VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 23

SENSOR FOR CONTINUOUS VENTILATING RATE MEASUREMENT IN LIVESTOCK BUILDINGS

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SENSOR FOR CONTINUOUS VENTILATING RATE MEASUREMENT IN LIVESTOCK BUILDINGS

by

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Written for presentation at the 1990
AIVC Conference

Villa Carlotta
Belgirate
September 18-21, 1990

Summary : A new turbinemeter is developed to be used as a ventilating rate sensor in livestock buildings. Starting from a previous sensor, which we introduced in 1983, several improvements were done to become a low cost air flow rate sensor with an acceptable accuracy of 60 m^3/h in a range from 200 to 5000 m^3/h and this for pressure differences from 0 to 120 Pa. This sensor can be integrated in the climate control equipment of livestock buildings to improve process control.

1 PROBLEM

1.1 The importance of the ventilation rate and the airflow pattern in livestock buildings.

In literature for some time already the ventilating rate trough a livestock building is considered to be one of the most important parameters in inside climate control in livestock buildings. The recognition of barn ventilation problems and early investigations of these provided a milestone in environmental control of animal production structures (King, 1908; Bond, 1976). The influence of air flow rate on the values of physical micro-environmental parameters such as inside temperature, humidity and gasconcentration has been treated (Guss and Grout, 1973; Bruce, 1975; Clark and Cena, 1981). Carpenter (1974) explains the importance of the ventilation rate by its relationship with air temperature and with the air velocity around the animal. He explains the complexity of the role of the ventilation system by the interactions between the inside climate and on the other hand variables such as structure of the building, risk for disease, number of pigs in one pen, etc. The objective of the ventilation system is described as control of gasconcentration, of air humidity and of inside temperature (Bruce, 1981). Consequently the three quantitative criteria are known to determine the ventilation rate through the building and to provide the optimum conditions for the livestock by controlling the physical micro-environment. To achieve this, the two basic funtions of a ventilation system are to provide effective control over the ventilation rate and efficient control over the airflow pattern within the ventilated structure. These should be attained simultaneously (Randall, 1981).
The air flow pattern in the livestock building forms the physical link between the control inputs (ventilating rate and heat supply) and the physical micro-environment around the animal. An adequate control of the ventilating rate through the building indeed has been proven to be decisive in establishing a suitable air flow pattern as well (Randall, 1975; Randall and Battams, 1979; Barber et al., 1982).
The influence of the ventilating rate and its control on energy use in modern livestock buildings also has been described in literature (Buffington and Skinner, 1976; Surbrook et al.,1979; Berckmans et al., 1989).
From literature it can be concluded that the ventilating rate trough a livestock building is one of the most important variables to be conrolled.

1.2 Control of the ventilating rate in commercial livestock buildings

Based on the effect of inside climate control on productivity in modern livestock buildings and on the knowledge about the importance of the ventilating rate, research has been done on the efficiency of normal proportional air flow rate control. By statistical analysis of field data, analysis of control equipment in laboratory test installations and a theoretical approach it has been analysed why normal mechanical ventilating systems (proportional control) do not perform adequately in the field (Berckmans and Goedseels, 1986; Berckmans et al., 1988). Consequently a new controller has been developed, in an attempt to achieve improved control of air flow and hence more effective environmental control in livestock buildings . The principle of this improved air flow rate controller,introduced in 1983, is that the signal of a low cost air flow rate sensor (price of about 130 U.S. dollar) is used to realise a feedback control (Berckmans and Goedseels, 1983; Berckmans and Goedseels, 1986; Berckmans et al., 1986).

2 OBJECTIVES

It is clear that the efficiency of this higher mentioned principle, as it is used today by different European producers of control equipment, depends on the accuracy of the air flow rate sensor that is integrated in the control system. The previous sensor which we developed for the introduction of this controller in 1983 had a limited accuracy of about 600 m^3/h over a measuring range of 500 to 5000 m^3/h.

The objective of this paper is to present a new low cost air flow rate sensor with improved accuracy to be used in modern livestock buildings. This means more specifically that the resulting air flow rate sensor :

- must be usable in the hard environment of a livestock building with high temperature differences (from -10 °C to 55 °C in the exhaust chimney), with a lot of moisture and corrosive gases in the exhaust air and in a very dusty environment,
- must operate for a period of 10 years without any major maintenance,
- must permit to measure the ventilating rate in an exhaust chimney with a diameter of 0.5 m and a short tube length of 1 m and with turbulent air flow in it,
- has a small time constant to be usable as a control sensor,
- has a little weight for a mounting in the chimney,
- has a measuring range from 200 to 5000 m^3/h with an acceptable accuracy and low pressure influence,
- creates a minor pressure drop for the fan.
- must have a low price since it is the objective to integrate this sensor in the control equipment of livestock buildings..

3 METHOD

3.1 Position of the sensor

In most of the modern livestock buildings in Europe the underpressure system is used. In spite of the fact that the building envelope has been improved the last years and is carefully tight up against air infiltration, it still is hardly to realize a leakage-free construction in the field. An exhaust chimney with a diameter of about 0.5 m is a standard in Belgium and the Netherlands. To be sure of measuring the global ventilating rate through the building the air flow rate sensor is placed in the exhaust chimney (Figure 1).

3.2 Principle of the sensor

From the measurements of the characteristics of axial fans, it is known that a back flow effect can appear if the underpressure in the ventilated building is increasing. Because of the high pressure difference to overcome, the fan still keeps his rotational speed but takes the air back through its own impeller. The resulting total ventilating rate is decreasing while the fan still is using a high voltage (Figure 2). In these conditions the air flow in the exhaust chimney is subject to important changes of air flow direction within the considered outlet section. Because of these effects on the turbulent flow in the outlet chimney it is concluded that the sensor should cover the whole outlet section. Consequently a sensor that permits to measure in one point of the section is useless. Taken into account the acquired working conditions of the sensor (see Objectives) and the condition of a low price (to become a sensor for control purposes) the choice was made for the principle of a free running turbine covering the whole outlet section. In literature it is shown that a pressure correcting measurement can be realised with turbinemeters of smaller diameters (Lee et al., 1981, 1982).

3.3 Test rig

In the higher mentioned position a flow rate sensor of the considered principle will be subject to changes in air flow rate and variations in pressure difference over the test chimney as well. Because of the principle of a free running turbine the measurements are focused on the relationship between rotational speed of the turbine, the measured air flow rate and the pressure difference over the test section. Consequently the procedures for fan testing with variation of pressure difference should be applied for testing this air flow rate sensor.
To measure the characteristics of the air flow rate sensor a test chimney is used with a diameter of 0.5 m and a length of 1 m which are the dimensions of the standard chimney as used in the field (Figure 3). In this test chimney it is possible to realise an air flow rate from 0 to 5000 m^3/h and this for pressure differences (underpressures in the testroom) from 0 to 120 Pa. The test installation has been built accordingly to the German Standards DIN 1952, the Belgian Standard NBN 688 and the British Standard BS 848. The principle of this test rig is that the air flow rate can be controlled using the fan of the test rig (nr. 30 in figure 3) while an orifice is used (with compensation for air temperature and air humidity) to measure the flow rate (nr. 27 in figure 3). As long as the second fan (nr. 23 in figure 3) is producing the same air flow rate as fan nr. 30, there is no pressure difference between the test room and the environment. In these conditions fan nr. 30 is taking care for the pressure losses through the system. From the moment where fan nr. 23 is producing a higher air flow rate than the one that is

supplied by fan nr. 30, the underpressure in the test room is increasing. This makes it possible to change the pressure difference over the test section from 0 to 120 Pa and this for every air flow rate from 200 to 5000 m^3/h. Since it takes a lot of time to realize constant steady state conditions in the installation and to do reliable measurements (fluctuations of pressure etc.) the global test installation has been computerized (Figure 3) with the necessary statistical procedures in it. The orifice allows to measure the flow rate with an accuray of 10 m^3/h in the range of 200 to 1500 m^3/h and an accuracy of 30 m^3/h in the range of 1500 to 5000 m^3/h. The rotational speed of the turbine is measured by using a fixed approximity switch with 4 indication points on the turbine. The system allows to measure the rotational speed with an accuracy of +- 0.5 rev/min. The repeatability as calculated from 4 experiments to measure the characteristics of a fan was better than 3 %.

4 RESULTS

To realize the objectives, a number of improvements were carried through on the previous type of turbine. Each of these is described.

4.1 Reduction of the number of blades of the turbinemeter.

At the time of the introduction of the previous type of turbinemeter as a ventilating rate sensor for livestock buildings an identical impeller as the one from the fan was used for practical reasons. This still is done by most of the European producers of this type of equipment and consequently most of the existing turbinemeters for livestock buildings still do have 6 or 8 blades on their impeller (Stylianos, 1989).
Since it can be assumed that a high number of blades does increase the resistance to the fluid flow, experiments were done to measure the air flow rate at 4 different levels of voltage to the fan and this for turbines with 8, 4 and 2 blades.
From Table 1 it can be seen that the resulting air flow rates are higher (up to 900 m^3/h) when turbines with 2 blades are used instead of 8 blades. The pressure losses are minimized by using 2 blades. This is in agreement with the equation for the so called specific number of revolutions :

$$n_s = n \cdot Q / (\Delta P)^{3/4} \qquad (7)$$

Minimum pressure drop is achieved by maximizing n_s and high values of the specific number are characteristic for axial designs (largest flowsection for a given diameter). Moreover, the number of blades reduces as n_s increases. It can be concluded that, as opposed to current practices, it is desirable to use a turbine with only 2 blades on its impeller.

4.2 Position of the turbinemeter.

It has already been explained that the ventilating rate sensor should be positioned in the exhaust chimney (see Method 3.1).
For maintenance reasons the first sensor, as we introduced it in 1983, was positioned after the fan in the outlet-chimney. Since it can be expected that the fluid will have a more uniformly flow pattern before passing through the fan, it is tested whether the turbine should be positioned upstream or downstream with regard to the fan in the fluid flow. For both these positions of the turbine the air flow rate and the corresponding rotational speed of the turbine for different pressure differences over the test section have been measured. From Figure 5 it

can be concluded that the upstream position of the turbine with regard to the position of the fan does have an important positive influence on the characteristics of the turbinemeter. Notwithstanding the great pressure influence and symptoms of back flow below an air flow rate of 2000 m^3/h, there is a good linearity for an air flow rate higher than 2000 m^3/h (Figure 5).

4.4 Design of the turbine impeller.

The impeller of the turbine has different components : the **boss**, the **bearings** and the **blade**.

- The **boss** normally should be as small as possible since a bigger boss decreases the usable area for fluid flow and increases the air velocity. It is known that the pressure losses in ducted flows are proportional to the square of this fluid velocity. But the minimization of the dimensions of the boss are limited by the fact that the positioning of the blades becomes impossible on a too small boss. Moreover the use of too long blades should introduce turbulences. The boss used in this development has a diameter of 0.1 m.
- The **bearing** is a standard SKF enclosed bearing with low temperature Natrium-grease.
- The **length of the blade** is determinated by the difference between the tubediameter and the bossdiameter. When using an inside tubediameter (chimney with an outer diameter of 0.50 m) of 0.45 m and with a so called tip clearance of 0.05 m, the length of the blade is 0.17 m.
  The **chord of the blade** is not commonly calculated in design procedures. Mostly the design is evaluated and improved by windtunnel experiments. Following considerations should be taken in account : the force exerted by the air on the turbine is proportional to the area of the blade. This prompts the use of a large chord. On the other hand a large blade creates more disturbances in the fluid flow. As a compromise, the chord was taken equal to 0.1 m in this development.
  Calculation of the **blade angle** involves determination of the inlet angle β_1. The blade outlet angle β_2 can be determined on the basis of this inlet angle and the curvature of the blade (which is defined as β_2-β_1). Use has been made of the concept of velocity triangles. These vectordiagrams, drawn for a random point of the streampath through the machine, make a study of the different velocity components possible :
- absolute flow velocity V (stationary frame of reference).
- relative flow velocity W (rotating frame of reference)
- peripheral velocity of the rotor blades U , being equal to the product of radius and angular velocity of the rotor.
  The absolute velocity equals the sum of relative and peripheral velocity vectors.
  The calculation of the **blade angle** has to be preceded by the determination of a 'design point', which is to be chosen with the specific application of the turbinemeter in mind. In livestock buildings, the lower values of the ventilating rate are the most critical to control. A high rotational speed is necessary to minimize unstabilities at these low air flow rates. A blade geometry was calculated for a 500 m3/h-250 rev/min design point.
  Table 2 shows the variation of blade inlet angle with radius for this prototype.
  The **curvature of the blade** was expected to be small, since in this application, the turbine has to surmount merely friction losses (as it is not used for the transformation of kinetical to mechanical energy). This can be verified by calculation (Vandenbroeck, 1988) : the maximum curvature amounted to no more than 1°, and was in the actual design assumed to be zero.

4.5 Linearity of the relationship between flow rate and rotational speed of the turbine at different pressure differences.

To determine the steady state characteristics of the turbinemeter its rotational speed was measured at different flow rates and at various pressure differences. Flow rates varied between 200 and 5000 m^3/h. For low flow rates up to 1600 m^3/h, the measuring interval was held on 200 m^3/h. For larger rates, the interval was expanded to 500 m^3/h.Pressure differences varied from 0 to 120 Pa, with a step of 20 Pa. The axial fan in use during these tests was a Woods-2100 fan. The complete range of flow rate from 0 to 5000 m^3/h was measured three times in identical circumstances. A linear regression analysis was carried out on the data. The R^2 was calculated as a measure for the goodness of fit. A good fit implicates independence of pressure variations to a large extent. Calculation of the mean (MΔ) and maximum (MaxΔ) deviation from the regression-based predicted value enables to establish a criterion of accuracy.
Figure 6 shows the corresponding linear regression covering the complete flow rate interval from 0 to 5000 m^3/h. In figure 7 the result of a comparable analysis is shown for a turbinesensor of the previous type (diameter 0.45 m, pressure differences 0-120 Pa). Table 3 shows the comparison of the regression analyses on these results.
It is clear that there is an improvement in the linearity of the steady state characteristics of the turbinemeter used as an air flow rate sensor. The corresponding improvement in accuracy is important since the mean deviation has been reduced from 488 m^3/h to 91 m^3/h which value is 1.8% of the full scale (5000 m^3/h). Since the maximum deviation is occuring for the very low air flow rates (see Figures 6 and 7), the reducement of this value might be of importance for automation of the setpoints of minimum ventilating rate.

4.6 Pressure drop caused by the freely rotating impeller of the turbine flowmeter.

The air flow rate produced by an axial fan at a certain voltage supply is very sensitive to the static pressure to overcome by the fan as seen from the steady state characteristics of the fan. Consequently it is important to minimize the pressure drop caused by the air flow rate sensor positioned in front of the fan. Therefore the influence of a turbinemeter, used as an air flow rate sensor, on the fan characteristics was measured. More specifically, since feedback control of the air flow rate will be used, it is important to know what maximum air flow rate can be produced by the fan.
As already indicated, the testrig can be used to determine fan characteristics. This implicates the measurement, at various flow rates, of the power required to establish a certain pressure difference. The same pressure and flow rate intervals as in the foregoing were covered to realize a determination of the saturation values (the flow rates at maximum voltage supply of 220 V to the fan). The results are given in Table 4. Following observation can be made : the introduction of the turbinemeter in the measurement system has but a small effect on the realised maximum flow-rates. Maximum reduction of saturation values amounts to no more than 150 m^3/h.

4.7 Influence of the type of axial fan on the characteristics of the freely rotating turbine.

Since the efficiency of a turbinemeter is influenced by the flow pattern as the air reaches the blades, it can be expected that the type of fan (different blades, etc.) does influence the characteristics of the turbinemeter. Therefore the turbinemeter was subjected to an identical series of tests, the only modification being the type of axial fan used. Instead of the Woods-2100, a Siemens 2CC2 fan, one of the more competitive types on the market, was chosen.These results will be compared with the corresponding measurement data under 4.5 for the Woods-fan. Figure 8 and Table 5 give the result of a linear regression analysis. The whole flow rate range has been taken in consideration, pressure differences varying between 0 and 120 Pa. Compared to the Woods-2100, the mean deviation has been further reduced from 91 to 60 m^3/h which is 1.2% (instead of 1.8 % see 4.5) of the full scale (5000 m^3/h). In addition, it can be noticed from Table 6 that the reduction of saturation values for the Siemens-fan if combined with the turbinesensor is of an equally small measure as for the Woods. Maximum reduction amounts to 250 m^3/h.

5. CONCLUSIONS

5.1 The relationship between the rotational speed and the flow rate for the new developed turbinemeter is highly linear (R^2 = 0.995). This implies a flow rate measurement of satisfying accuracy, nearly independent of prevailing pressure differences (Mean deviation from regression-predicted values = 91 m3/h or 60 m3/h, depending on the type of axial fan in use). The comparable accuracy of the previous turbinesensors had the poor value of about 600 m^3/h.
It must be noted that this turbinemeter can be used as a ventilating rate sensor in a chimney with a diameter of 0.5 m and a length of 1 m and that this sensor then permits to measure the ventilating rate with an accuracy of 60 m^3/h in a range from 200 to 5000 m^3/h and this for pressure differences from 0 to 120 Pa. The price of each prototype was around 220 U.S. dollar and should decrease for bigger series.

5.2 The free running impeller of the flow rate sensor causes only limited pressure losses. Saturation values of the fan suffer a reduction of maximum 250 m^3/h.

5.3 The behaviour of the flow rate sensor is dependent of the type of axial fan in use. This is demonstrated by comparing the linearity of the relationship of rotational speed versus flow rate and this for the flow rate sensor in combination with two different axial fans (Woods-2100 and Siemens 2CC2, diameter 0.45 m). The flow rate sensor will only operate satisfyingly when combined with an appropriate fan. In that case the improved turbine will not give the higher mentioned problem of changing its direction of rotation.

NOTATION

g	accelaration due to gravity = 9.81	m/s^2
n	number of revolutions	$1/s$
n_s	specific number of revolutions	
p	pressure	Pa
ΔP	pressure difference	Pa
r	root mean square of inner and outer radius of flow passages at turbine blading	m
Q	volume flow rate	m^3/s
v	absolute velocity of fluid flow	m/s
w	relative velocity of fluid flow	m/s
u	peripheral velocity of rotor blade	m/s

greek symbols

β	blade angle	grad.

subscripts

a	axial component
θ	rotational component
1	inlet section
2	outlet section
in	outside
out	inside

Table 1: Air flow rates (m^3/h) as measured by a 2-, 4- and 8-blade turbine with a same fan at different voltages to the fan and for different pressure differences (ΔP) to overcome.

ΔP (Pa)	0	19.6	39.2	58.9	78.5	98.1	117.7
(Volt)	turbine with 2-blade impeller						
100	1600 m^3/h						
140	3800	2700	1500				
180	5500	5200	4800	4200	3600	2000	
220	5500	5500	5500	4500	4000	3000	1500
(Volt)	turbine with 4-blade impeller						
100	1900						
140	3500	2500	2100				
180	5100	5000	4500	4000	3500	2100	2000
220	5500	5000	5000	4500	3900	2000	2000
(Volt)	turbine with 8-blade impeller						
100	1600						
140	3100	2300	1700				
180	4600	4500	4100	3600	3200	2000	1600
220	5000	4500	4500	4000	3500	2000	2000

Table 2 : Variation of inlet blade angle β_1 with radius for the prototype; design/point : 500 m2/h - 250 rev/min.

V (m)	Va (m/s)	V (m/s)	β_1 (°)
0,01	0,87	0,202	17°
0,02		0,524	31°
0,03		0,785	42°
0,04		1,047	50°
0,05		1,309	56°
0,06		1,571	61°
0,07		1,833	64°
0,08		2,094	67°
0,09		2,358	69°
0,010		2,620	71°
0,011		2,880	73°
0,012		3,142	74°
0,013		3,403	75°
0,014		3,665	76°
0,015		3,927	77°
0,016		4,184	78°
0,017		4,451	78°

Table 3 : Results of the regression analyses of rotational speed of (the previous and the new) turbinemeters as a function of the air flow rate (from 0 to 5000 m^3/h) and for pressure differences from 0 to 120 Pa.

	Previous Sensor	Prototype
R^2	0.672	0.992
Mean (m^3/h)	488	91
Max (m^3/h)	1267	346

Table 4 : Effect of the turbinemeter to the maximum air flow rates produced by a Woods 2100 fan at a voltage of 220V.

ΔP (Pa)	with turbinemeter (m^3/h)	fan without meter (m^3/h)
0	5550	5550
19.6	5300	5300
39.2	4950	5050
58.9	4550	4650
78.5	4050	4200
98.1	2450	2600
117.7	1950	2000

Table 5 : comparison of the R^2 and the mean deviation of the turbinemeter characteristics if combined with two different types of fans for a flow rate range of 200 to 5000 m^3/h and pressure differences from 0 to 120 Pa.

	Woods-2100	Siemens-2CC2
R^2	0.992	0.995
Mean	91	60

Table 6 : Effect of the turbinemeter to the maximum air flow rates produced by a Siemens 2CC fan at a voltage of 220V.

ΔP (Pa)	with turbinemeter (m^3/h)	fan without meter (m^3/h)
0	5025	5225
19.6	4625	4800
39.2	4125	4350
58.9	3500	3700
78.5	2125	2300
98.1	1250	1575
117.7	750	1000

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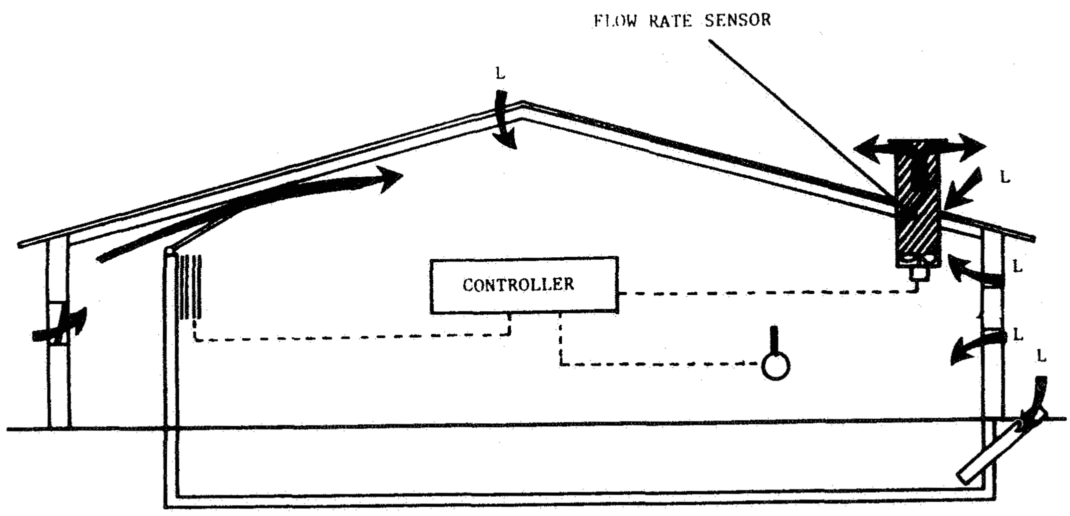


Figure 1. Place where the flow rate sensor should be mounted because of possible leakages in the building envelope (L: Leakage).

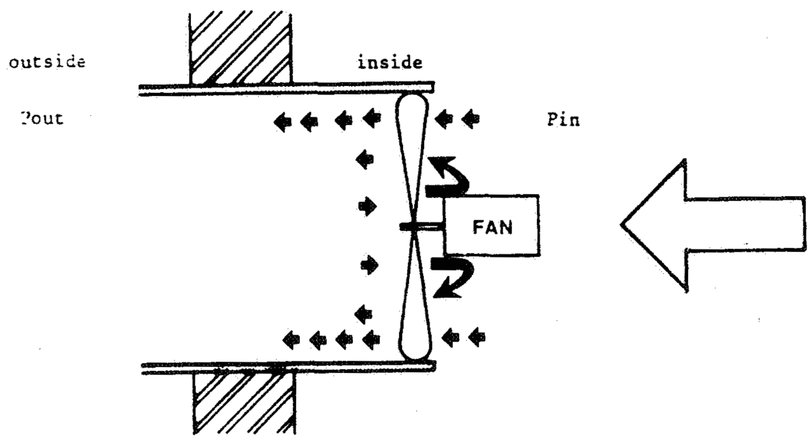
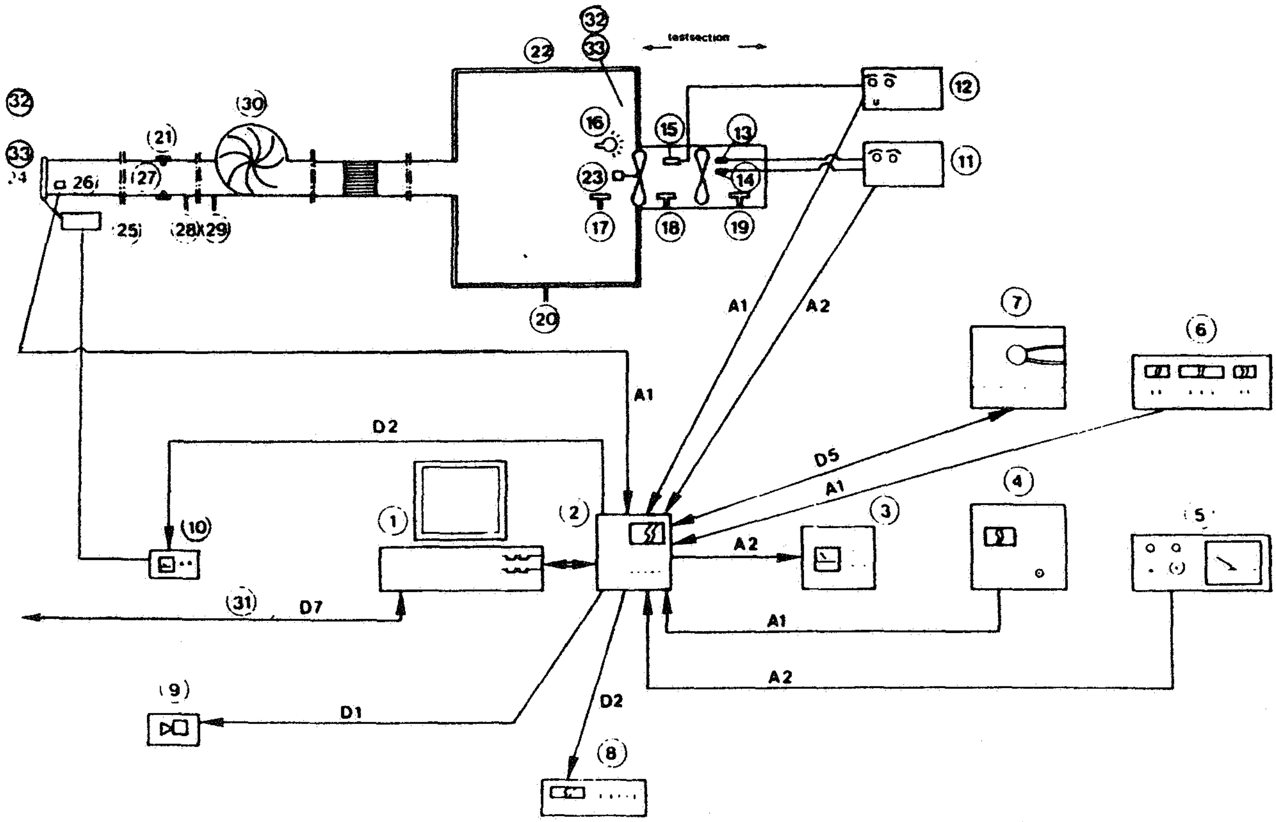


Figure 2. Back flow effect of an axial fan ($P_{in} << P_{out}$).



1. Minicomputer for controlling the test rig and storing results.
2. Parallel-coupled measure and control interface for digital and analogue signals.
3. Linearised and filtered transmitter 0-10 V_{DC} to 0-220 V_{AC}.
4. Pressure-sensor 0 to 26 Pa with digital display and analogue output (Furness control). (Furness Control 040 ; Accuracy 1 Pa)
5. Reference pressure-sensor for calibration 0 to 200 Pa with analogue output (Furness control).
6. Power-meter 0-200 kW, analogue output.
7. Controller for scanning 10 pressure-channels.
8. 3-channel digital voltmeter.
9. Digital controlled acoustic signal.
10. Unit for flow rate control.
11. Transducer for fan speed and direction measurements of turbinemeter
12. Transducer for fan speed measurements of fan
13 and 14: proximity switches.
15. Photocell. (Sprague ULN 333 Y)
16. Light resource.
17, 18, 19, 20: Location of measuring points for pressure-measurements.
21. Tubes to produce pseudo-laminar air flow.
22. Testchamber.
23. Testfan
24. Diaphragm for air flow rate control.
25. Motor for diaphragm control.
26. Sensor to measure the position of the diaphragm.
27. Shaped orifice.
28, 29 Pressure-measurement over the orifice. Furness Control FC040 (range 0-2500 Pa, Accuracy 10 Pa)
30. Auxiliary fan.
31. Serial data-transmission to Tektronix 4052 A graphic workstation.
32. Humidity sensors (Vaisala HMP 23U; Accuracy <3% RH)
33. Temperature sensors (Analog Device AD 590 ; Accuracy 0.5°C)

Figure 3. Schedule of the testrig.

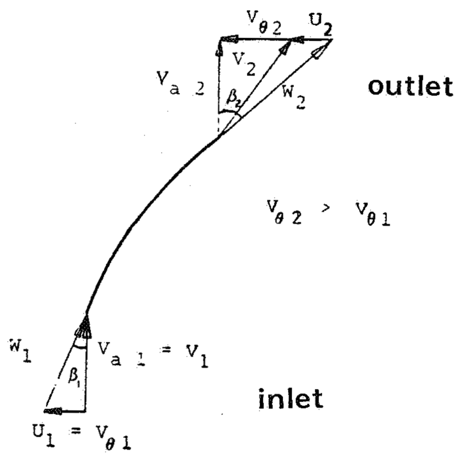


Figure 4. Velocity triangles at inlet and outlet of a turbine blade

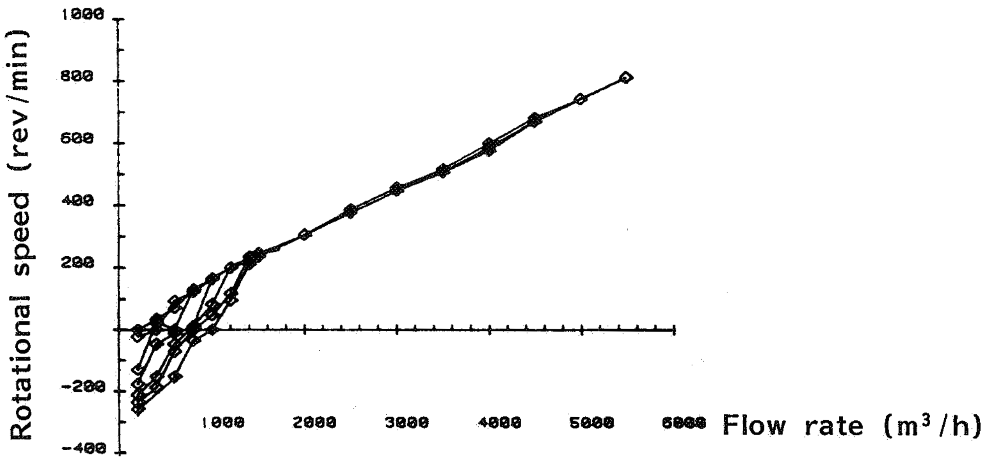


Figure 5a. Rotational speed vs. flow rate for a 2-blade impeller, downstream position.

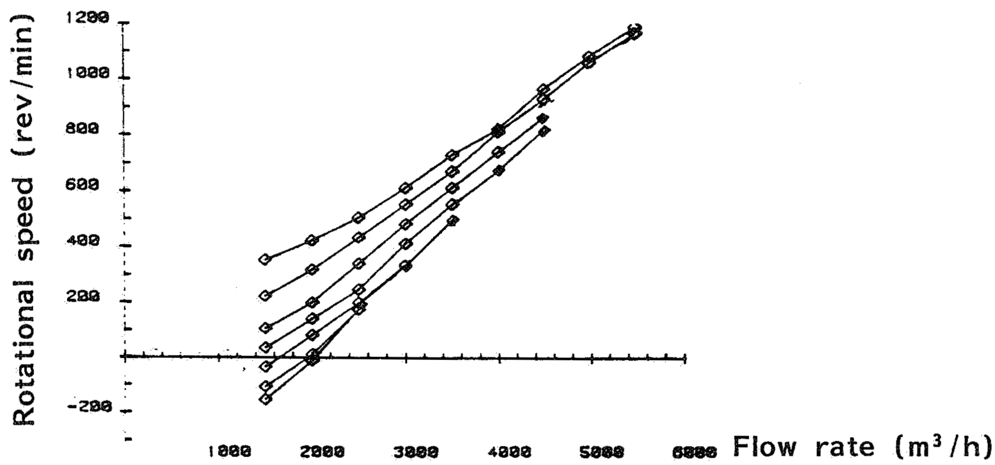


Figure 5b. Rotational speed vs. flow rate for a 2-blade impeller, upstream position.

Figure 5. Comparison of turbinemeter performance (old design) for upstream and downstream positioning (flow rate : 1500-5000 m^3/h; Δp = 0-120 Pa).

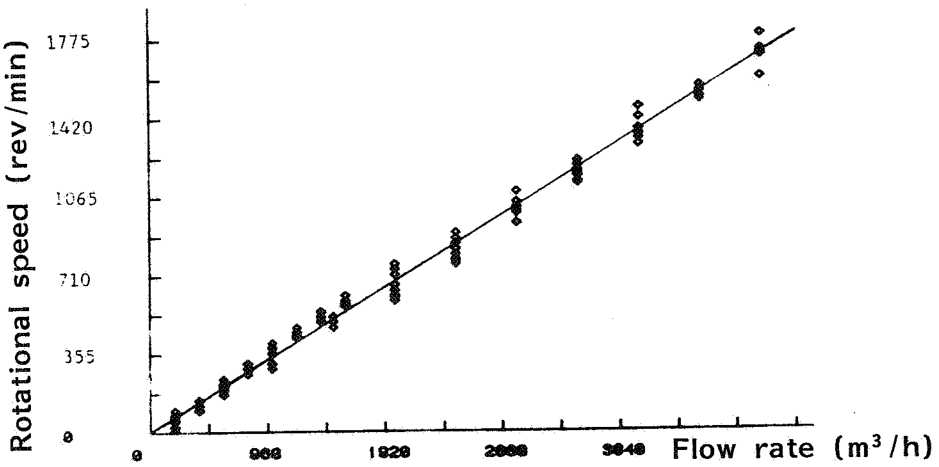


Figure 6. Linear regression analysis. 2-blade impeller prototype (flow rate : 200-5000 m^3/h, Δp : 0-120Pa), upstream position, combined with Woods 2100-fan.

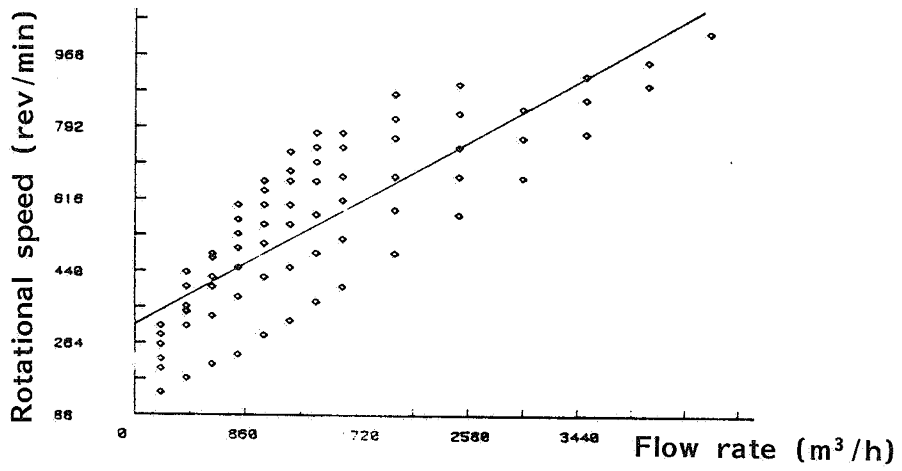


Figure 7. Linear regression analysis. Turbinemeter old design (flow rate : 200-5000 m^3/h, Δp-120 Pa), downstream position, combined with Woods 2100 - fan.

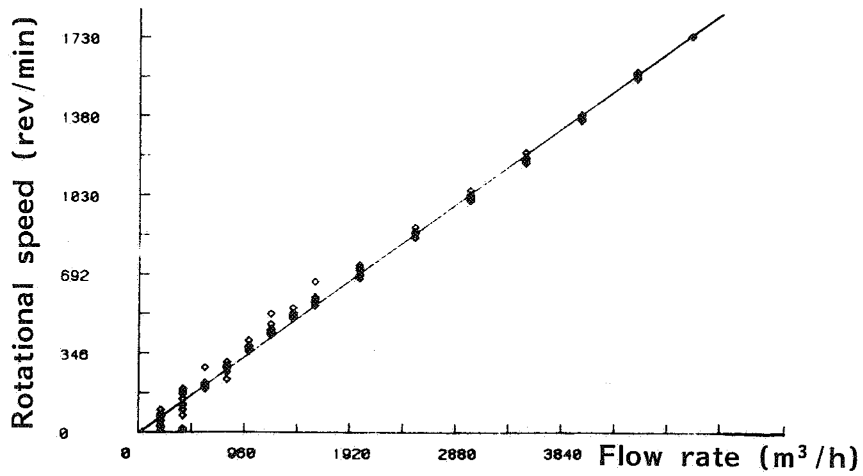


Figure 8. Linear regression analysis. 2-blade impeller prototype (flow rate : 200-5000 m^3/h, Δp : 0-120Pa), upstream position, combined with Siemens 2CC2 - fan.