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The use of axial-fan for the measurement of flue gas volume flow

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Summary

This study is based on the previous studies on power plant emission monitoring carried out by Indmeas Oy. The company has been searching for more accurate, reliable and cost-effective ways to determine the volume flow of flue gas in stack. The target of this study is to test the accuracy of a linearized model for an axial-fan and to estimate the measurement uncertainty of volume flow determined by this model.

In this study the linearization error of the axial-fan curve was evaluated by comparing model based values to the corresponding fan curve values. Within a typical operation range of the axial fan the linearization error is 1 – 2 % of the Q value. Moreover, the model parameters A, B and C were estimated by using volume flow values and total pressure difference values for which the measurement uncertainty is known. Then the value for Q was calculated and its uncertainty was determined. It was observed that Q value can be determined with the measurement uncertainty of about ± 5 % (2σ).

This study has been done during the 1st year of the MMEA research program and it is one of the tasks in WP4.2.2 Measurements in extreme conditions.

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1 Background

This study is based on the previous studies on power plant emission monitoring carried out by Indmeas Oy. The company has been searching for more accurate, reliable and cost-effective ways to determine the volume flow of flue gas in stack. The target of this study is to test the accuracy of a linearized model for an axial-fan and to estimate the measurement uncertainty of volume flow determined by this model.

During this project also other aspects of industrial fans were tentatively considered. Interesting questions are related to the energy efficiency of industrial fans and to practical means to improve the efficiency. Moreover, the monitoring of changes in efficiency could be used for proactive maintenance and energy saving in the use of fans.

The background information of this study consists of a theoretical and experimental analysis of fan models /1./, the standard for industrial fans /2./, a handbook for fans /3./, and of the accumulated experience of Indmeas Oy personnel in power plant measurements.

This study has been done during the 1st year of the MMEA research program in workpackage 4 Particles and Emissions.

2 Axial-fan model

2.1 Axial-fan curve versus radial fan curve

As stated by Yli-Juuti /1./ the operation of different fan types can be governed by the affinity rules. In the case of axial fans these rules are, however, more complicated than those for radial fans. This is caused by the more complicated flow field created by axial fans and thus there is no simplified expression for the volume flow (Q) as a function of blade angle (Θ) and pressure difference (Δp). This must be overcome by the use of approximate equations which describe the relation between Q and the parameters Θ and Δp. These equations can be created by the experimental data provided by fan manufactures. Fan specific information in the form of fan curves is provided by fan manufactures at least for the larger fans in stacks or in scrubber units.

The use of axial-fan curves is further complicated by the inaccuracies in the measurement of Δp during field experiments. These issues include malfunctions due to blockage of pressure connections, averaging pressure data over duct area, and sometimes due to limited space available for proper pressure measurements. The uncertainty of blade angle determination is typically $\pm 0{,}5^0$ (2σ, blade angle control range about $10^0 \ldots 30^0$) which corresponds to about 1-5 % relative uncertainty.

It is expected that the combination of these uncertainties leads to a larger measurement uncertainty in the case of axial-fans in comparison to radial fans.

In the following lines the Monte-Carlo method is used for the estimation of measurement uncertainty for axial-fans. The only reason for using this method is that it simplifies the calculation of uncertainties in cases where several models has to be compared as the writing of partial derivatives is eliminated. The data analysis is done with Crystal Ball® software /4./.

2.2 Estimation of model error in Axial-fan curve

As stated by Yli-Juuti /1./ the axial-fan curve can be approximated by a linearized version of the measured fan curve. Moreover, the expected changes in the plant curve are small.

Yli-Juuti proposes to use the following linearized fan curve set :

$$Q = A \cdot \Theta + B \cdot (1 + \alpha \cdot \Theta) \cdot \frac{\rho_0}{\rho} \cdot \Delta p + C \text{, where} \qquad [1]$$

A, B, C and α are constants to be determined by a calibration measurement, and $\alpha \cdot \Theta$ deals with the small flattening of the fan curve at large values of Θ. The term ρ_0/ρ compensates the difference in gas densities between the reference plant curve measurement and the measurement when the formula is used for flow determination.

In this model, two of the parameters (B and α) are, however, correlated as they are multiplied by each other in the term $B \cdot (\alpha \cdot \Theta) \cdot \frac{\rho_0}{\rho} \cdot \Delta p$. This leads to convergence problems in the estimation of the parameters. Moreover, the parameter α seems to be artificial as the blade angle itself takes care of the flattening of the fan curve in the model.

The model can thus be simplified as follows :

$$Q = A \cdot \Theta + B \cdot \Theta \cdot \left(p_{tF}^{\exp} \cdot \frac{\rho_0}{\rho_{\exp}} - p_{tF}^{0} \right) + C \text{, where} \qquad [2]$$

p_{tF}^{0} is the total pressure difference in the reference plant curve measurement,

$p_{tF}^{\exp}$ is the total pressure difference during the actual use of the fan as flow meter. The gas density ratio $\frac{\rho_0}{\rho_{\exp}}$ corrects the possible change in gas density caused by changes in gas composition, absolute pressure or in temperature.

In this theoretical study the error induced by the use of the linearized axial-fan curve is estimated as follows :

- Firstly three plant curves are created as there are no experimental data available at the moment. These curves should cover a typical range of the operational values of an axial-fan during the expected life-span of the fan.
- Secondly the model parameters {A, B, C} are estimated by the least-square method (SigmaPlot®) by using (Q, Δp) values at the intersection points of the plant curves and axial-fan curves given by the fan manufacturer. Here 12 points at the intersections of the 3 plant curves and fan curves at $\theta=10^o$, 15^o, 20^o, 25^o as data points for the estimation of A, B and C.

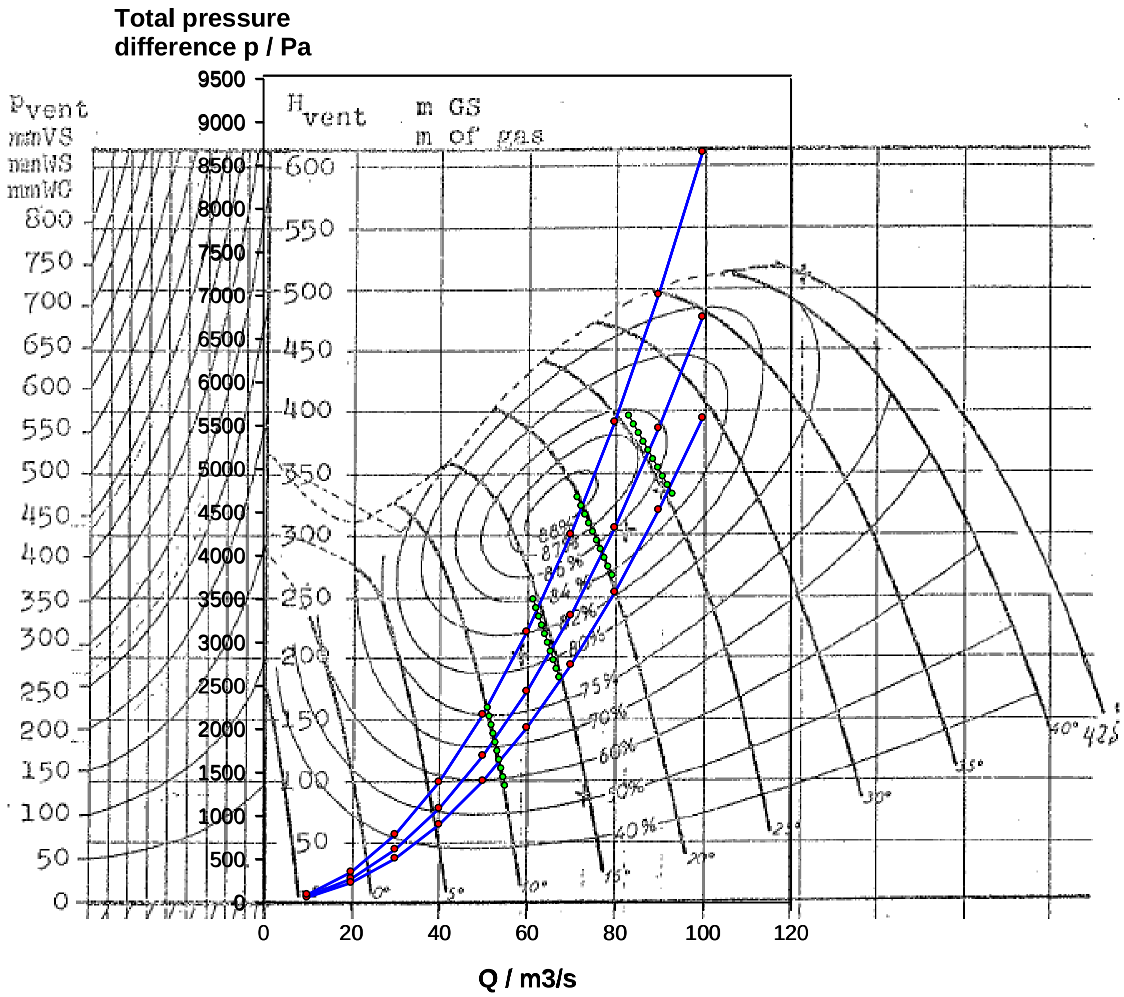
- Thirdly the flow values given by the linearized model Q_{model} is compared to exact values Q_{fan_curve} obtained from the fan curve. The difference (in percentages) between Q_{model} and Q_{fan_curve} is compared and it is given at different Δp values.

The plant curves used here are as follows :

- basic line : $p = 0.6747 \cdot q^2$, where p is the total pressure drop in the pipeline, and q is the volume flow
- second line : $p = 0.5585 \cdot q^2$, at 460 Pa lower than basic line
- third line : $p = 0.8651 \cdot q^2$, at 530 Pa higher than basic line

In Fig. 1. the plant curves are marked by blue curves, and the fan curves for an axial-fan (700 kW, Variax ASM-1848/1120 BP43, Helsingin Energia Oy, Hanasaari B) are shown by the black lines. Plant curves cover a typical range of the operational values of an axial-fan during the expected life-span of the fan : total pressure increase/drop in the stack caused by fouling or switching on and off scrubber and precipitator units.

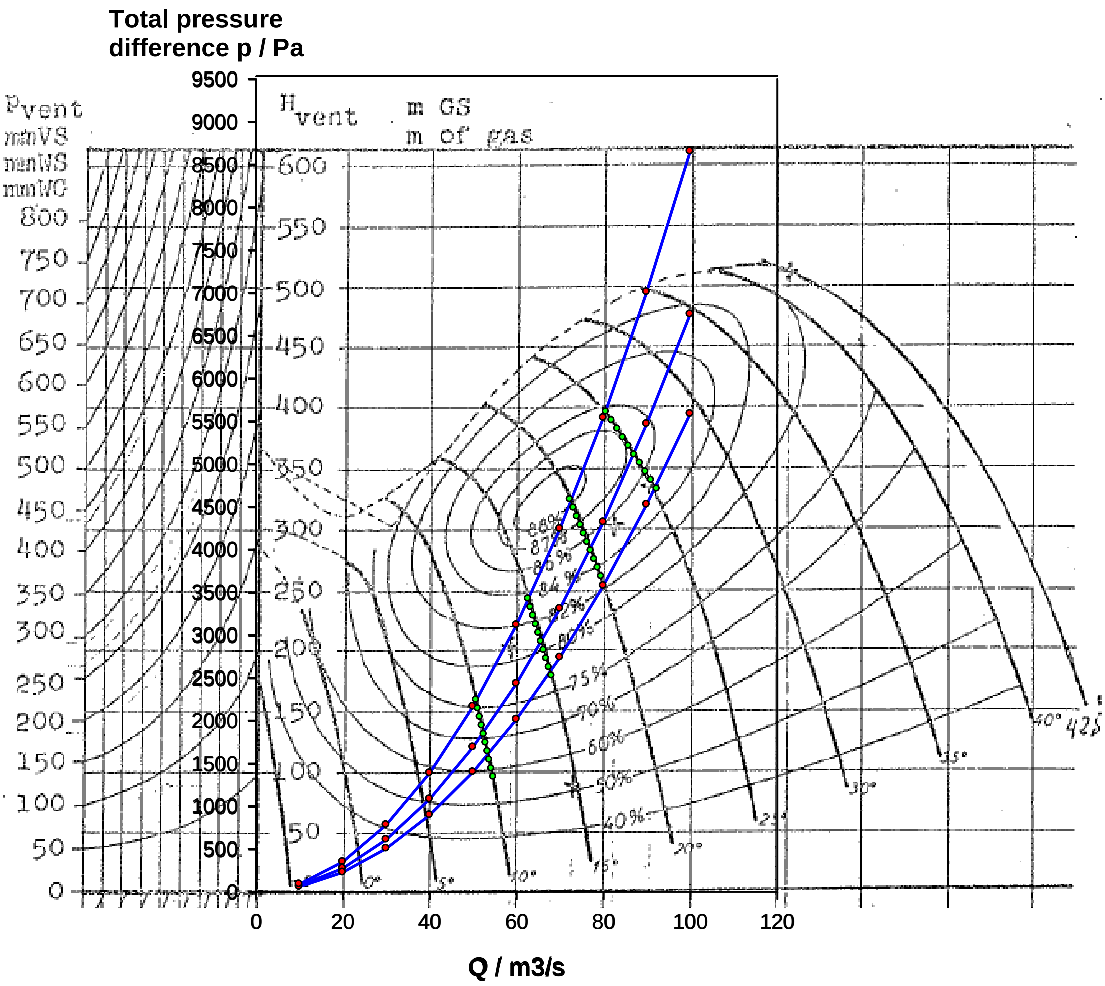
Fig. 1. Qualitative estimation of the linearization error.



In Fig. 1 the green dots indicate the results of the linearized model (formula 2). The set of four green linear lines at blade angle θ=10°, 15°, 20°, 25° is obtained by using the p_{tF} and Q values of the θ=10° curve for the parameter B in formula 2. This estimation shows that the error in volume flow values when using the linearized model is ≤ 1 % if $|\Delta p| < 500$ Pa at 10°<θ<20°. However, at θ=25° the difference in Q-value is about 2%. The error induced by the use of the linearized model can be made smaller if more experimental data is available. The fan curve data has been available here only in pictorial form and not in digital form. It should also be noted that the original fan curve picture is slightly deformed as can be seen eg. in the Q and p_{tF} axes.

An improved set the linearized model parameters can be obtained if more experimental data is available. If the parameters C and B are separately calculated for each θ-value the fit of the linearized model improves as shown in Fig. 2. In this case the error in Q-value is about ≤0.5 % or the line width of the graph.

Fig. 2. Improved fit of the linearized model



The generality of this result can be improved by checking the situation with other axial-fans. However, it seems to be difficult to obtain exact information on fan curves in digital form. Moreover, the manufacturer has not provided any uncertainty estimates to the fan curves.

The error induced by the use of the linearized model will be later added to the estimate of the total measurement uncertainty. The errors in the linearized model can be further decreased if more experimental data can be obtained during the reference plant curve measurement. There will be a compromise between the costs of reference measurements and the accuracy of the linearized model.

2.3 Total measurement uncertainty of Q by using the linearized axial-fan model

The next step is to estimate the total uncertainty of volume flow Q as the uncertainties of parameters A, B, C and the measured values p_{tF}^{exp}, p_{tF}^{0} and Θ are now known. The measurement uncertainty estimates are calculated with Crystall Ball® software.

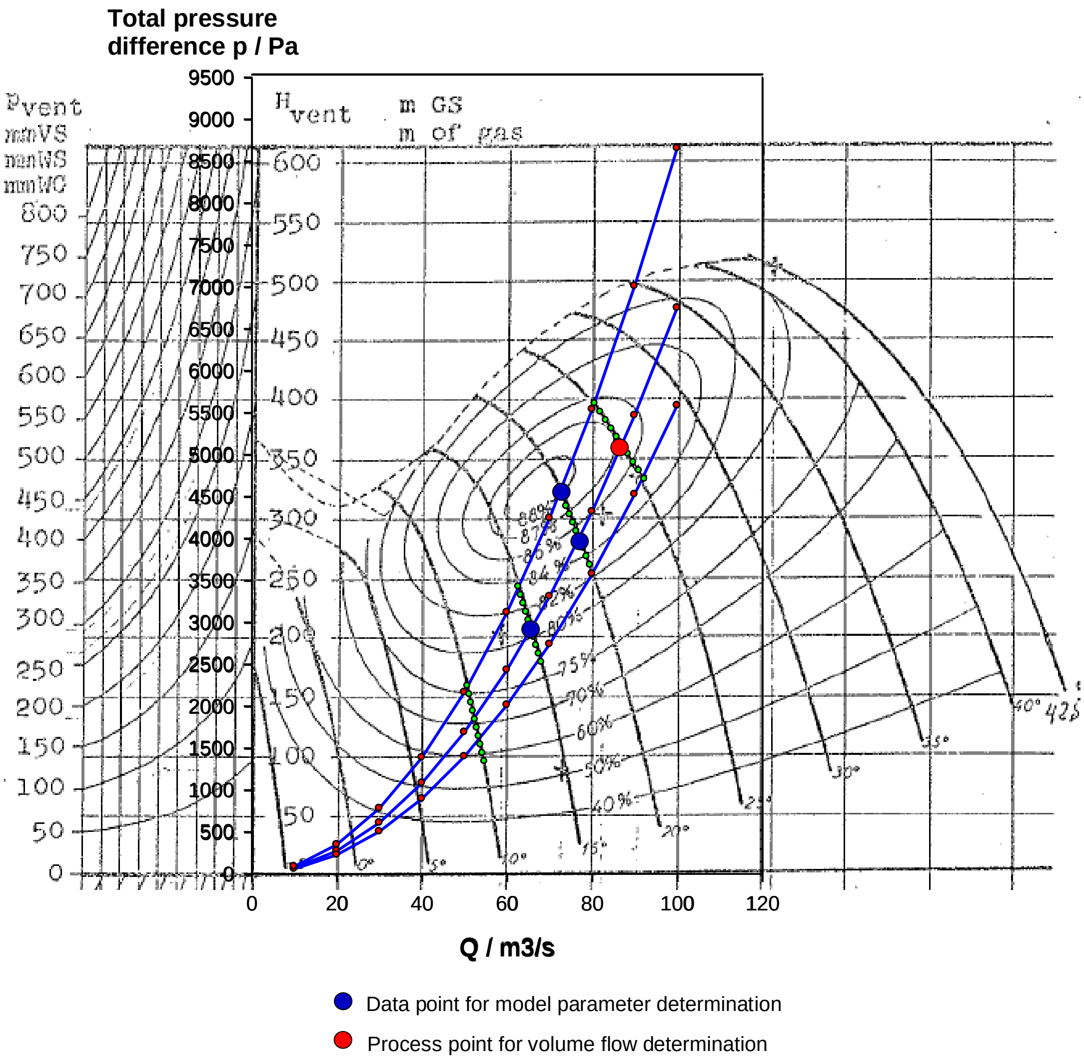
The following uncertainty values will be used in the calculation :

- ΔQ= 1 % (2σ , based on /1./)
- ΔΘ = 0.5 ^{0} (2σ , based on /1./)
- Δp_{tF}^{exp} = 10 Pa (2σ, based on /1./)
- Δp_{tF}^{0} = 10 Pa (2σ, based on /1./)
- ΔA = 0.06 - 0.19 (σ, based on SigmaPlot® refinement result)
- ΔB = 1.5e-5 - 5e-5 (σ, based on SigmaPlot® refinement result)
- ΔC = 1 – 3.4 (σ, based on SigmaPlot® refinement result)
- $\Delta \rho_0$ = 0.03 (typical value 0.87 kg/m^3 ± 0.03, based on estimated variation of moisture contents in flue gas, 12%-25%-vol, T=130^{0}C, p=1 bar)
- $\Delta \rho_{exp}$ = 0.03 (-“-)
- error induced by the use of the linearized model : 1-2 % (2σ) depending on the amount of data available for the estimation of the model parameters A, B and C.

The total measurement uncertainty of the volume flow depends on the number and range of individual measurements. The results will be given here for further comparisons in 3 scenarios.

Scenario 1.
- Linearized model is used for the axial-fan (formula [2])
- minimal amount of isotope flow measurements is used : the model parameter A, B and C are determined by 3 measurements at θ = 15^0 and 20^0 (see Fig. 3), two of them is done at the reference plant curve and one is done at higher pressure side by changing the plant curve by adding extra flow resistance. At θ = 15^0 only one flow measurement is done, and at θ = 20^0 two flow measurements are done. At θ = 20^0 the pressure difference is zero at the plant curve, and it has a specific value at the higher pressure side.
- The model parameters are as follows : A=2.25 ± 0.19 (1σ), B= -3.7e-4 ± 5e-5(1σ), C=31.8 ± 3.4(1σ)
- With these parameter values the volume flow and its uncertainty at θ=25^0 are as follows :
- **Q = 87.1 ± 6.0 (1σ) or ± 7 %** , or **Q = 87.1 ± 12.0 (2σ) or ± 14 %.**

Fig. 3. Determination of the linearized model parameters and volume flow



Scenario 2.

- Linearized model is used for the axial-fan (formula [2])
- the flow and pressure difference measurements are repeated for 3 times at the same points as shown in Fig. 3 to improve the estimate of flow and pressure difference.
- The model parameters as now as follows : A=2.25 ± 0.11 (σ), B= -3.7e-4 ± 2.9e-5(σ), C=31.8 ± 1.9(σ)
- With these parameter values the volume flow and its uncertainty at $\theta=25^0$ are as follows :
- **Q = 87.1 ± 3.6 (1σ) or ± 4 %** , or **Q = 87.1 ± 7.2 (2σ) or ± 8 %.**

Scenario 3.

- Linearized model is used for the axial-fan (formula [2])
- the flow and pressure difference measurements are repeated for 10 times at the same points as shown in Fig. 3 to improve the estimate of flow and pressure difference
- The model parameters as now as follows : A=2.25 ± 0.06 (σ), B= -3.7e-4 ± 1.5e-5(σ), C=31.8 ± 1.0(σ)
- With these parameter values the volume flow and its uncertainty at $\theta=25^0$ are as follows :
- **Q = 87.1 ± 2.2 (1σ) or ± 2.5 %** , or **Q = 87.1 ± 4.4 (2σ) or ± 5 %.**
- Here the error induced by the use of a linearized model should be included in total uncertainty estimate, and it becomes significant as at lower or larger θ values the error is 1-2 % as shown in Fig 1.
- The error in the use of the linearized model can be decreased if more experimental data points at different θ values can be used. If plant curve measurements can be done eg. at all the four angles θ = 10^0, 15^0 , 20^0 , and 25^0 , a simple interpolation of the model parameters at intermediate θ values could be used.

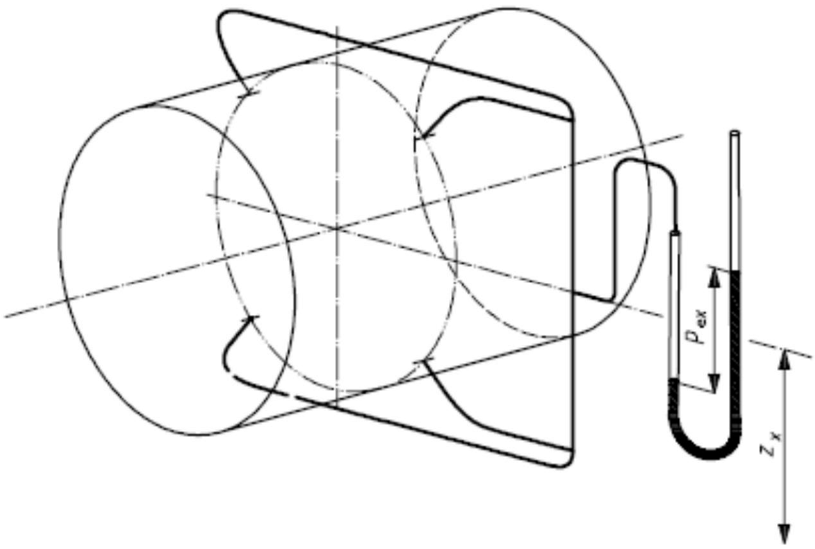
These scenarios show the expected measurement uncertainty of the volume flow Q with different amounts of repeated experimental data points. In power plants, it is, however, often difficult to carry out measurements with changed plant curve. In such a case the parameter B in the axial model must be determined by using the fan curve data alone.

2.4 Consideration of inaccuracies in θ, p_{tF} and ρ measurements

Blade angle θ
It is mentioned in /1./ that the uncertainty value for ΔΘ = 0.50 (2σ) can be obtained for an axial-fan if maintenance and calibration of the blade angle control unit is done regularly. It is, however, quite probable that the calibration of the angle adjustment unit can be done only once per year during the annual maintenance period. In this study the mentioned uncertainty for θ was used as there was no other information available for the specific axial-fan.

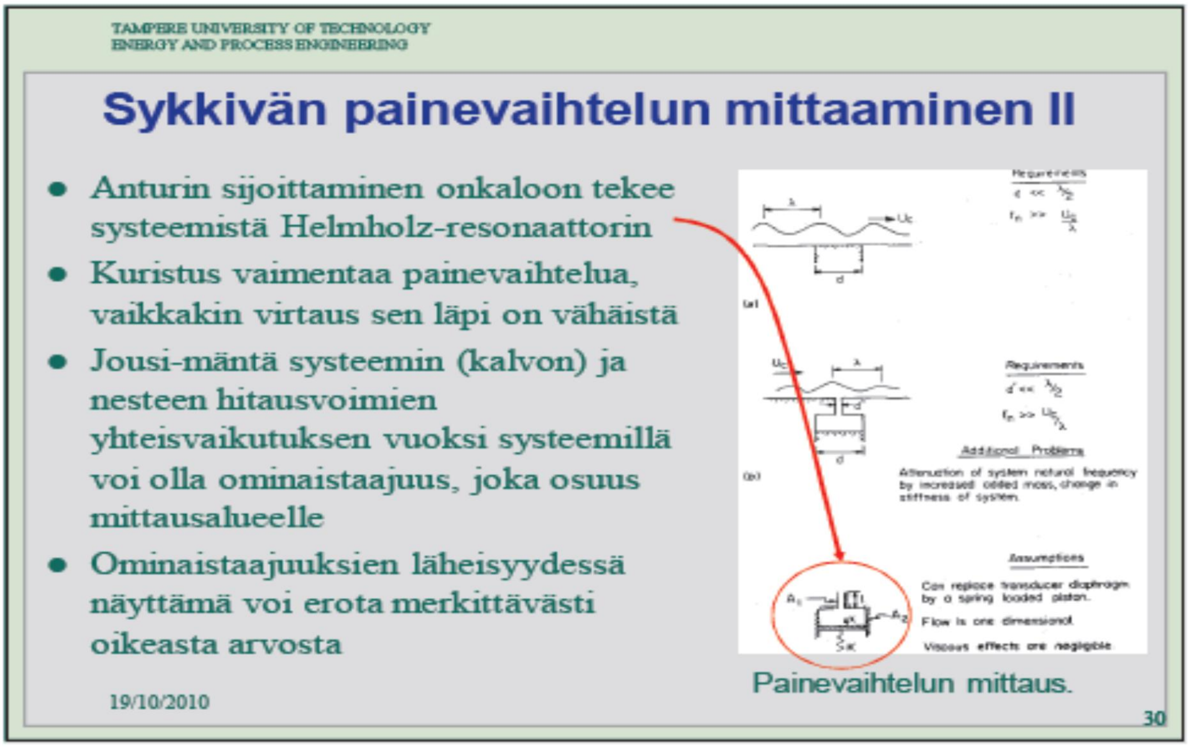
Pressure measurement
Field observations of Indmeas Oy show that the measurement of the pressure difference p_{tF}^{exp} may be disturbed by dirt and liquid in pressure tubes. This seems to be related to mechanical structure and positioning of the pressure tubes. Moreover, the pressure difference measurements may be inaccurate because of the single-point measurement method typically used in power plant conditions. A more accurate method would be the use of a four-point measurement as shown in Fig. 4. This method has been applied in the standard EN ISO 5801:2008 "Industrial fans. Performance testing using standardized airways" /2./.

Fig. 4. A four-point tapping connection for average static pressure measurement /2./



In plant conditions there may be also pulsating pressure changes caused by noise. This may be an additional source of error in pressure measurements. In 2004-2005 VTT carried out some laboratory experiments which were meant to demonstrate noise induced errors in pressure sensors. The experimental setup was developed to find an error-free coupling of a pressure sensor to a process tube. In this setup L- and S-type pitot tubes with pressure sensors and a loudspeaker were attached to a Φ= 250 mm tube. With the loudspeaker both sinusoidal noise and noise recorded in a power plant´s stack near a fan could be injected to the tube, and the response of the pressure sensor to the noise could be observed. In a test with a generated sinusoidal noise it was found that with L-type Pitot tube the error induced by the noise could be 15%, and with S-type tube about 40% depending on the intensity and the location of the noise source. The origin of these phenomena is related to the characteristic resonance frequency of a system consisting of a pressure sensor and a tube which connects the sensor to the process tube. Such a system acts as a Helmholz resonator. If there is noise in the process tube with frequencies near the resonance, the observed pressure value may differ significantly from the true value (see Fig. 5 /5./).

Fig. 5 Effects of pulsating pressure changes /5./



Gas density

The linearized model contains also the gas densities ρ_0 and ρ determined during the reference plant curve measurement and during the use of the fan as a flow meter, respectively. Typically in power plant conditions the density of the flue gas is not measured continuously but its value can be calculated by using fuel characteristics and other process data. It is thus interesting to know how sensitive the volume flow measurement is to density variation.

The effect of gas density variations to the uncertainty of volume flow values can be estimated eg. by allowing the density to vary between typical limits determined by the moisture contents. In biomass use the water contents in flue gas may vary between 12 %-vol - 25%-vol, and this induces the density variation for ρ = 0.90 kg/m3 – 0.84 kg/m3, or ρ = 0.87 kg/m3 ± 3.4% (T=130 ^{0}C, p = 1bar). This variation in density gives to the volume flow estimate the following value : 71.4 m3/s ± 0.8 (1σ), or ± 2% (2σ) variation at the upper blue point at θ=20^o in Fig 3. In the actual use of the linearized model the limits of variation of the volume flow must be determined by using the plant specific variation in flue gas density. Other aspects related to density variation such as ambient air pressure, changes in gas composition, fuel changes and process changes have been treated by Yli-Juuti /1., pp 52-55/.

The effect of combustion air moisture to flue gas moisture or to flue gas density is minimal. Even with a fuel containing 0.3 w-% water the change of combustion air moisture from 10 % to 70 % (relative humidity) increases the flue gas moisture from 10 % to 11 %, and decreases the flue gas density by 0.5 %.

3 Conclusions

In this theoretical study the data of a specific axial-fan (Variax ASM-1848/1120 BP43) was used here as the base for the calculations.

Firstly, the linearization of the axial-fan curve was tested by comparing a calculated value to the corresponding fan curve value. An obvious deficit of the previously proposed model for axial-fan was removed, and a simple linearized model is used here. Within a typical operation range the linearization error (or the difference between the calculated value and the fan curve value) is 1 – 2 % of the Q value. Secondly, the model parameters A, B and C were estimated by using volume flow values and total pressure difference values for which the measurement uncertainty is known. Then the value for Q was calculated and its uncertainty was determined. It was observed that Q value can be determined with the measurement uncertainty of about ± 5 % (2σ). This result should be compared to the radial fan case where the measurement uncertainty is estimated to be about ± 2 % (2σ) /1./. The main reason for this difference is the more complicated flow field of axial-fan and the larger uncertainty of the blade angle value. For the axial-fan type the total measurement uncertainty of Q is mainly determined by the number of repetitions of Q and p_{tF} measurements and the number of data points at different θ value.

This task will be continued during the 2nd year of the MMEA research program. The task will contain field measurements and the use of axial-fan model for the determination of volume flow in stack.

4 References

1. Erkki Yli-Juuti, Puhaltimen mallintamiseen perustuva savukaasujen virtausmittaus, Lisensiaatintyö, TKK, 1995
2. EN ISO 5801:2008 standard, "Industrial fans. Performance testing using standardized airways”
3. Olli Pukkila, Puhallintekninen käsikirja, Ilmateollisuus Oy, 1986
4. Oracle Crystal Ball, Fusion Edition, release 11.1.2.0.00
5. Pentti Saarenrinne, Paineen mittaus, www.tut.fi/units/me/ener/kurssit/ENER-1100/.../Paineen%20mittaus.pdf, TUT, 2010