MEASURING THE FLOWRATE OF FANS BY EMBEDDED SENSORS

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SUMMARY

The measurement of the flow rate of fans installed in systems or appliances with sensors on the fan itself allows better control of airflow as the system varies with time. This paper presents experimental results on a double-inlet forward-curved (FC) centrifugal fan equipped with static pressure taps flush-mounted in the inlet bells. Additional measurements have been made with a device designed and commercialized by NICOTRA Gebhardt consisting of a pressure probe clipped onto an inlet arm of the fan. The flow coefficients of both systems are determined as a function of the fan speed and operating point, the actual flow of the fan being determined on a reference test rig equipped with orifice plates. An uncertainty analysis is made taking into account the measurement uncertainty of the instruments used in the tests.

INTRODUCTION

When a fan is installed in a system or equipment, its volume flow is sometimes not accurately known for a number of reasons: pressure loss of the system under or over estimated, fan system effect, a misunderstanding of the performance curve provided by the fan manufacturer, changes to the flow rate with time. All of which may lead the installer to oversize the fan to reduce the risk of a lack of flow, which results in an increase of energy consumption and noise. For that reason it may be useful to measure the flow rate with sensors embedded into the fan in order to know and control it at any time if necessary. A pressure tap is often installed by the fan manufacturer on the inlet nozzle of plug fans to allow the installer to assess the flow from the pressure measurement and the value of the flow coefficient C provided by the manufacturer. For the same reason pressure taps are sometimes put on the inlet nozzle of backward-curved centrifugal fans. Other types of fans, i.e. axial fans and forward-curved (FC) centrifugal fans are usually not equipped with such probes.

This paper presents an experiment performed on a double-inlet FC centrifugal fan to assess its volume flow either with pressure taps in the inlet bells or with a pressure probe designed by

NICOTRA Gebhardt clipped onto an inlet arm of the fan. For both systems the flow coefficient is measured as a function of the rotational speed and operating point.

In a second step, the paper presents how a model law can be built to calculate the flow rate from the pressure measurements. The accuracy of the calculated flow rate is discussed according to the specifications of the sensors used for the measurements.

EXPERIMENTAL CONFIGURATION

Test setup

The experimental work has been performed on a double-inlet FC centrifugal fan of 200 mm impeller diameter. Figure 1 shows the main dimensions of the volute. The fan, which has been tested according to ISO 5801 [1] test configuration A, blows into a pressure chamber (Figure 2) connected to a duct and an auxiliary fan. The volume flow is measured with an interchangeable orifice plate[1] located in the middle of the duct downstream of the pressure chamber.

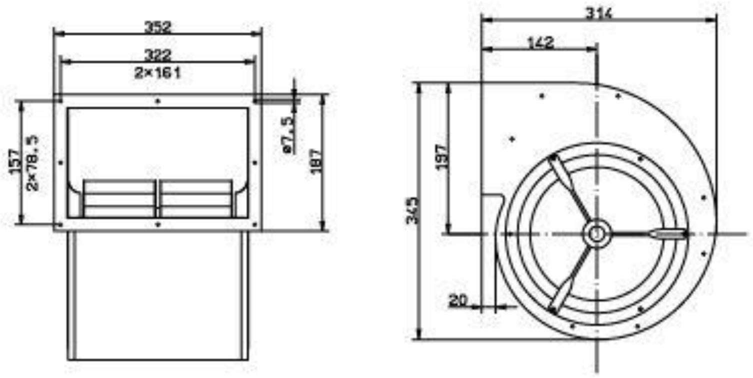

Figure 1: Main dimensions of the volute (in mm)

Figure 2: Fan fitted to the pressure chamber

The fan is driven by a single-phase AC motor whose rotation speed may be modified by changing the voltage of the electrical supply.

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[1] The orifice plate is automatically changed via a robot

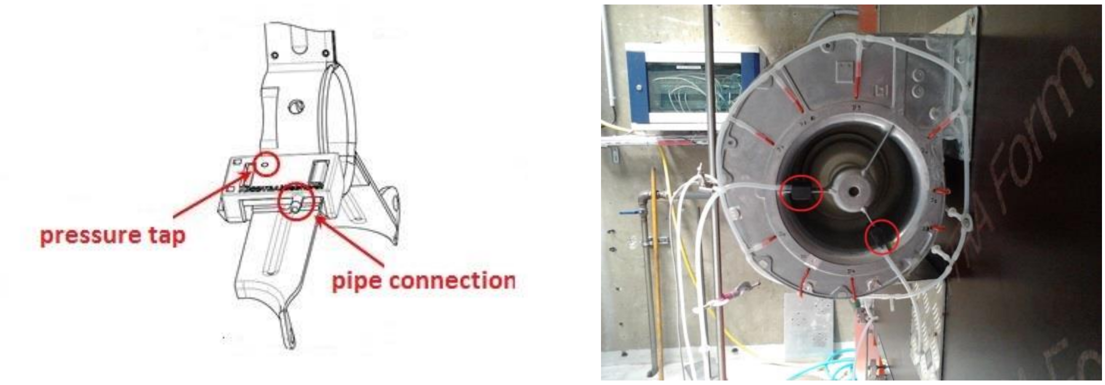
Pressure taps in the inlet nozzles

Nine pressure taps have been flush-mounted on the neck of each inlet nozzle (Figure 3). Due to the presence of the three arms the taps are not equidistant on the circumference but they are equally spaced between two successive arms. The vinyl pipes connecting the taps to the pressure transducer are red in Figure 3. The pressure is either measured for each individual tap or averaged over the 18 pressure taps using pressure rings.

Figure 3: Layout of the pressure taps on the right inlet nozzle

Pressure probe on the inlet arm

A sketch of the NICOTRA pressure probe is shown in Figure 4-a. The probe is clipped onto the arm in such a way that the flow is mainly tangential to the pressure tap on the probe surface. Two probes were available during the tests: they are surrounded by red circles in Figure 4-b. They were placed successively on the three arms at several radial locations to investigate the influence of the probe position on the flow coefficient.



a) *b)*

Figure 4: Nicotra pressure probe
a) Description (Nicotra documentation) b) View of the probes on 2 inlet arms

RESULTS

Fan curve

The air performance of the fan was measured at several rotational speeds. The tests were undertaken at two constant voltages 230 V and 60 V and two constant speeds N_{max} = 1140 rpm and 0.6 N_{max} along the fan curve to determine if the speed had an influence on the results. Table 1 shows the speed range of the four test conditions. When the voltage is constant (230 V or 60 V) the speed varies along the fan curve due to the motor slip between a minimum (high flow rate) and a maximum (low flow rate). To obtain a constant speed (N_{max} or 0.6 N_{max}) over the fan curve the voltage had to be adjusted at each operating point. Figure 5 shows the fan static pressure curves measured at the four operating conditions and converted at a constant speed N = 1140 rpm with the fan laws. All the curves merge when taking into account the measurement uncertainties.

Table 1: Rotation speeds for the different test conditions

Test conditions	N (rpm)
230 V	1155 – 1423
60 V	242 – 607
N_{max}	1140
0.6 N_{max}	672

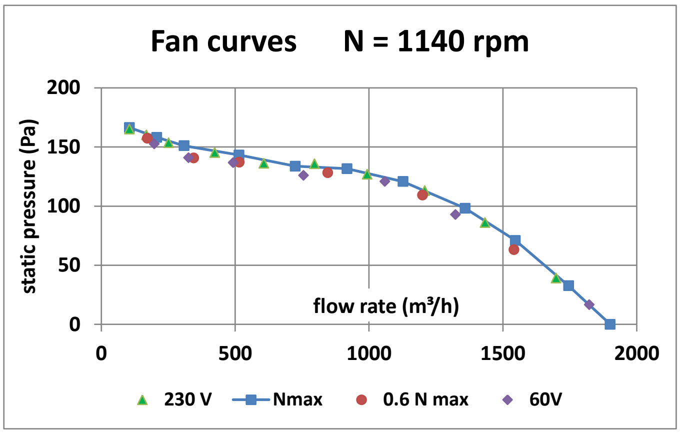


Figure 5: Fan curves at 4 rotating speeds converted to N = 1140 rpm

Pressure taps in the inlet nozzles

From the measurement of the flow rate, the relative pressure in the nozzles and the air density the flow coefficient C is calculated:

$$C = q_v / \sqrt{-2\Delta p / \rho} \tag{1}$$

where:

q_v: volume flow measured with the orifice plates (m³/s),

Δp: pressure difference between the average pressure over the 18 taps and the external pressure (Pa)

ρ: air density at the fan inlet (kg/m³)

N: rotation speed of the fan (rpm)

Figure 6 presents the flow coefficient C as a function of the flow rate for the four test conditions. In this figure the flow rate on the x axis has been converted to a constant speed N = 1140 rpm with the fan laws:

$$q_{v,1140} = q_v \times \frac{1140}{N} \tag{2}$$

where $q_{v,1140}$ and q_v are respectively the converted and the measured flow rates in m³/h and N is the rotation speed in rpm.

Figure 6 reveals a nearly perfect merging of the different curves, which proves that C does not vary significantly with fan speed even if the speed range is quite large as shown in Table 1. On the other hand, the flow coefficient varies with the operating point as shown in figure 6.

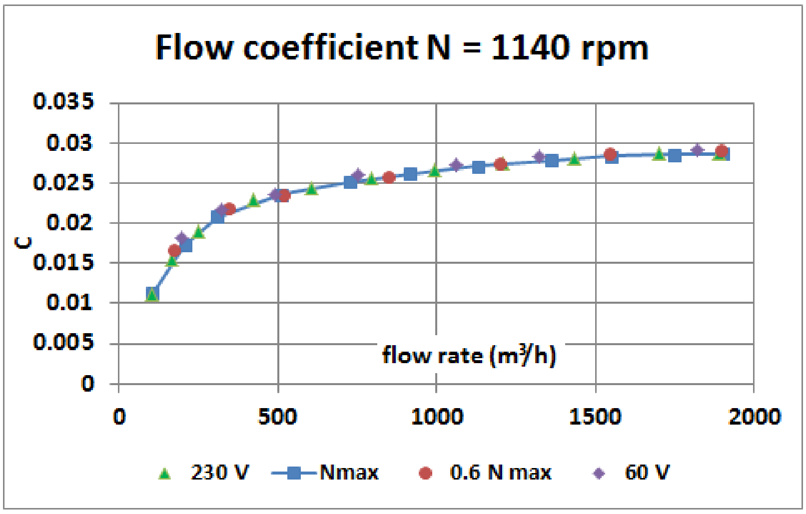


Figure 6: Evolution of the flow coefficient with flow rate at 4 speed conditions – C averaged over the 18 pressure taps

As it is not realistic to install 9 pressure taps on the inlet nozzle of commercial fans the next step was to investigate the influence of the circumferential location of the tap on the flow coefficient in order to observe if one or two taps wisely placed in the nozzle could replace the nine taps that are numbered in Figure 7.

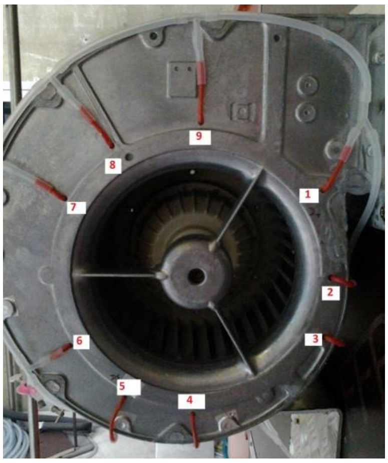


Figure 7: Pressure tap numbers

Figure 8 presents the flow coefficient measured with the individual pressure taps of the right and left nozzles respectively at low, medium and high flow rates, i.e. at 250, 790 and 1720 m³/h for N = 1140 rpm. The average flow coefficient over the 18 pressure taps of the two inlet nozzles is also represented by a dotted line in the figure. These results show that C depends on the location of the pressure tapping and that the lower the flow rate, the greater the difference between the measured data is. For the low flow rate tap position 3 shows no value of C as the static pressure Δp is positive instead of negative due to the fact that at this particular circumferential location the flow escapes instead of entering into the impeller. This figure also shows that the flow coefficient depends very little on the nozzle side because of the symmetry of the fan volute.

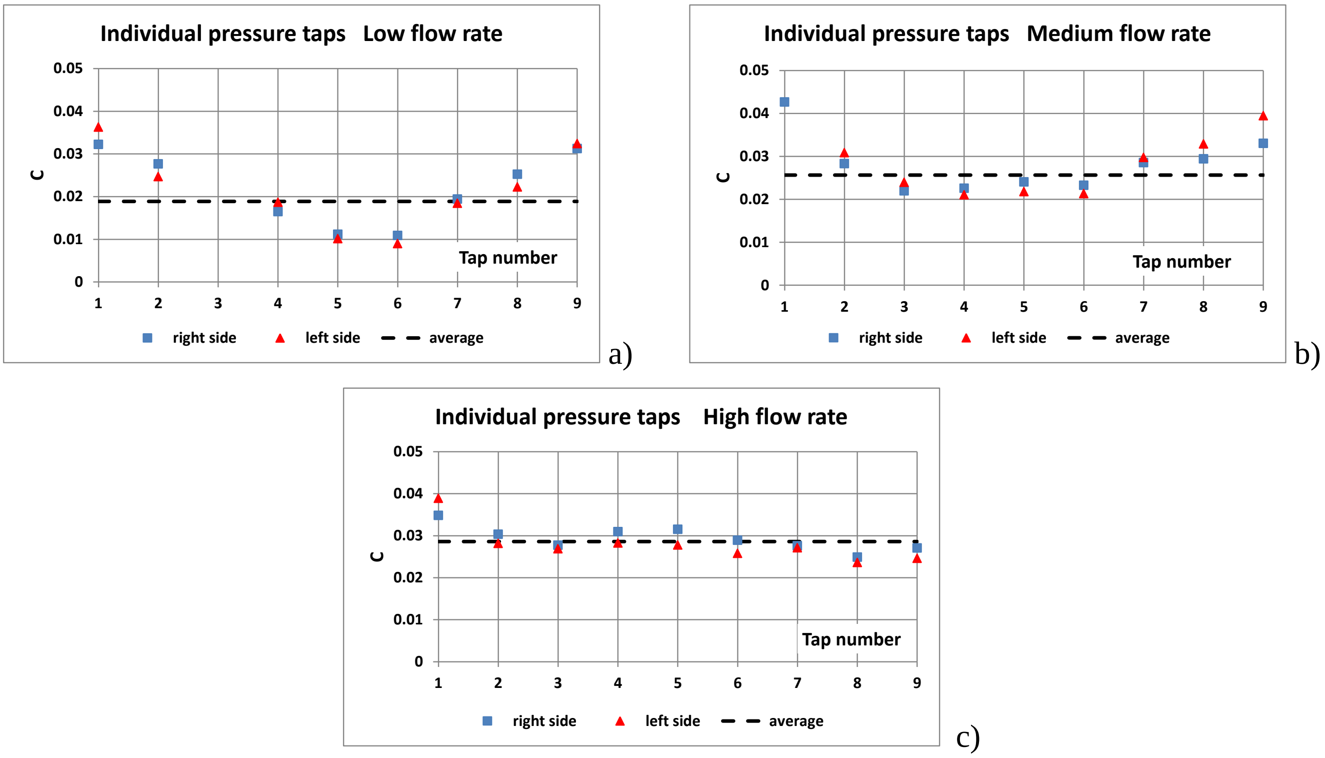


Figure 8 Flow coefficient for each pressure tap – 230 V
a) Low flow rate b) Medium flow rate c) High flow rate

From the results of Figure 8 and others at a lower rotation speed it has been observed that the average of C on taps 4 and 7 is very close to the average over the 18 taps whatever the operating point. That means that two taps located at positions 4 and 7 could be enough to deduce with a reasonable accuracy the flow rate from the pressure measurement and the average flow coefficient at these two positions.

Pressure probe on the inlet arms

The three arms, numbered 1, 2 and 3 in Figure 9, were successively equipped with the two Nicotra probes. Figure 10 shows the characteristic of C with respect to q_v with the probe on the three arms at the minimum and maximum radial locations, i.e. against the bearing (Figure 10-a) and against the nozzle (Figure 10-b). The average flow coefficient obtained with the pressure taps on the nozzle is presented for comparison in Figure 10-a. For both radial locations the probe on arm 3 is not appropriate since the characteristic of C with q_v appears erratic unlike the other curves. With the probe close to the nozzle on arm 1 (Figure 10-b) a peculiar value of C is obtained at 250 m^3/h. In the end, only the probe located on arm 2 provides satisfactory results. The same trends are observed at a lower rotation speed.

Finally, Figure 11 shows the influence of the radial location of the probe on arm 2. The value of C slightly depends on the radial location of the probe in the most flow range but the shape of the three curves is nearly the same. This result is encouraging as it means that the flow coefficient of the probe does not strongly depend on its radial location.

For this reason it is recommended to set the Nicotra probe on arm 2 which is nearly on the opposite side from the volute cutoff.

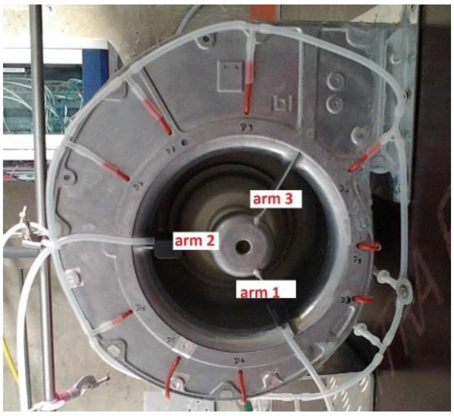
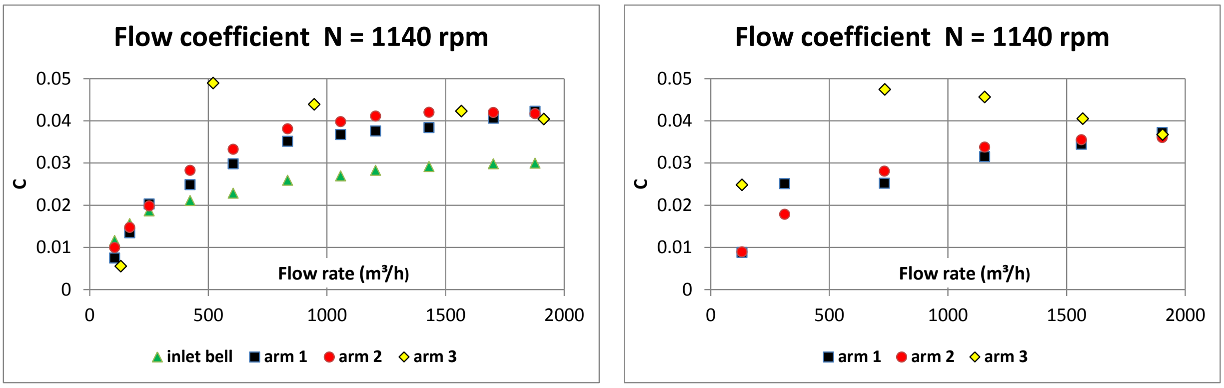


Figure 9: Inlet arm numbers



a) b)

Figure 10: Characteristic of the flow coefficient with flow rate
a) Probe close to the bearing b) Probe close to the nozzle

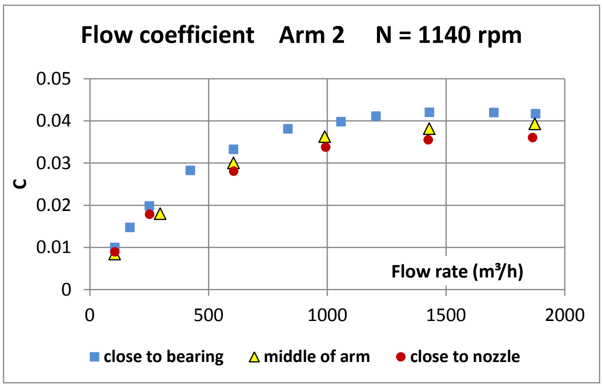


Figure 11: Influence of the radial position of the probe on the flow coefficient – Arm 2

Determination of the flow rate from the pressure measurement in the inlet nozzle

A model is established to calculate the flow rate from the pressure measurement according to the following relation:

$$q_{v,1140} = b_0 + b_1 \times \sqrt{\frac{-2\Delta p_{1140,1.2}}{1.2}} \tag{3}$$

where:

$q_{v,1140}$ is the converted flow rate for a constant speed N = 1140 rpm

$\Delta P_{1140,1.2}$ is the pressure converted to a constant speed N = 1140 rpm and a density of 1.2 kg/m^3

$$\Delta P_{1140,1.2} = \Delta P \times \left(\frac{1140}{N}\right)^2 \times \left(\frac{1.2}{\rho}\right) \tag{4}$$

and b_0 and b_1 are the coefficients of the model, calculated from the test results with the pressure taps in the nozzles (18 pressure taps) at N_{max}.

Taking into account the uncertainty measurement on flow rate, pressure and air density, the model is built using a generalised Gauss Markov regression [1], [2], implemented in the REGPOLY software developed by LNE (French National Laboratory for Metrology and Testing) [3].

Figure 12 shows the result of this model with the measurements (blue symbol), the calculated flow rate (red line) using the established model. The range limited by the red dotted lines represents the uncertainty of the calculated flow rate when using the model law.

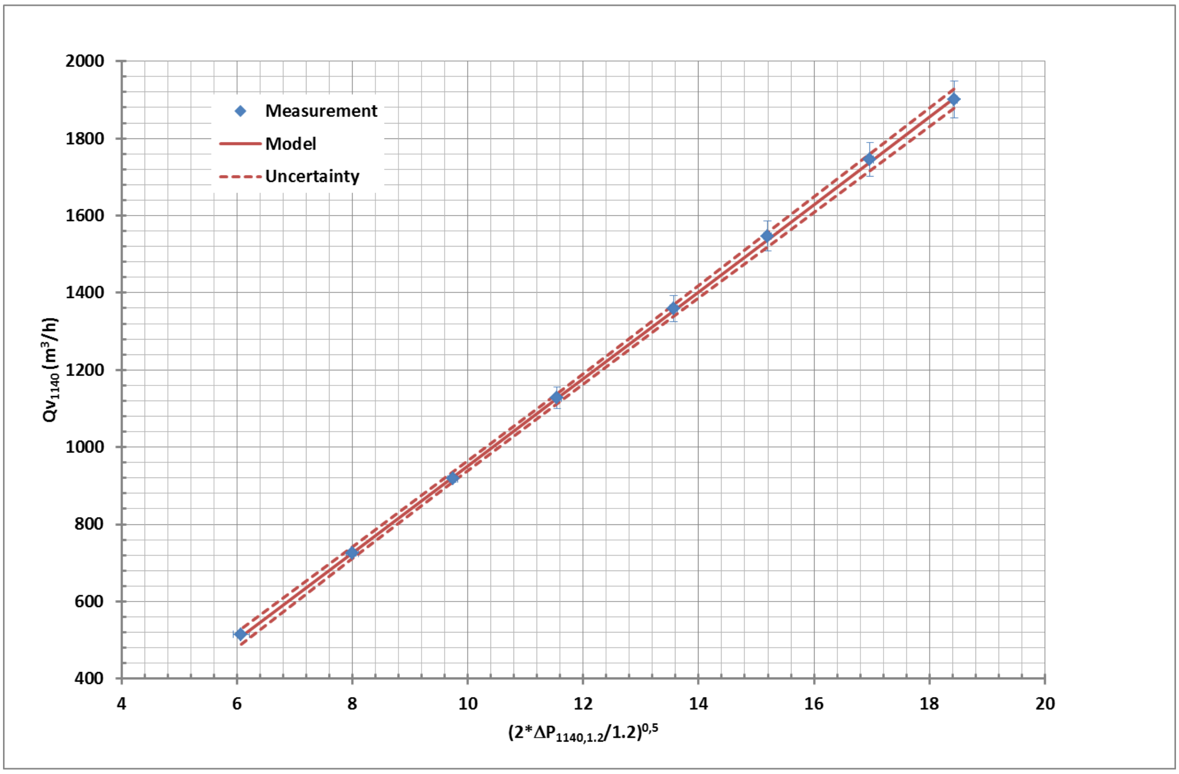


Figure 12 Relation between the pressure and the flow rate at 1140 rpm and 1.2 kg/m^3

The accuracy of the model has been checked at the other test speed conditions 0.6 N_{max} and the two constant voltages 230V and 60V. Once the corresponding pressure measurements are converted to 1140 rpm and 1.2 kg/m^3, the model is used to calculate the corresponding converted flow rate. Then, the calculated flow rate converted to actual conditions of rotation speed and air density is compared to the flow rate measured on the test rig. The error obtained is shown in ***Figure 13***, where Q_v is the calculated flow rate in actual conditions, and Q_{vr} is the reference flow rate measured on the test rig.

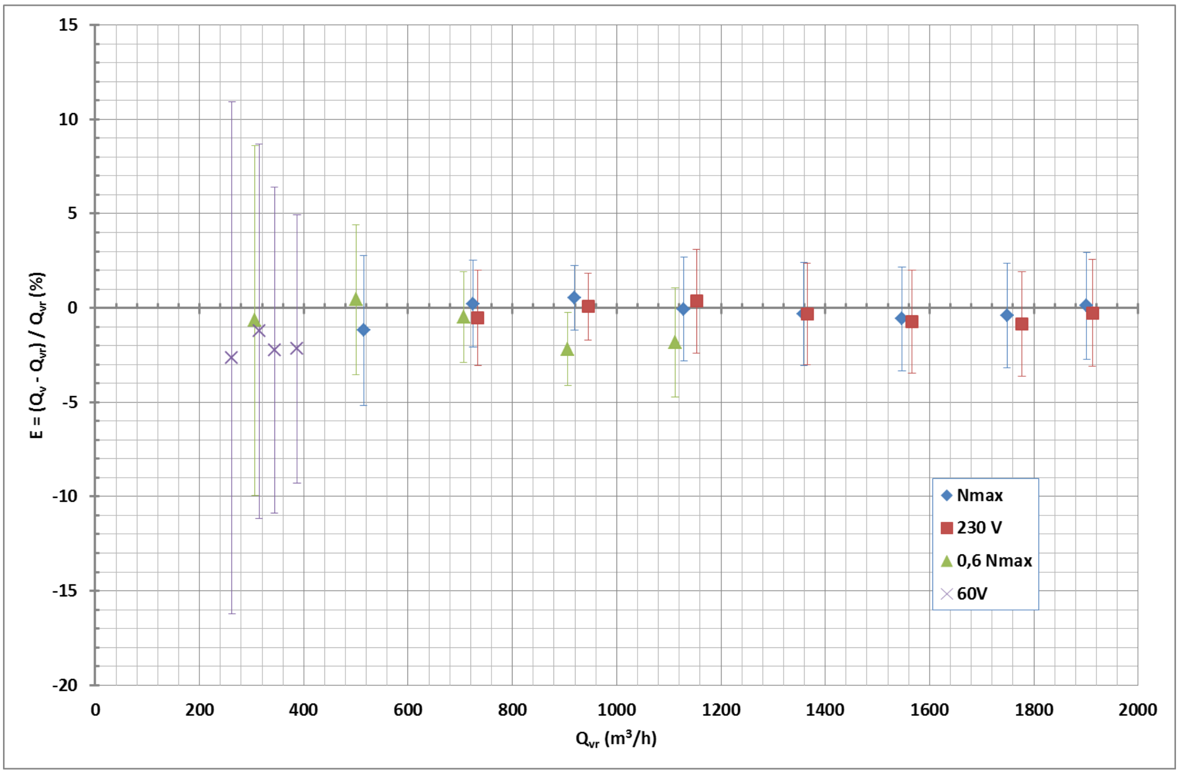


Figure 13 Error on the calculated flow rate at different flow conditions

Figure 13 shows the good agreement between the calculations with the model and the measurements.

The vertical bars represent the measurement uncertainty at each flow rate. This uncertainty depends on the quality of the established model and on the measurement uncertainty of the sensors used.

In particular, the choice of the pressure sensor is critical. In the tests the pressure ranged from 5 Pa (low flow rate) to 200 Pa (high flow rate) and the corresponding measurement uncertainty of the pressure sensor used was 1 Pa. This value explains why the relative uncertainty increases at low flow rates.

For using this model in industrial applications the following algorithm should be applied:

1. Measurement of pressure, rotation speed and density,
2. Calculation of the converted pressure at a rotation speed of 1140 rpm and a density of 1.2 kg/m^3,
3. Use of the model to calculate the corresponding flow rate at a rotation speed of 1140 rpm,
4. Calculation of the actual flow rate and of the measurement uncertainty at the actual rotation speed

The calculated flow rate is known with an uncertainty depending of the model law and the uncertainty of the sensors used for the measurement. These sensors may be different from those used to establish the model law. However, to minimize the flow rate uncertainty of industrial applications the choice of the industrial pressure sensor is important (range and accuracy).

CONCLUSIONS

The experiment performed on a double-inlet FC centrifugal fan has shown that it is possible to assess its volume flow either with pressure taps in the inlet nozzles or with a pressure probe designed by Nicotra clipped onto an inlet arm of the fan.

The tests have shown the most appropriate positions of the pressure taps in the nozzle or on the arms.

After having converted the pressure measurement to reference conditions at a speed N = 1140 rpm and an air density of 1.2 kg/m^3 it is possible to calculate the flow rate using the established law, as well as the uncertainty on the flow rate taking into account the measurement uncertainty of the pressure, rotation speed and density as well as the accuracy of the model.

This experiment has been carried out without any obstacles close to the fan inlet. It will be extended in a near future to configurations in which fan system effect is simulated

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