LABORATORY ASSESSMENT OF THE FLUID DYNAMIC PERFORMANCE OF FAN-COILS

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ABSTRACT

Usually, the performance of fan-coils is defined and measured in the laboratories only through thermal quantities. However, comfort conditions within a room depend also on the air flow pattern determined by the appliance. Therefore, an experimental procedure to evaluate the fluid dynamic performance of fan-coils has been developed.

The fan-coil is tested using a thermostatic chamber (calorimeter) in which, along with the usual thermal quantities (temperatures, heat flows), the main air flow pattern of the jet (air velocity and turbulence intensity) has been measured in isothermal conditions using an ultrasonic anemometer.

The thermal and flow fields in the chamber have also been determined both in the heating and cooling operation modes, and the results have been used to evaluate the global and local discomfort in the test room.

KEYWORDS

Air conditioning, air flow pattern, chamber study, full-scale experiments, thermal comfort.

INTRODUCTION

The air flows inside a room are driven by complex phenomena where natural convection and buoyant plumes, usually, interact with air jets and forced convection resulting in highly 3D turbulent motions (Jouini et al., 1994).

In spite of the important role played by the air distribution in order to assure proper levels of thermal comfort and indoor air quality, there are few efficient methods to simulate such phenomena using mathematical models. In fact, hand calculation and simplified models are usually unable to provide a detailed description of the flow and thermal fields inside the enclosure. At present, the only method that, in theory, assures a proper approach is computational fluid dynamics (CFD). However, due to the complexity of the procedure, the success and reliability of the air flow patterns simulation performed with CFD codes is not certain *a priori*. This implies that it is particularly helpful to gather detailed and complete[1] sets of experimental data related to fluid dynamic fields inside a room in order to evaluate and, eventually, optimize the numerical methods (Hu et al., 1996, Jouini et al., 1994).

Field experiments for investigating space air distribution may be carried out by means of different procedures. For what concerns the aspects related to local air change rate and air quality (i.e. the evaluation of the proper quantities of supply ventilation air) a suitable procedure makes use of the air distribution indices, which may be measured by means of tracer gas techniques.

When, on the other hand, the aim is to verify the presence of disturbing drafts, the following quantities have to be measured:

- local air velocity and temperature,
- thermal gradients
- local turbulence intensity.

Most currently available data regarding space air distribution are related to systems adopting air diffusers while, so far, little attention has been paid to fan-coils fluid dynamic performance, in spite of the fact that, at least in Italy, fan-coils are by far the most common terminal unit for air conditioning.

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[1] The turbulence characteristics are seldom measured.

In the light of these remarks an experimental campaign has been carried on using a laboratory thermostatic chamber (*calorimeter*) in order to assess both thermal and air distribution characteristics of a typical fan-coil, considering isothermal, cooling and heating operation conditions.

Results are presented in terms of distributions of air velocity, of PPD (Predicted Percentage of Dissatisfied) and PD due to draft inside the room, and of Air Distribution Performance Index, ADPI (see, e g, Ashrae Handbook, 1997).

EXPERIMENTAL APPARATUS

The Italian Standard UNI 8728 for space air outlets tests requires that the test chamber has the following features:

- the test room height should not be less than 2.8 m, its width should be within 1.5 and 2.2 times the height, and the length should not be less than 7.5 m
- all surfaces should be plane and smooth and should meet at normal angles
- air should be extracted from a site distant from the measurement points

The calorimeter used for this work is made of two adjacent rooms (fig. 1) of equal size (4.2x4.2x3 m). Therefore, the width of the room is 30 cm smaller than required, and its length (4.2 m) is far below the 7.5 m required by the Standard. However, such a high length is required for air outlets - having high throw features - not for fan-coils, whose flow field has usually an extension of no more than 2.0 m.

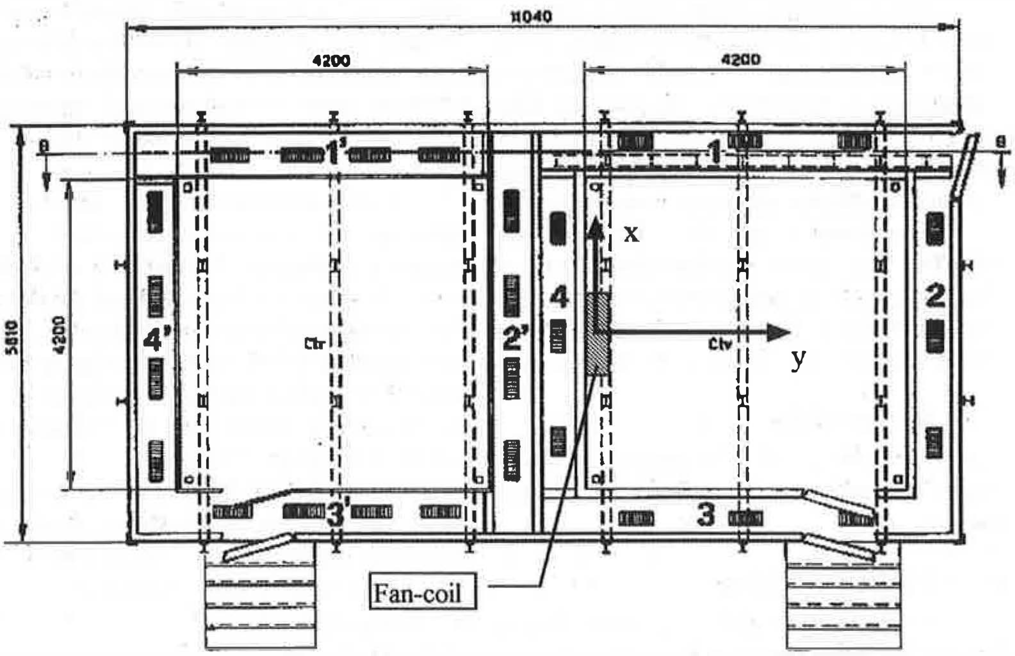


Figure 1 - The calorimeter adopted as a test room (z axis is the vertical axis)

Therefore, the calorimeter was considered as substantially meeting the Standard requirements.

One of the two chambers of the calorimeter, the adiabatic chamber, has heavily insulated walls (100 mm of polyurethane), while the other is enclosed by low thermal resistance walls (25 mm of chip board). Both chambers are surrounded by air cavities 0.6 m wide in which air is introduced at controlled temperature and humidity conditions.

During the measurement campaign on fan-coils, the adiabatic chamber has been used. Within this chamber a so-called "compensator" is present, equipped with cold and hot heat exchangers, which allow the control of temperature and humidity in the chamber.

During the tests the fan-coil has been positioned at the center of the wall facing the compensator.

Beyond the instrumentation and the control systems of the calorimeter (see Saggese, 1984, for further details) the measurement apparatus included Gill WindMaster ultrasonic anemometer, able to measure the three components of the air velocity vector with a precision of ±1.5 % (between 0 and 20 m/s) and a resolution of 0.01 m/s. A sampling frequency of 1 or 4 Hz may be chosen. On the anemometer a Pt100 shielded resistance thermometer has been positioned in order to measure also the air temperature field in the room.

COMMENTS TO THE TESTS

An intrinsic problem which had to be solved was how to keep the conditions within the chamber at the desired level without disturbing the flow field. Steady-state conditions are achieved in the chamber only when all the heat removed (or introduced) by the fan-coil is introduced (or removed) from the room

i) through the walls (external compensation),
ii) by the compensator (internal compensation)

Actually, being the walls heavily insulated, external compensation was negligible.

Furthermore, in order not to create interference with the flow field, the compensator had to be used with its fans switched off in the heating mode, and was replaced, in the cooling mode, by two 1 kW electric heaters.

As a consequence, the temperature within the room could not be kept at the desired set-points (22 °C in heating mode and 25.5 in cooling mode). In the heating mode the ambient temperature drifted to 26 °C, while in the cooling mode it was not possible to keep it below 26.6 °C.

Therefore, when comfort evaluations were performed, all the temperature field had to be shifted, correcting it by - 4 °C in the heating mode, in order to achieve a comfort value of 22 °C, and by -1 °C in the cooling mode, in order to achieve a comfort value of about 25.5 °C.

This does not affect the reliability of the data, because only the velocities and the variations in the temperature field are interesting when local discomfort is examined (Int-Hout, 1981).

In all cases at distances greater than 1.5 m air velocities induced by the fan-coil were comparable with those determined by natural convection induced by the walls and the compensator. Therefore, only a "near field" of 2.0x1.5 m was monitored.

MEASUREMENT CAMPAIGN

The first phase of the experimental campaign was aimed at the definition of the thermal performance of the fan-coil, and, in particular, at the assessment of the repeatability and reliability of the measurements, i. e., to "certificate" the calorimeter. Numerous tests, both in the heating and cooling modes, have been performed. The results are not reported here, but the interested reader may find a detailed description by Anglesio et al. (1996).

The second phase of the work was the assessment of the fluid dynamic performance of fan-coils, in order to determine:

a) air jet profile, throw, and 0.5 m/s isolevels in isothermal conditions
b) air velocity, temperature and PPD fields and ADPI in the room in isothermal, heating, and cooling modes
c) detailed air flow field around the fan-coil in isothermal, heating, and cooling conditions

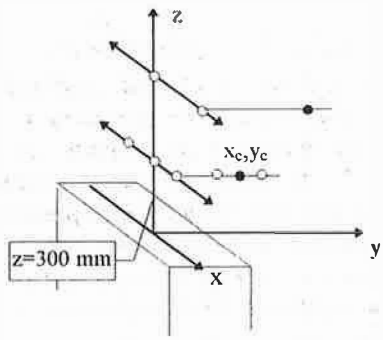
This last measurement campaign, whose results are not shown here, was carried out with the aim of a future CFD testing analysis.

Jet characterization

These measurements have been performed according to the Italian Standard UNI 8728/88, only in the isothermal mode.

UNI 8728/88 describes an investigative technique to determine the throw and the diffusion width of the air jet produced by an ADD (Air Diffusion Device).

- The anemometer is first positioned at a distance z = 300 mm, normal to the grille center (see Fig. 2),
- the anemometer is moved parallel to the grille until the maximum velocity value, V_{max}, is found at x_c
- the anemometer is moved along y, until V_{max} is found at y_c
- now the anemometer is moved away from the grille to a distance z < 1.3 m and the previous steps are repeated in such a way that, at each z, a new value of V_{max} is found



(• = V_{max} points)

Fig. 2 - UNI 8728 procedure for measuring the ADD throw.

The procedure stops when V_{max} becomes smaller than 0.5 m/s. A linear interpolation on a logarithmic chart provides the throw value, corresponding to V_{max}=0.5 m/s.

Flow and thermal field

Air velocity and temperature are measured on five horizontal planes (0.84, 1.34, 1.84, 2.34, and 2.84 m above the floor) on a regularly spaced (0.25 m each step) sampling grid starting from the fan-coil center, with manual positioning of the anemometer (see Table 1)[2]

At each measurement point, in order to obtain accurate values of the time averaged velocity and turbulence intensity, the sampling period was 120 s for the isothermal case and 60 s for the heating and cooling mode, well above the recommended value of 30 s (Hu et al., 1996).

Tab. 1 - No. of points for field analysis

plane height above the floor	isothermal mode	heating mode	cooling mode
0.84	52	21	23
1.34	41	35	29
1.84	51	48	46
2.34	56	57	55
2.84	56	59	-
total	**256**	**220**	**153**

Furthermore, thanks to the adopted sampling rate (4 Hz), it has been possible to measure correctly the air velocity fluctuations. In all the test zones high turbulence levels have been observed (up to 60 - 70 %). Moreover, it has been pointed out that the turbulence intensity is not meaningful (values higher than 100 %) in those areas where the local air flow fields, due to unavoided disturbance, have the same magnitude of the velocity fluctuations.

PRESENTATION OF RESULTS

Jet characterization

For the fan-coil tested, a throw value of 2.45 m was found. Fig. 3 shows five horizontal sections of the 0.5 m/s velocity profile.

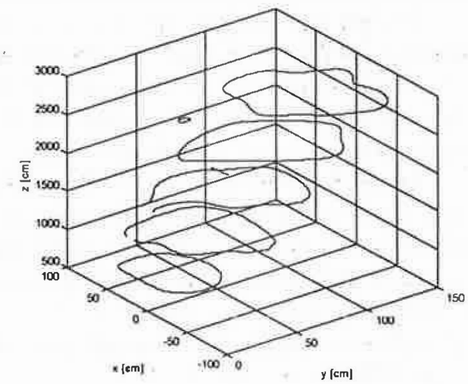


Figure 3 - Iso-velocity (0.5 m/s) contours at different height

[2] measurements were not performed at locations where velocity was below 0.05 m/s approximately.

Flow and thermal field (comfort analysis)

Among the numerous parameters used to define the comfort conditions in air conditioned rooms, the Air Diffusion Performance Index (ADPI) and the modified Air Diffusion Performance Index (ADPI*) have been used since a rather long time.

ADPI (Ashrae, 1997) is defined as the percentage of locations in a room where the two following criteria are met:

- effective draft temperature is comprised between -1.5 and 1.1 °C, and
- air velocity is lower than 0.35 m/s.

In turn, the effective draft temperature is defined as:

$$\theta = (T_x - T_c) - 8 \cdot (V_x - 0.15) \quad (1)$$

where:

T_x is the local air temperature

T_c is the average room temperature

V_x is the local air velocity

ADPI* is a modified index introduced by Abu-El-Hassan et al. (1996) including a further requirement, i e, that the parameter called PD < 20 %. PD is the predicted percentage of dissatisfied people due to draft, local air temperature and turbulence intensity:

$$PD = (34 - T_x)(V_x - 0.05)^{0.622}(3.14 + 0.37 V_x Tu) \quad (2)$$

where:

Tu is the local turbulence intensity.

Table 2 - ADPI and ADPI* values.

Index	Mode	Near field	Whole room
ADPI	Isothermal	69	90
	Heating	6	80
	Cooling	26	86
ADPI*	Isothermal	23	74
	Heating	5	80
	Cooling	20	84

It is possible to see that, for the heating and cooling mode, the difference between ADPI and ADPI* is less than 30 % (the influence of turbulence intensity on the comfort determines a decrease in the ADPI value), while for the isothermal case the influence of turbulence is much higher and the ADPI value decreases by about 70%. The results presented in Table 2 show that:

- the whole room behaviour is satisfactory and the air space distribution is correctly achieved (for all tests, both ADPI and ADPI* are higher than the minimum value, 60 %, recommended by Ashrae).
- Comfort in the near field is highly compromised in cooling and heating mode, while, in the isothermal case, there are large differences between ADPI and ADPI* information. According to the ADPI criterion comfort in the room appears to be sufficient, while according to ADPI* a discomfort condition arises.

In order to provide a better picture of the air distribution, as an example, the velocity profiles for the heating mode are shown in figures 4 to 8. They show the gradual decrease of velocity with height, and the displacement of the plume center far from the wall where the fan-coil is placed. Table 3 shows the maximum velocity values and locations on the y-axis[3] at different z-levels.

Table 3 - V_{max} and location

Plane	Y_{max} [cm]			V_{max} [m/s]		
[m]	Cool.	Iso.	Heat.	Cool.	Iso.	Heat.
0.84	25	25	25	2.44	2.38	2.00
1.34	50	50	50	1.52	1.47	1.83
1.84	75	50	75	1.28	1.13	1.68
2.34	125	75	100	0.93	1.04	1.43
2.84	-	100	125	-	0.65	1.08

The global comfort was evaluated through the well known PPD concept (Fanger, 1972). Its values are plotted for a z-level of 1340 mm in figures 9 and 10, both for the heating and cooling modes. For the heating mode the maximum PPD is less than 9 %, while values of PPD up to 50 % are reached in the cooling mode, even if this is limited to a small area close to the fan-coil.

Coming to local discomfort analysis, adopting eqn (2), the iso-PD lines have also been drawn. In figure 11 to 13 it is possible to see the PD fields, for the cooling mode, on three different horizontal planes, located at different heights above the fan-coil.

[3] the maximum velocity always falls on x=0.

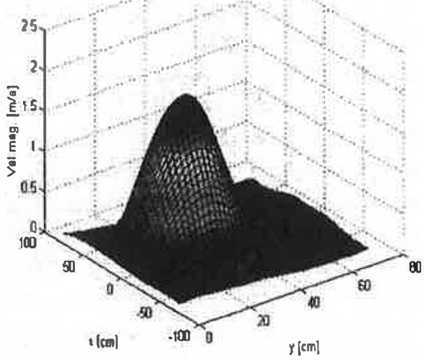


Figure 4 - Velocity magnitude - heating mode
z = 840 mm

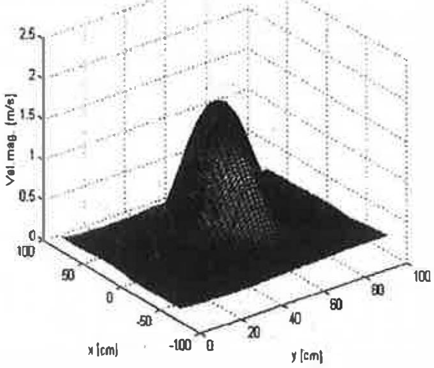


Figure 5 - Velocity magnitude - heating mode
z = 340 mm

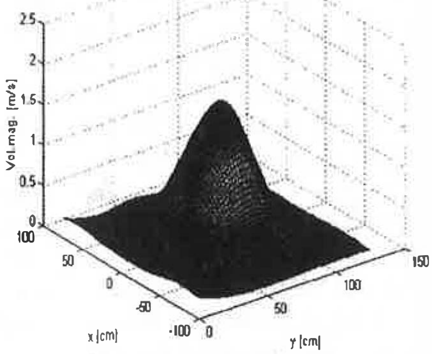


Figure 6 - Velocity magnitude - heating mode
z = 1840 mm

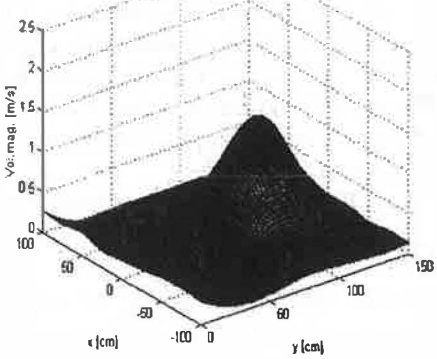


Figure 7 - Velocity magnitude - heating mode
z = 2340 mm

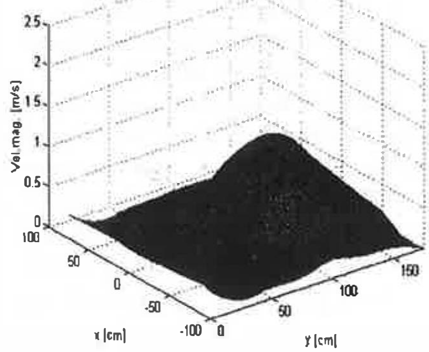


Figure 8 - Velocity magnitude - heating mode
z = 2840 mm

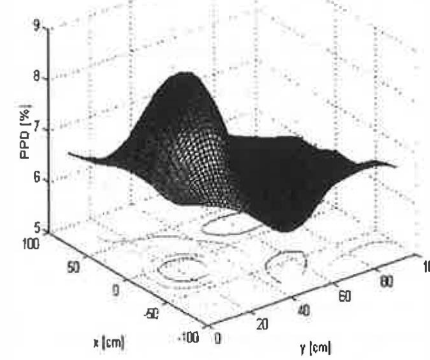


Figure 9 - PPD values - heating mode
z = 1340 mm

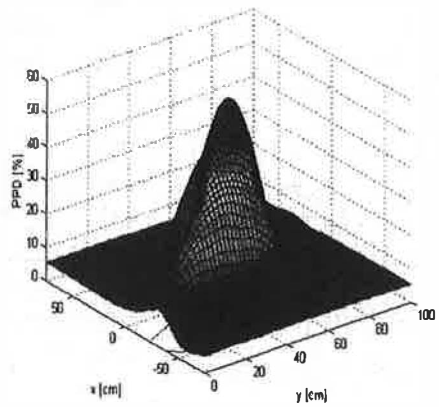


Figure 10 - PPD values - cooling mode
z = 1340 mm

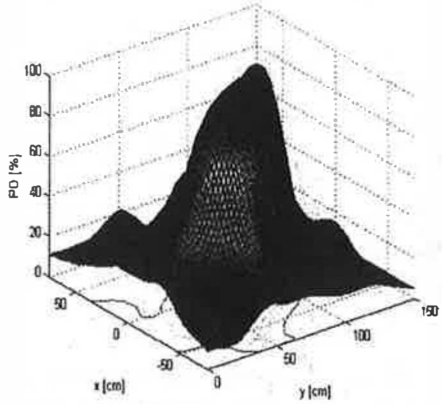


Figure 13 - PD field - cooling mode
z = 1840 mm

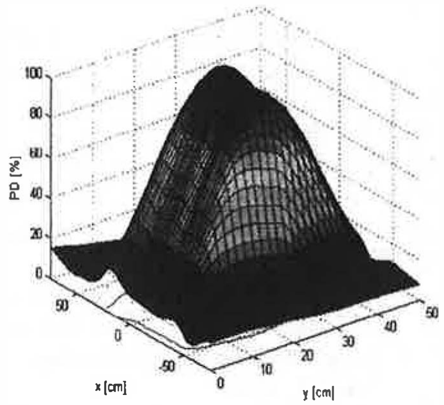


Figure 11 - PD field - cooling mode
z = 840 mm

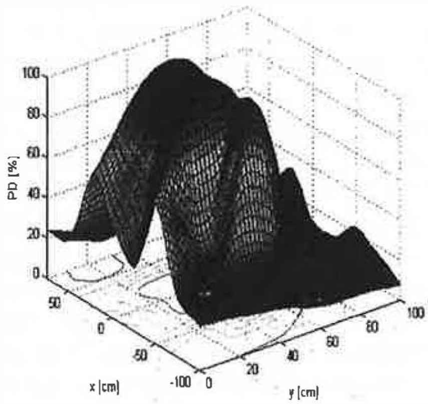


Figure 14 - PD field - isothermal mode
z = 1340 mm

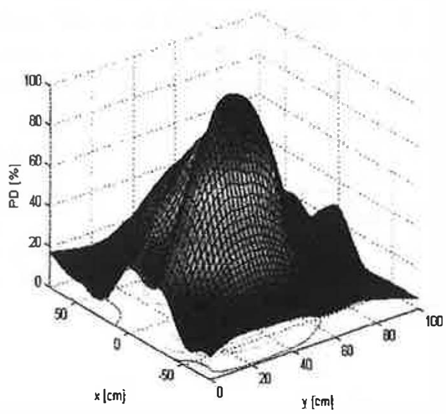


Figure 12 - PD field - cooling mode
z = 1340 mm

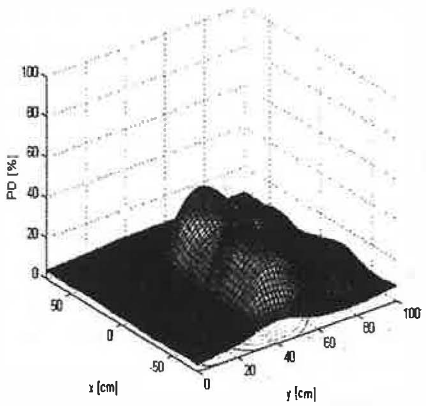


Figure 15 - PD field - heating mode
z = 1340 mm

They underline the high local discomfort due to low air temperatures, high air velocities and quite intense turbulence. Comparing these charts with figures 14 and 15, related to isothermal and heating mode (plane z = 1340 mm), it is possible to see that the discomfort level is almost the same for the first case while it decreases in the last one, although a quite large portion of the near field still lies above the 20% limit

CONCLUSIONS

This paper presents an extensive set of measurements of both thermal and velocity fields generated by a fan-coil located in a calorimeter.

Results are presented in terms of component performance, and of thermal comfort in the surrounding of the fan-coil and in the room. The comfort analysis has been carried out taking into account two types of performance indicators: the first one is the typical one-number indicator such (ADPI and ADPI*); the second is based on detailed field description of global (PPD) and local (PD) discomfort. Data provided by these analyses convey useful information and may profitably help the designer in the choice of the space air diffusion strategy and in the positioning of the air terminal devices. In particular, the discomfort field description appears as a powerful tool for detailed evaluation of fan-coil performance in a full-size room.

Moreover, the adopted experimental procedure is reliable and easy to carry out (even if time consuming). 3-D ultrasonic anemometer is adequate for what concerns accuracy and stability, to measure air velocity fields, and is able to determine turbulence characteristics. On the other hand some problems may arise in the far field, where the air velocity is very low, due to the compensator inside the chamber. This item should be carefully analyzed and managed during the test planning in order to minimize disturbances on the measures.

Future developments of this research will go towards a CFD analysis. In fact, thanks to the detailed knowledge of the thermal and flow fields and operating boundary conditions, it is possible to perform reliable and detailed comparison between numerical and experimental results.

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