed.

INTRODUCTION

This paper describes the design of an inlet-chamber fan test rig coupled with a multi-nozzle system for flow-rate measurements, providing a procedure for the rig dimensioning and a discussion on some issues and ambiguities encountered within the ISO 5801 Standard.

By April 2018 the new ISO 5801 Standard [1] shall be adopted as national norm in every European country, superseding the previous 2008 version [2]. The new Standard is the latest achievement of a long and complex process of international standardization, which began in 1963 collecting recommendations from several previously existing national fan testing codes [2, p. viii]. Likely because of the complexity involved in this exercise of synthesis, some of the preceding versions of the *Industrial fans-Performance testing using standardized airways* codes were not so simple to read and use. Wallis, in fact, claims that preceding fan test codes came out from a "*piecemeal process*"[3]. In past years, the use of the Standard even required additional documents specifically addressed to the user (e.g., [4]). The latest version of the Standard [1] results from a remarkable effort of simplification and improvement. In spite of this enhancement, however, some issues still remain from the perspective of a rig designer:

a) even if the requirements of all the components and rig parts are identified, a direct rig design procedure is not available in the Standard or in the literature;

b) some issues and incoherent informations exist within the Standard.

The second item, *b)*, is particularly tedious for the rig designer, because issues and ambiguities translate into a deeper analysis of the flow phenomena and the consequent interpretation of the Standard. This increases the time required for the design process and, in turn, increases the costs.

In 2017 a rig was designed and constructed according to the ISO 5801:2008 code at the University of Padova, to test fans with a maximum inlet diameter equal to 0.8 m. The rig features an inlet chamber configuration and a multi-nozzle system for the flow-rate measurements. The facility was put into operation in July 2017 and is currently operating both for research and didactic purposes.

In this work, parts of the technical and design achievements acquired from that experience are collected. In the first part of the paper, the design procedure used for the rig is presented: the procedure resulted in an inexpensive design that was easy to assemble. The operative differences on rig-chamber dimensioning between the new ISO 5801:2017 Standard and the previous 2008 version are highlighted. In the second part, some issues and inconsistencies encountered within the Standard are highlighted and discussed.

This paper is addressed toward fan testing rig designers and users. Due to the technical matter of the discussion, previous knowledge of the Standard is required. In particular, all the symbols and nomenclature are coherent with references [1, 2, 5].

The aim of the paper is twofold: *i)* provide a direct procedure that will result in a satisfactory design of an inlet chamber test rig, and *ii)* opening a discussion on some issues contained within the ISO 5801 Standard and suggesting possible corrections/improvements.

RIG DESIGN

Rig Configuration Selection

According to preliminary research performed on existing rigs, fan manufacturers appear to prefer the chamber rig configuration (i.e., the A type [1]), instead of ducted ones (i.e., B,C,D layouts [1]). One reason that makes this configuration valuable for the industries is that chamber rigs allow for testing fans with widely different sizes (e.g., [6, p. 354]) without substantial rig modifications.

Ideally, the selection of the *inlet* or *outlet* chamber configuration provides no difference in terms of assessing the fan performance. However, for the outlet-configuration layout the flow-field within the chamber is complicated by the fact that the fan-exhaust-jet kinetic energy must be absorbed by the flow-settling means upstream of the measurement planes and this results in some recirculation at the jet boundaries.[1] However, similar conditions might also occur for an inlet-chamber rig that features a multi-nozzle flow-measurement system, if a high-velocity jet is exhausted by some of the nozzles.

Both Inlet and Outlet test rigs were originally installed in the *Laboratorio di Macchine Aerauliche e Termiche* at the University of Padova (see Fig. 1), with the first employed mainly with 315 mm axial fans (e.g.. [7]) and the second with smaller cross-flow fans (e.g., [8]). The inlet chamber layout was lastly selected in continuation with an on-going research on axial fans.

Design differences exist from the structural point of view, as walls are subject to a negative pressure (i.e., they are *squeezed* toward the inside) in the inlet chamber case, while in the outlet chamber case

[1]The 2017 ISO Standard [1, p. 26] requires that tests are performed to verify that such recirculation is not excessive at the fan static pressure measurement plane in the outlet chamber case.

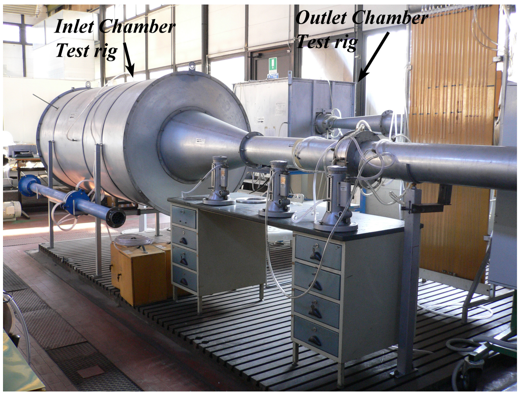

Figure 1: Original Inlet and Outlet Chamber Test rigs.

walls are *inflated*, and therefore pushed toward the outside (further insights are reported in the Section on Structural Design).

A rectangular chamber layout is considered, with h and b identifying the height and breadth of the chamber section, respectively. In large installations the rectangular layout is generally preferred with respect to the circular one, because of: *i)* an easier construction and assembly (the large planar panels that compose the rectangular section can be easily transported, while circular cylinders are usually manufactured in a single piece, with increased transport complexity); *ii)* ease of access and operation inside the chamber for any technician or experimenter.

Flow-rate Measurement System

The rig configuration selection is also related to the flow-rate measurement system. The rigs in Fig.1 feature in-line orifice-plate systems for the flow-rate measurements. When the flow-rate is lower or larger than the measurable range the orifice plate must be changed and this operation is time-consuming. A relatively wide range of measurable flow-rates ($0.325 \leq q_v \leq 8\, m^3/s$) was required for the new rig. Thus, a multi-nozzle system appeared as most appropriate, due to the rapidity that this solution allows in shifting from a nozzle (or a nozzle-combination) to another of different size or number. A *type 3* inlet chamber is indicated in the 2008 version of the ISO Standard [2, p. 114] for using a multi-nozzle system; thus, the *type 3 Inlet Chamber* was chosen as final configuration for the new test rig.

Definition of the Main Dimensions

The design of the rig is an iterative procedure that starts with the definition of the main dimensions. The facilities for aerodynamic tests are generally quite voluminous: as larger volumes involve higher costs (space$\simeq$money), the test rig should be as small as possible, while permitting tests on larger fans.

The definition of the transversal and longitudinal dimensions of an inlet chamber rig with a multi-nozzle system (type 3 chamber [2]) is parametric in terms of:

- the largest fan inlet diameter ($D_{1_{max}}$) that is required for the tests;
- the throat diameter of the largest nozzle (d_{max}).

Sizing of the Transversal Section

The starting point of this design procedure requires the assumption of $D_{1_{max}}$. Such value normally comes from the fan manufacturer's need to test fans up to a specific size. The Standards [1, 2] require the area of the chamber measurement section A_3 to be at least 5 times the inlet area A_1 of the fan under test, thus:

$$A_3 \geq 5 \cdot \frac{\pi}{4} D^2_{1_{max}} \tag{1}$$

where $A_3 = h_3 \cdot b_3$. Equation 1 allows a preliminary selection of the height and breadth of the chamber section. In addition, the 2017 Standard [1, pp. 22; 29] imposes two constraints on the dimensions of h and b:

$$\frac{2}{3} < \frac{b_3}{h_3} < \frac{3}{2} \tag{2}$$

$$b_3 \text{ or } h_3 \geq 2 \cdot D_{1_{max}} \tag{3}$$

Selecting values of h_3 and b_3 that satisfy Eq.s 1, 2, and 3 allows the preliminary sizing of the chamber section according to the 2017 Standard [1], which removed the distinction among *type 1, 2 and type 3* chambers that was present in previous standards (not only in the 2008 version [2] but also in the 1998 one). What operationally distinguishes one chamber type from another is unclear and not explained. Furthermore, additional dimensional constraints were imposed on h_3 and b_3 for the *type 2* chamber [2, p. 110], while were not for the *type 3* solution. Similar issues exist for the sizing of the rig length, which is the object of the following sub-section.

Sizing of Longitudinal Dimensions of the Rig

Figure 2 (or the equivalent one from [1]) is the key to perform the exercise of sizing the rig longitudinal dimensions. The fore part of the rig (i.e., the part at the left of the multi-nozzle system) is dimensioned according to Fig. 2 and does not present particular issues, while the minimum longitudinal overall length l_{min} of the chamber downstream of the multi-nozzle plane is given by:

$$l_{min} \geq \text{largest nozzle length + nozzle-settling means buffer zone +} \\ \text{settling means axial length + settling means-endingwall chamber minimum distance}$$

According to [1, p. 26], all these longitudinal dimensions are parametric in terms of the *hydraulic* diameter of the chamber ($D_3 = D_h = \frac{4 \cdot A}{2 \cdot (b+h)}$), whose value derives from the transversal section sizing, and of the throat diameter of the largest nozzle d_{max} (see Fig 2). Note that, although not strictly necessary, it might be advantageous to allow for some space within the chamber to accommodate any inlet-side equipment (e.g., an inlet-drive motor). Thus, the previous formulation changes into:

$$l_{min} \geq \text{largest nozzle length + nozzle-settling means buffer zone +} \\ \text{settling means axial length + settling means-inlet-side fan equipment minimum distance +} \\ \text{inlet-side fan equipment axial length.} \tag{4}$$

For simplicity, the axial length of the inlet-side equipment can be assumed as a multiple of the inlet diameter of the fan (i.e., $= k \cdot D_{1_{max}}$, where k is a suitable constant). According to the 2017 Standard [1], Eq. 4 translates into:

$$l_{min} = 1.6 d_{max} + 2 d_{max} + \text{settling means axial length} + 0.5 D_3 + k \cdot \frac{D_3}{\sqrt{5}} \tag{5}$$

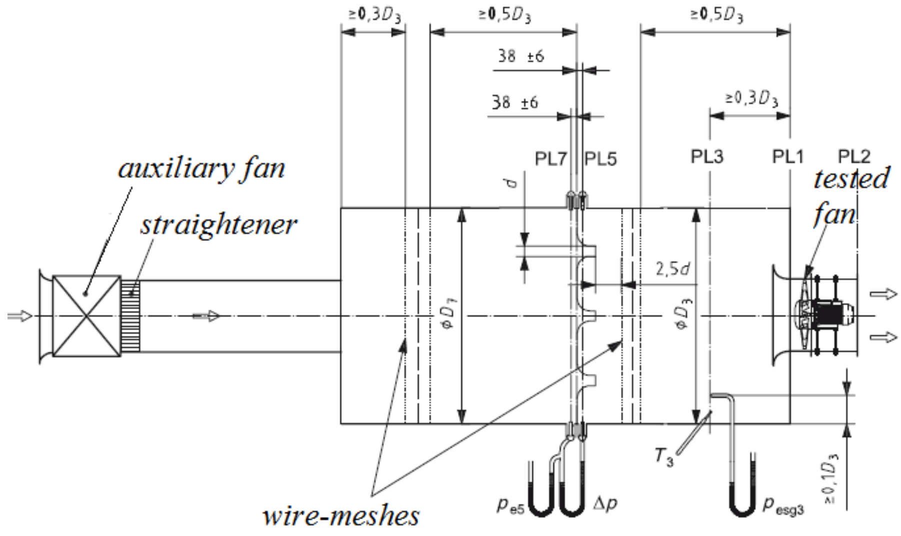

Figure 2: Axial dimensions of the A-installation type 3 test rig (adapted from Fig. 40e in [2]).

where the nozzle length (i.e., $1.6d_{max}$) is to be chosen in agreement with the length-to-diameter ratio [1, pp. 75-76]. The previous version of the Standard [2] required a larger distance between nozzles and settling means ($2.5d_{max}$, see Fig. 2) than the new Standard [1] ($2d_{max}$), and defines a minimum distance ($= 0.1 \cdot D_h$) between wire meshes or screens [2, p. 112]. Furthermore, differently from [1], a distinction between chamber diameter D_3 and hydraulic diameter D_h is reported in [2]. Accordingly, the analogous formulation of Eq. 5 for the 2008 Standard (that was used for the new rig design) is:

$$l_{min} = 1.6d_{max} + 2.5d_{max} + (0.2D_h + 3 \cdot th_w) + 0.5D_3 + k \cdot \frac{D_3}{\sqrt{5}} \tag{6}$$

where th_w is the wire or screen-plate thickness.

Note that Eq. 6 includes the 2008 Standard distinction between the hydraulic diameter and D_3. In fact, the 2008 Standard [2, p.114] computes the D_3 diameter with different formulas according to the type of chamber:

$$D_3 = \sqrt{b_3 \cdot h_3} \qquad \text{type 2 chamber} \tag{7}$$

$$D_3 = \sqrt{\frac{4(b_3 \cdot h_3)}{\pi}} \qquad \text{type 3 chamber} \tag{8}$$

It is evident that Eq. 8 provides a D_3 that is $\sqrt{\frac{4}{\pi}} \simeq 1.13$ times the value given by Eq. 7, at equal h_3 and b_3. Thus, the length of the *type 3* inlet chamber is slightly longer than the *type 2* chamber.

The measurement section 3 is to be suitably positioned in the chamber, between the settling means and the first part of the fan inlet-side equipment (see Fig. 2). The knowledge of d_{max} is required to size the test rig; however, the design of the multi-nozzle system is a lengthy procedure which has to account for many requirements (e.g., the accuracy of the pressure sensors, minimum and maximum flow-rates, continuity of measure within the entire flow-rate range, etc..). It is not reported here because it goes beyond the scope of this paper. For a preliminary chamber sizing, a reasonable value of d_{max} can be chosen (e.g., from similar existing installations), accounting for a limited safety margin.

The transversal and longitudinal sizing of the chamber is thus concluded and the procedure can proceed to the structural design and verification.

Structural Design and Verification

The structural design of the rectangular negative-pressure chamber turned out to be more complex than the corresponding positive pressure chamber. It was found that there is less interest in rectangular negative-pressure assemblies in the structural literature than the pressurized case. The information reported in this Section can mitigate this challenge.

A simple but effective structural layout has been suggested for the new rig: a *primary* metallic frame forms the skeleton of the rig, on which the stiffened panels are positioned (see Fig. 3). From the

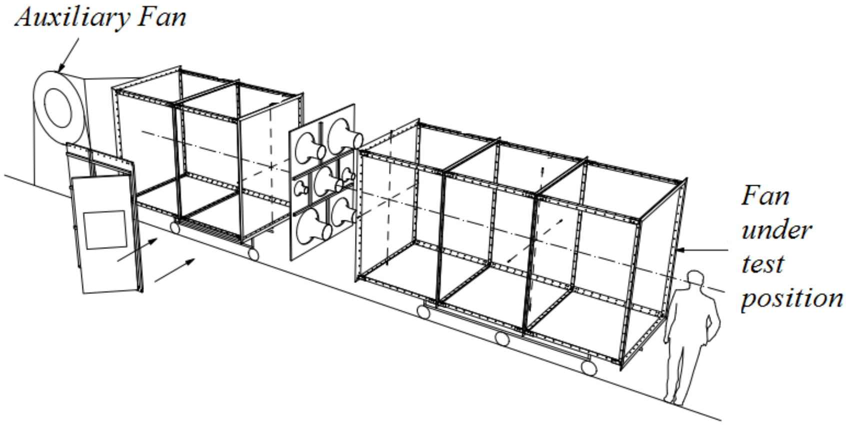

Figure 3: The metallic frame of the rig that carries the main part of the load and on which the stiffened panels are installed.

design perspective, this layout features the advantage of separating the duty required to the different structural components:

- the main loads (e.g., the *squeezing* action of the pressure) are carried out by the structure along the beams' longitudinal direction;
- the panels (i.e., the chamber walls) transfer the pressure load to the metallic structure. Thus, they are only required to be stiff enough to maintain the cross section dimensions b and h within a given tolerance (see in the following).

While the inlet chamber works almost exclusively under negative pressure loads, some attention must be provided to the upstream part of the rig (the left side in Fig. 2, ahead of the multi-nozzle system) that can experience both negative and positive pressure (it is positive when the auxiliary fan is in operation).

Frame Verification

The frame must be verified under the maximum static load (e.g., the maximum achievable fan static pressure, 10000 Pa in our case). In particular, each beam has to be verified against stresses and instability:

- $\sigma_{max} < \nu \cdot \sigma_{yi}$
- $P_{max} < \nu \cdot P_{crit}$

where $\sigma = \frac{P}{A}$ is the longitudinal stress at the beam section, σ_{yi} is the yield stress of the material, ν is a suitable safety factor and P_{crit} is the Eulerian beam instability limit. Suitable references on the topic can be found in structural handbooks (e.g., [9]). Particular attention must be provided to the verification of the junctions among the different beams, as these intersections carry both normal and longitudinal loads.

Sizing of the Stiffened Panels

The panels must be stiff enough to avoid relevant modifications of the chamber section at the maximum load condition. The Standard [2, p.26] provides a tolerance on the duct diameter of $\pm 0.01D$. Thus, a quarter of such value (i.e., $0.0025 \cdot D_3$) may be taken as preliminary reference for the maximum panel deformation, as all the chamber sides must be taken into account.

The maximum deflection w_{max} of a squared *simply supported* panel subject to an uniformly distributed load (i.e., a pressure Δp) occurs at the center, and is given by [9, p.234]

$$w_{max} = 0.0443 \cdot \Delta p \frac{a^4}{E \cdot th_p^3} \tag{9}$$

where E is the Young's Modulus of the material, a is the panel side length, and th_p is the panel thickness.[2] At first, in sizing the panel a suitable stiffener position is to be chosen, so defining the side length a. Secondly, the minimum panel thickness can be computed from Eq. 9 as

$$th_{p_{min}} = \sqrt[3]{0.0443 \cdot \Delta p \frac{a^4}{E \cdot w_{max}}} \tag{10}$$

The panels that compose the walls of the rig are made of 25 mm poplar plywood, stiffened with 30 mm okumè plywood (see Fig. 4). Further minor verifications of the panels deal with shear stresses at the bolt positions and at the supporting edges (i.e., where the panel is installed on the metallic frame).

The panel dimensioning concludes the structural design process.

ISSUES RELATED TO THE ISO 5801 STANDARDS

This part of the paper highlights some issues and inconsistencies of the Standards that result in extra work for the rig designer, involving a deeper analysis of the related flow phenomena and a consequent interpretation of the Standard.

The main problems encountered during the study that preceded the design of the rig include:

1) the operational difference between *type 1-2* and *type 3* chambers in the 2008 Standard [2, p. 112] appears unclear (as a matter of fact, such distinction has been removed from the 2017 Standard). Beyond the dimensional differences previously mentioned, a second unclear issue is associated with the total pressure measurement within the type 3 chamber. In particular, it is not clear the reason why the previous version of the Standard required the mandatory use of the Pitot-tube, whereas the present version admits also the use of wall pressure tappings.

[2] The general case for the rectangular plate is reported in [9]. Equation 9 is obtained assuming the Poisson ratio of the material equal to 0.3

2) Both the previous and the new Standard are ambiguous about the positioning of the nozzles with respect to the chamber axis: they do not clarify if a symmetrical allocation (with respect to the chamber longitudinal axis) is mandatory (and why) or only recommended.

3) Regarding the surface tolerance of the nozzles, the Standards impose an unclear *peak-to-peak* limit on the surface waviness.

For each of the previous items, some insights deriving from the rig design experience are reported and some improvements are proposed at the end of the Section.

Chamber Type and Use of the Pitot-Tube

The distinction among chamber types was not explained in the previous versions of the ISO 5801 Standard [2, 10]. It is possible that it was still a remnant of the standardization process from the different national codes and, as a matter of fact, has been removed from the latest Standard version [1]. Operative differences that were distinguishing the *type 3* chamber from the other types were: *i)* the allowance of using a multi-nozzle system for flow-rate measurements [2, pp. 114-115], and *ii)* the mandatory use of a pitot-tube to measure the air total pressure within the chamber [2, p. 125]. Although the new Standard has removed the imposition of the Pitot tube, allowing the use of wall static pressure taps within the chamber [1, p. 27], the issue still deserves a brief discussion.

The reasons for imposing the use of the pitot probe to measure the air total pressure p_{t3} within the chamber were likely to accommodate: *i)* possible inaccuracies in the area section A_3 due to the rig manufacturing and assembly process (that have to be considered when dealing with the typically large dimensions of such rigs), and *ii)* possible leakages from the chamber walls that would increase the flow-rate flowing through the measurement section 3 with respect to the q_v measured by the multi-nozzle system. In fact, when wall pressure tappings are used, the total pressure inside the chamber is computed as:

$$p_{t3} = p_3 + \frac{1}{2}\rho_3 \left(\frac{q_v^2}{A_3^2} \right) \tag{11}$$

where p_3 is the average static pressure measured at the taps. The computation of the dynamic pressure term in Eq. 11 is subject to uncertainties on both the flow-rate q_v and on the area A_3. Leakages from the chamber walls, when present, shall be identified and minimized (see for instance the *Chamber Leakage Test Procedures* in [2, p. 211]). However, uncertainties on A_3 are quite difficult to be treated, due to the large dimensions of such assembled structures working under different load conditions. Such difficulties were likely the reasons to make the pitot solution mandatory in the previous versions of the Standard. On the other hand, a pitot probe might provide erroneous measurements in presence of non-uniform velocity profiles within the chamber, caused by ineffective flow-settling means.

The new rig includes both the Pitot-tube sensor and the wall-pressure tappings at the measurement section, in agreement with [1]. Preliminary comparative tests between the two measurement methods performed on a 800 mm vane-axial fan running at ~ 800 rpm (see Fig.s 6) provided negligible differences in the measured quantities (i.e., differences in fan pressure $p_f \sim 1$ Pa).

Symmetrical Positioning of the Nozzles

The ambiguity regarding the positioning of the nozzles for flow-rate measurements is the most relevant, being present in the previous Standards [2, 10] and in the new one [1] as well.

At p. 69 of [2] and similarly at p. 75 of the newest code [1] is reported:

"For tests in standardized airways, multiple nozzles shall be used within inlet or outlet

> *chambers. The nozzles may be of varying sizes but* ***shall be*** *symmetrically positioned relative to the axis of the chamber, as to both size and radius."*

On the contrary, [2] at p. 112 reports:

> *"Multiple nozzles shall be located as symmetrically* ***as possible****"*

thus allowing for a possible non-symmetrical allocation. The same ambiguity persists in the new Standard [1], which shows a non-symmetrical allocation of the nozzles in the figure at page 132.

Apart from such incoherent indications, mostly important is the reason for requiring such symmetrical nozzle positioning (that is not explained in the norms). The most obvious reason for requiring a symmetric allocation is to avoid unbalanced flows within the downstream part of the chamber. However, this occurrence implies not only the symmetric *geometrical* allocation of the nozzles (both in size and radius) but also the simultaneous use of opposite nozzles. This means that equal nozzles at opposite positions with respect to the chamber axis shall be used simultaneously, to increase the uniformity of the velocity profile approaching the flow-settling means. However, the authors are aware of fan manufacturers currently using rigs with non-symmetrical nozzle allocation and non-symmetrical operating (i.e., opened) nozzles.

Evaluating whether the non-symmetrical use of the nozzles might affect the measurements requires a demanding systematic investigation, which is not currently available to authors' knowledge. Striving to ensure a uniform flow at the measurement plane, with the new rig the nozzles:

1. have been positioned symmetrically (to both size and radius) with respect to the axis of the chamber (see Fig. 4*a)*;
2. are always opened or closed pair by pair (except for the single central one) during a test.

a) *b)*

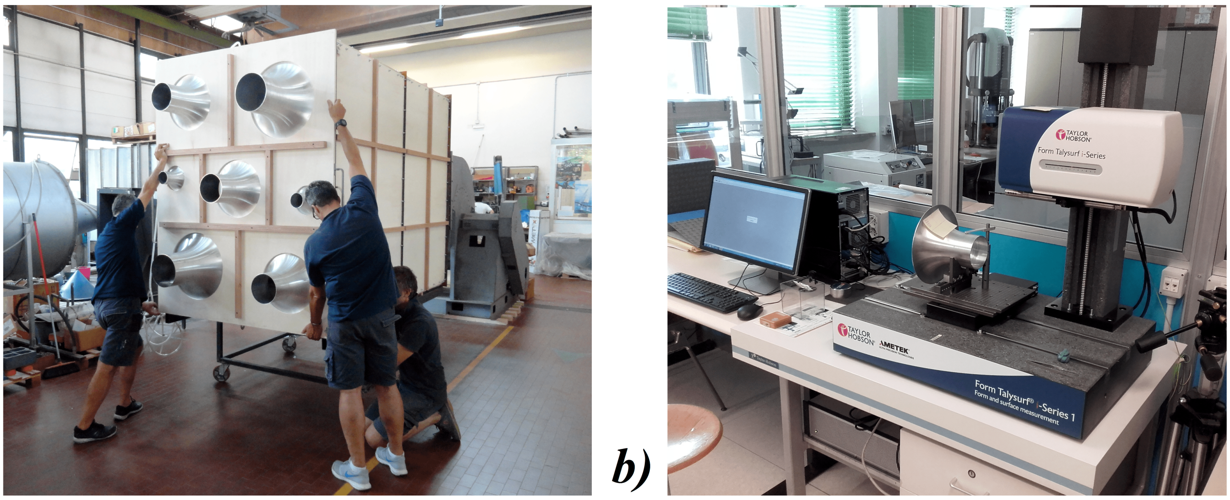

Figure 4: a) *The installation of the multi-nozzle wall featuring the symmetrical allocation.* b) *Metrological tests to determine whether the nozzle's surface is in compliance with the tolerances.*

Nozzle Surface Waviness Requirement

The third issue on the list deals with a *technological* aspect of nozzle manufacturing: the allowed tolerance on the surface waviness for the nozzle's interior side. As reported previously, the Standards

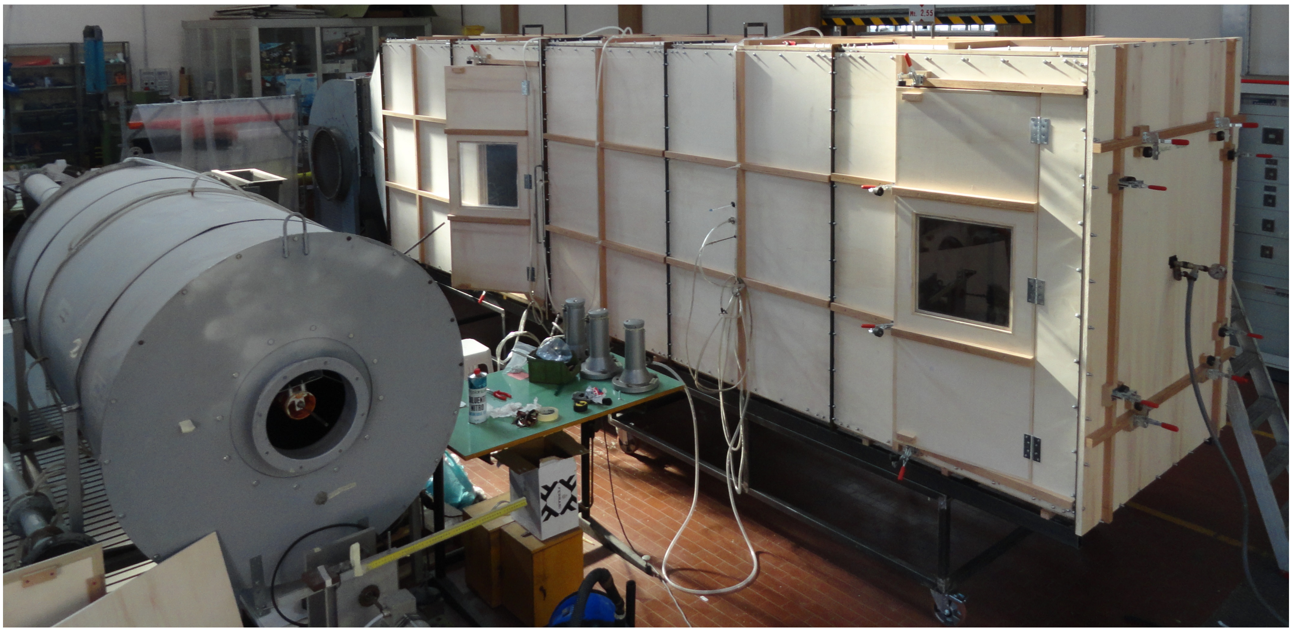

Figure 5: A pressure-decay test performed on the new rig. On the left, the previous smaller inlet-chamber rig is visible.

[1, 2, 10] limit the maximum *peak-to-peak* distance for the surface waviness to one thousandth of the nozzle throat diameter (i.e., $0.001 \cdot d$). The sense of such requirement is rather mysterious, as finding a correlation between the peak-to-peak distance and the evolution of the flow-field within the nozzle is rather difficult. Also from the point of view of technology experts such requirement is problematic, because it is rather difficult to deal with such waviness tolerances on equipments (i.e., the nozzles) that are obtained from machining technology.

As nozzle fluid-dynamics is affected by surface smoothness, we hypothesize that such surface tolerance is affected by a misprint and the correct requirement would imply a *peak-to-valley* distance. Such tolerance would also have an immediate fluid-dynamic explanation, as it would be related to the typical requirement for smooth duct surface.

The rig's nozzles were carefully machined, polished and were subjected to different metrological tests, both dimensional and of surface-characterization (see Fig. 4b)). 3D-laser scans were performed as well, to determine whether the nozzle profile was in accordance with the nominal one. The results of the tests confirmed the achievement of the required shape and surface tolerances.

Proposal

On the basis of the previous discussion, we believe that it would be worth accompanying any dimensional/operative requirement in the Standard with a short explanation. The 2017 code [1] already features appreciable steps towards this direction, but there is still margin for improvement.

RIG COMMISSIONING

The rig construction started on May 19, 2017 and was concluded at the end of June 2017. Leakage tests (see Fig. 5) were performed following both the *pressure-decay* method [2, p.211] and the *two-phase* procedure [2, p.214]. The second procedure (i.e., the two-phase one) turned out to be effective in identifying the chamber leakages and consequently minimizing them. After the completion of all the preliminary tests, the rig was put into operation on July 17, testing the 800 mm preswirler-rotor *a)* *b)*

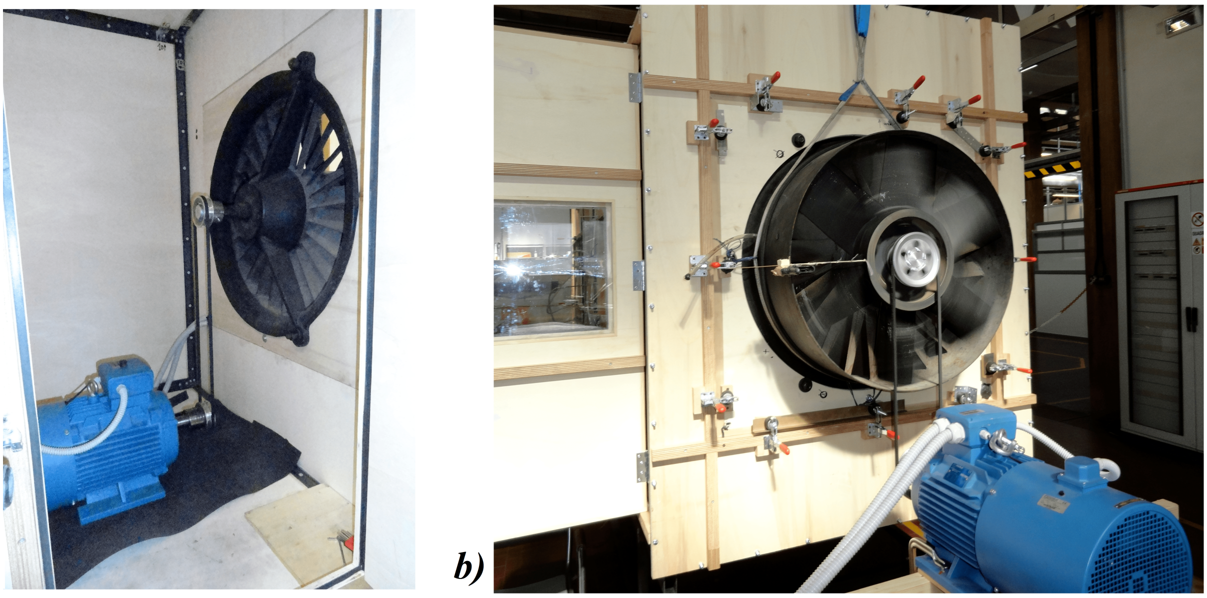

Figure 6: Tests on the 800 mm Preswirler-Rotor fan with the motor on the inlet-side (a) *and outlet-side* (b).

fan visible in Fig 6. The fan is intended to be driven from the inlet side; accordingly, a 15 kW electric motor was placed inside the chamber. Seven different tests at different speeds (from ~ 400 up to ~ 800 rpm) were performed with such inlet-side motor configuration (instrumentation and related uncertainty are presented in [11]). As the electric motor is self-cooled by a 80 W axial fan, there were some worries on eventual detrimental effects on the flow-field inside the chamber and, in turn, on the fan performance characteristic. As verification, a dedicated outlet-side transmission system has been manufactured and the tests repeated with the motor outside of the chamber (Fig. 6*b)*). No appreciable differences on the fan performance characteristics were observed (i.e., the differences are within the measurement uncertainty required by the Standard [2]). While the 800 mm fan size was the maximum testable on this rig according to [2], the new constraints imposed by Formulation 3 of the new Standard [1] limits the fan size to 700 mm.

Since July 2017 the rig has been used extensively both for research and didactic purposes.

CONCLUSIONS

The design and construction of an inlet chamber test rig featuring a multi-nozzle system for flow-rate measurements has been described. The simple structural layout used for the rig features an internal frame that supports the stiffened panels of the chamber walls. This layout resulted in an effective and easy to manufacture assembly, which is now installed at the University of Padova.

The study that was carried out before the rig design highlighted three relevant issues of the ISO 5801 Standards:

- the distinction between chamber types and the related (mandatory) use of the pitot probe to measure the air chamber total pressure;
- the mandatory/preferred allocation of the nozzles with respect to the axis of the chamber;
- the tolerance specification on the waviness of the inner surface of the nozzles.

These arguments required time for a deeper investigation during the rig design process; the achieved insights have been reported within the paper. Suggestions for an eventual improvement of a future Standard have also been provided. In particular, the importance of providing a brief explanation of the Standard requirements is remarked.

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ACKNOWLEDGEMENTS

The rig structure has been financed by the University of Padova-Department of Industrial Engineering with *Incentive to Didactics* funds. The authors are grateful to prof. M. Carbonaro of the Von Karman Institute (Belgium) for valuable structural design suggestions, to proff. G. Concheri, R. Meneghello, and dr. F. Medeossi for performing the surface conditions tests, and to prof. C. Bettanini for his valuable help with the rig instrumenation. The authors acknowledge Cappellaro Snc and Metalpebo Srl for their excellent work. The recommendations and suggestions of an anonymous reviewer have been sincerely appreciated.