Investigation of noise generation in ventilation systems

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Abstract

The ventilation systems are becoming more compact so as to save buildings space. In return, the velocity, in ventilation units, is increased and the air flows are also more turbulent. The hotwire investigation aims to correlate this increase of turbulence with the increase of the noise level.

The investigation was done for industrials purposes and is based on empirical researches completed with theoretical knowledge.

For the investigation, the hotwire sensor is chosen for its ability to detect small velocity fluctuations at high frequency. Two different prototypes are designed in order to highlight the influence of the turbulence level in the sound generation, especially at ventilation outlets. A procedure is also introduced, in which the hotwire is used for the turbulence measures and a reverberation room for the sound measurement.

General conclusions are finally identified and explain the influence of the turbulence in the sound generation mechanisms. The influence of the prototypes geometries, on both the sound and the turbulence, is analyzed and the master thesis describes how the air flow velocity in ducts and the static pressure could modify both the turbulence and the sound levels.

The stated conclusions imply that the designers of new ventilation systems should take into account the turbulence generated by their experimental product if they want to conserve good sound properties.
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1 INTRODUCTION

Space in modern buildings is more and more valuable and it requires designing more compact ventilation system to save space. Due to this consideration, the duct section and outlets are also smaller and it increases significantly the noise in the ventilated rooms. FläktWoods wants a better understanding of how the flow and turbulence close to the ventilation outlets influence the sound generation in order to design more quite future systems.

The investigation on the noise generation due to turbulences needs to start by a theoretical approach dealing with the characterization of turbulence in ducts, more precisely orifices and expansions in ducts.

The hotwire system is used to measure the fluctuations of velocity in a certain direction depending of the used probes and of their positioning. These fluctuations can be determined for very high frequencies. This system is a good and accurate solution for measuring velocity and characterizing turbulences at high frequency. The hotwire tool will be studied in order to evaluate the turbulence level in the ducts.

Secondly, the use of the reverberation room, for measuring the sound power level and the hotwire system will give us the opportunity to state the influence of turbulence on the noise and in that effect, a signal analysis will be done to quantify the noise generation due to turbulence.

In this thesis, we consider, according to the geometry of the ventilation outlet designed by FläktWoods, a duct with nozzles. Two different geometries are studied. One is a normal duct or straight duct with nozzles and another with a duct expansion before the nozzles. The turbulence, created by the duct configuration, is different depending of the case and this effects the sound generation (Figure 1). For instance, we can easily hear during the measurements that the expansion increases considerably the sound power level (SPL) in the ducts. We need to explain, with theory and practice, this statement of fact to clearly establish the main noise mechanisms involved in the sound we can hear.
The thesis is organized as follows. We will start by explaining the theory on turbulence and the noise generation mechanism involved. Then the measurement method will be described in detail and the hotwire probe, the cornerstone of the project, will be scrutinized. The results synthesis and the analysis will be done in Chapter 4. The main conclusions are made in this part and will be summarized in the conclusion section.

To conclude this introduction, the initiative for the project was taken by FläktWoods, in cooperation with KTH (Kungliga Tekniska Högskolan). Half of the study was done in the FläktWoods manufacturing site based in Jönköping (Sweden) where FläktWoods provided the materials and assistance in order to realize the different prototypes and the measurements.
2 THEORY

2.1 TURBULENCE

A flow is characterized as turbulent when the velocity field, \( U(x, t) \) is random. The term “random” means that if we make an experiment measuring the velocity of the turbulence flow, we shall get different results each time.

The deterministic nature of the classical mechanics is not involved. The Navier-stokes equation (for incompressible fluids) gives us the pressure and the velocity field for a flow by using the following formula (Munson, Young, & Okiishi, 1997):

\[
\frac{\partial \vec{v}}{\partial t} + (\vec{v} \cdot \nabla) \vec{v} = -\frac{1}{\rho} \nabla p + \nu \nabla^2 \vec{v} + \vec{f} \tag{2.1}
\]

where \( p \), the pressure(Pa), \( \vec{v} \), the velocity vector(\( m \cdot s^{-1} \)), \( \rho \), the mass density(\( kg \cdot m^{-3} \)), \( \nu \), the cinematic viscosity(\( m^2 \cdot s^{-1} \)), \( \vec{f} \), the force density(\( N \cdot kg^{-1} \)).

For compressible fluids, we found other equations more complex, which are available in fluid mechanics books.

Despite the equation above, the solution remains unpredictable due to different reasons:

- In a turbulent flow, there are difficulties to avoid the perturbation in initial conditions, the influence of boundary conditions and the material properties.
- Turbulent flow is more sensitive to such perturbations (due to its chaotic nature).
- The statistic and the experimental approach seems a good compromise. We will learn more about the turbulence phenomena in 2.1.

2.1.1 ENERGY CASCADE PHENOMENON AND LENGTH SCALE

Kolmogorov described how energy is transferred from the larger eddies to the smaller one. The Reynolds’s number is very high, which is a characteristic of the turbulent flow and decrease with the eddies size until the Reynolds number become lower and the viscous forces can dissipate the energy.

FIGURE 2: RELATIVELY LOW REYNOLDS NUMBER, (POPE, 2000)
The Reynolds number can be calculated in general with the formula:

\[ R_e = \frac{UL}{v} \]  

(2.2)

Where \( U \) is the mean flow velocity and \( L \), a characteristics geometrical length.

From here, we can also write that the eddies of size \( l \) have a characteristic velocity \( u(l) \) and a timescale \( \tau(l) = \frac{l}{u(l)} \).

The largest eddies have a characteristic lengthscale \( l_0 \) which is of the same order than the flow scale \( L \), and \( u(l_0) \) is comparable to \( U \).

For turbulence characterization, there is another parameter commonly used, the turbulence kinetic energy defined as:

\[ k = \frac{1}{2} \langle u_i u_i \rangle \]  

(2.3)

Finally, depending on the size of the eddies, we have different length scale ranges illustrated with Figure 4.

Kolmogorov states in his first similarity hypothesis that: “in every turbulent flow at sufficiently high Reynolds number, the statics of the small scale motions \((l < l_{EI})\) have a universal form that is
uniquely determined by $v$ and $\epsilon$ “(Pope, 2000). Kolmogorov argued that all the information (about the direction and the geometry of the eddies) are lost as the energy passes down the cascade. That is an explanation to the universal characteristics developed by Kolmogorov. $l_{EI}$ is precisely the limit between the “universal equilibrium range” and the “energy containing range” and $l_{EI}$ is estimated as $\frac{l_0}{6}$ (for high Reynolds number). $l_{EI}$ is then the limit between the large scale anisotropic eddies and the small scale isotropic eddies and takes into account the Kolmogorov’s hypothesis of local isotropy, which states that: “at sufficiently high Reynolds numbers, the small-scale turbulent motions $(l \ll l_0)$ are statistically isotropic” (Pope, 2000).

In his second similarity hypothesis, Kolmogorov states that: “in every turbulent flow at sufficiently high Reynolds number, the statistics of the motions of scale $l$ in the range $l_0 \gg l \gg \eta$ have a universal form that is uniquely determined by $\epsilon$ independent of $v$” (Pope, 2000). It means that in this intermediated range, there is no dissipation by the viscous forces. Then a new length scale appears $l_{DI}$, and is estimated to be equal to $\approx 60 \eta$ with $\eta$, the size of the smallest eddies.

As seen in Figure 4, the universal equilibrium range is spitted into two subranges:

- The inertial subrange ($l_{EI} > l > l_{DI}$) where the viscous forces can be neglected and only the inertial effects determine the motions
- The dissipation range ($l < l_{DI}$) where the viscous forces dissipate the energy.

The three Kolmogorov hypothesis describe the different kinds of eddies which are in turbulences, and the different steps from the energy production to the dissipation. This is the cascade phenomenon.

Furthermore, in Figure 3, we can see there are different lengths scales depending on in which subrange we are. More information is given in (Pope, 2000) about the different length scales in the different subranges.

In the energy containing range, the larger eddies can be characterized by their lengthscale $l_0$ which is of the same order than the flow length scale $L$. The large eddies of size $l_0$ have a velocity $u_0$ and a timescale $\tau_0 \equiv l_0/u_0$ and the characteristic velocity $u_0 \equiv u(l_0)$ is comparable to the root mean scare (r.m.s) of the turbulence intensity $u' = \left(\frac{2k}{3}\right)^{\frac{1}{2}}$

Then $l_0 \equiv \frac{k^2}{\epsilon} \text{ where } \epsilon (m^2.s^{-3}) \text{ is the energy dissipation rate.}$

The Reynolds number of the large eddies is known as being written $Re_L$. The turbulence Reynolds number is defined as:

$$Re_L = \frac{k^2 l_0}{v} = \frac{k^2}{\epsilon v} \quad (2.4)$$

In the Dissipation range, the Kolmogorov scales describe the smallest eddies in the flow in which the energy is dissipated by the viscous forces.
These scales are formed using the two parameters \( \nu \) (used for the dissipation effect) and \( \epsilon (m^2.s^{-3}) \), the energy dissipation rate.

The Kolmogorov scales (Pope, 2000) can be formed as the following length, velocity and time scales:

\[
\text{length scale} : \eta = \left( \frac{\nu^3}{\epsilon} \right)^{\frac{1}{4}}
\]

\[
\text{velocity scale} : u_\eta = (\epsilon \nu)^{\frac{1}{4}} \rightarrow Re_\eta = \frac{\eta u_\eta}{\nu} = 1
\]

\[
\text{time scale} : \tau_\eta = \left( \frac{\nu}{\epsilon} \right)^{\frac{3}{2}}
\]

\[
\left( \frac{u_\eta}{\eta} \right) = \frac{1}{\tau_\eta}
\]

The Kolmogorov Reynolds number \( Re_\eta \) of the small eddies is equal to 1. It seems to be consistent with respect to the cascade phenomenon and the fact that eddies become smaller and smaller until the dissipation can occur at low Reynolds number.

Kolmogorov scales are used to scale and characterize the flow fields due to the fact that in the universal equilibrium range the flow fields are statistically identical for high Reynolds number and can be classified (and quantified) depending of the Kolmogorov scales.

In the inertial subrange, we have no dissipation. The energy is transferred to the smaller eddies. The Kolmogorov scales (Pope, 2000) can be formed as the following length scale:

\[
\lambda \approx \left( \frac{10 \nu k}{\epsilon} \right)^{\frac{1}{2}}
\]

\[
\lambda = \sqrt{10} \eta^{\frac{1}{3}} l_0^{\frac{1}{3}}
\]

Where \( k \) can be expressed as: \( k = \left( \frac{2}{3} \right) u'^2 \)

The Taylor-scale Reynolds number \( R_\lambda \) is defined as:

\[
R_\lambda = \frac{u'\lambda}{\nu}
\]

\( R_\lambda \) can be related to the turbulence Reynolds number:

\[
R_\lambda = \left( \frac{20}{3} Re_\nu \right)^{\frac{1}{2}}
\]

The energy transfer rate can also be defined as \( \tau(\ell) \) and illustrated below. The energy transfer rate in the inertial subrange is in that case equal to \( \epsilon \) and proportional to \( \frac{u(l)^2}{\tau} \). Figure 5 illustrates the notion of transfer of energy to successively smaller scales. The energy is transferred from the energy containing range to the dissipation range.
2.1.2 The energy spectrum

As said previously, we have defined \( k \) the turbulent kinetic energy. The most important things, we should try to measure, is the energy spectrum, in order to know how the energy is distributed among eddies of different sizes. The energy spectrum will be written, starting from here, \( E(\kappa) \) and corresponds to the energy of eddies of size \( l \) and wavenumber \( \kappa \) with \( \kappa = 2\pi/l \).

\( k \) is also given by the formula:

\[
k = \int_{0}^{\infty} E(\kappa) d\kappa
\]  
(2.10)

In the inertial subrange, we have seen that the second similarity hypothesis, due to the fact that there are no viscous effects, \( E(\kappa) \) can only depend on \( \kappa \) and \( \varepsilon \). By the following dimensional analysis, we get:

\[
\left[ \frac{2}{\varepsilon^{2} \kappa^{-5/3}} \right] = m^3 s^{-2}
\]  
(2.11)

And then:

\[
E(\kappa) = C \frac{2}{\varepsilon^{3} \kappa^{-5/3}}
\]  
(2.12)

This is what we call the Kolmogorov -5/3 spectrum (Pope, 2000). \( C \) is the universal constant of Kolmogorov. It is determined experimentally as being equal to 1.5.

Otherwise, there is a simple model spectrum working on all the subranges. The energy-spectrum function can be written:

\[
E(\kappa) = C \frac{2}{\varepsilon^{3} \kappa^{-5/3}} f_L(\kappa L) f_\eta(\kappa \eta)
\]  
(2.13)

Where \( f_L \) and \( f_\eta \), two non dimensional functions
$f_L$ determines the shape of the Energy containing range and tends to unity when $\kappa L$ becomes large. Similarly, $f_\eta$ determines the shape in the dissipation range and tends also to unity when $\kappa L$ becomes small. $f_L$ and $f_\eta$ are unity in the inertial subrange to get the Kolmogorov -5/3 spectrum.

$f_L$ is defined as:

$$f_L(\kappa L) = \left(\frac{\kappa L}{[(\kappa L)^2 + c_L]^2}\right)^{\frac{5}{2} + p_0}$$  \hspace{1cm} (2.14)

Where $p_0$ is equal to 2 and $c_L$, a positive constant.

For small $\kappa L$, $E(\kappa)$ varies as $\kappa^{p_0} = \kappa^2$. If we take $p_0 = 2$, it is known as the von Kármán spectrum with $E(\kappa)$ varying as $\kappa^4$ for small $\kappa$.

$f_\eta$ is defined as:

$$f_\eta(\kappa L) = \exp\left\{-\beta\left\{\left[(\kappa \eta)^4 + c_\eta^4\right]^{\frac{1}{4}} - c_\eta\right\}\right\}$$  \hspace{1cm} (2.15)

Where $\beta$ and $c_\eta$ are positive constants.

In general $\beta$ is equal to 5.2 (Saddoughi and Veeravalli (1994)) and $c_\eta \approx 0.40$.

In Figure 6, we can observe the different subranges described above. The curve has different slopes as we should expect according to the equation. For instance the characteristic slope -5/3 appears in the inertial subrange.

**FIGURE 6 : THE ENERGY SPECTRUM ON THE DIFFERENT SUBRANGES.**
If our measurements are consistent, the energy repartition shape should match with the energy spectrum described on Figure 6.

2.1.3 TURBULENCE MEASUREMENTS FROM FLOW VELOCITY

The turbulence flow can be characterized by using a statistic approach. We can assume that the velocity in the flow is known as function of the position and the time.

A simple indicator is the turbulence intensity, which is the ratio of the root mean square of the velocity over the mean velocity. It quantifies the fluctuation level:

\[ \text{Turbulence intensity: } TU = \frac{U_{\text{rms}}}{U_{\text{mean}}} \]  
(2.26)

The turbulence intensity (TU) can be calculated for one position over the time.

A spectral analysis of the velocity can also be a solution to determine at which frequency the turbulence mainly occurs. The time domain velocity that we measured with a hotwire probe can be transformed to the frequency domain with a Fourier transform. The Fourier transform of the velocity (converted in Third Octave Band) can also be easily compared to the sound power level (SPL).

Before going further, we need to explain one of the most important assumptions we made in the measurement method. This is the Taylor’s hypothesis (Taylor 1938) or the Frozen-turbulence approximation.

The partition of the energy among eddies is determined by measuring the velocity in several points. The spatial autocorrelation between these velocities give us the power spectrum in function of the size of eddies. Doing a spatial autocorrelation implies to measure the velocity in different positions in the same time

Ideally to measure autocorrelation an array of hotwire probes (Figure 7) which give us the velocity in several positions in the same time should be used.

![Figure 7: Linear Hot-Wire Array in Industrie. Illustration from Auspex Corporation](image-url)
But often as in this work, the Taylor’s hypothesis is used to use only one probe in a fixed position. It is quite accurate as long as $\frac{u'}{U_1} \ll 1$. The turbulence intensity is small compared to unity (Pope, 2000).

The temporal autocorrelation is then given by:

$$R_{ij}^m(s) = \langle [U_i^m(t) - \langle U_i^m(t) \rangle][U_j^m(t + s) - \langle U_j^m(t + s) \rangle] \rangle \quad (2.27)$$

where $U_i$, the velocity in the $i^{th}$ direction.

It can also be written:

$$R_{ij}^m(s) = \langle u_i^m(t)u_j^m(t + s) \rangle \quad (2.28)$$

where $u_i$, the turbulent component of the velocity in the $i^{th}$ direction.

The one dimension spectra can be calculated by using the following integral based on Taylors hypothesis:

$$E_{ij}(\kappa) = 1/\pi \int_{-\infty}^{\infty} R_{ij}(Us)e^{-iks}d(Us) = U/\pi \int_{-\infty}^{\infty} R_{ij}(s)e^{-iks}d(s) \quad (2.29)$$

where $U$, the mean flow velocity.

In our measurements, which will be described more in detail later, we measure the radial component crossing the hotwire probe (Figure 8). Indeed a hotwire probe measures the heat exchanges of the hot wire with its surround and we can measure the perpendicular velocity $U_r = U_{1r}$.

For each measurement, the measured velocity is assumed to be in a constant direction called $x_1'$ in Figure 8. Energy spectrum we calculate is:

$$E_{1r1'}(\kappa) = U/\pi \int_{-\infty}^{\infty} R_{1r1'}(s)e^{-iks}d(s) \quad (2.30)$$
As shown in (Pope, 2000) for isotropic turbulence we can relate the one dimension energy spectrum we measure to the general 3D by:

\[ E(k) = \frac{1}{2} k^3 \left( \frac{d}{d\kappa} \left( \frac{1}{k} \frac{dE_{1,1}(\kappa)}{d\kappa} \right) \right) \]  

(2.31)

2.1.4 TURBULENCE GENERATION OF AN EXPANSION

We want to analyze the turbulences generated by an expansion. We need to know exactly where we have turbulence in the duct after the expansion.

Figure 9 explains where we have, after an abrupt expansion, a fully expanded flow (where the flow is exactly the same if no expansion was mounted). The reattachment point is the point where the flow starts to be fully expanded. Before this reattachment point, we have a turbulent eddy. In this section, \( b \) is the length between the expansion and the reattachment point. The reattachment length is approximately \( 6 < \frac{b}{d} < 12 \) for Reynolds number in excess of \( 2 \times 10^3 \) where \( d \) is defined on the figure. This equation is needed if we want to check if the air flow is not fully expanded where we are investigating the turbulence.

![Figure 9: Air Flow in an Expansion Duct (Blevins, 2003).](image)

2.2 HOT-WIRE THEORY

Hot-wire sensors are commonly used to measure velocity fluctuations in a flow from 0 Hz up to high frequencies, with a high spatial resolution and also a low noise level. Hotwire probes are considered as particularly accurate and small.

The hot-wire probe anemometer is constituted of the electrical leads, a probe body and a wire support which are described on Figure 10. The wire, where the heat transfer by convection occurs, has electrical fluctuations (for instance in voltage) and it is directly connected to the velocity fluctuation across the wire, see section 2.2.2.
FIGURE 10: A SCHEMATIC FOR A HOT-WIRE PROBE

The wire is about 1 – 1.2 mm in length and between 5 μm and 70 μm in diameter. They are made generally of nickel or platinum plated tungsten depending on where they are used.

2.2.1 HOTWIRE PROBES

Depending on the flow properties and what we need to measure, different kinds of probes can be used.

The single wire probe is used to measure a single component of velocity, an X-type (cross type) or a split film probe to measure two components velocity and finally a triple sensor probe for three components velocity. Figure 11 is showing different geometries.

FIGURE 11: DIFFERENT KIND OF PROBE (TSI, 2008).

2.2.2 GOVERNING EQUATIONS

In this section, we consider the hot-wire probe as a thin heated wire mounted and exposed to a velocity U.
The heat balance of the wire can be written:

\[ W = Q + \frac{dQ_i}{dt} \]  
(2.32)

where \( W \), the power generated by the current, \( W = I^2 R \), with \( R \) (\( \Omega \)) the wire resistance, \( Q \), the heat power transferred to surrounding, \( Q_i = CT_w \), the thermal energy store in wire, \( C \), the heat capacity of wire, \( T_w \), the wire temperature.

We consider the wire at the equilibrium, which is the case, because a hot-wire probe is always used with a constant temperature. Then the temperature of the wire \( T_w \) is constant in time. It means that \( \frac{dQ_i}{dt} = 0 \).

\( Q_i \) is the heat power transferred by conductivity to surrounding, and can be expressed as:

\[ Q = hA(T_w - T_f) = \frac{A}{d}Nu k_f (T_w - T_f) \]  
(2.33)

Where, \( h \) is the heat transfer coefficient (for convection and conduction), \( A \) the transfer area, \( T_f \) the surround temperature, \( Nu \) is Nusselt number, \( d \) the wire diameter, \( k_f \) is thermal conductivity for the fluid.
In a forced convection regime with $0.02 < Re < 140$ ($Re$, the Reynold’s number), we can write the following formulas:

$$Re = \frac{\rho Ud}{\mu} \quad & \quad Nu = A_1 + B_1 (Re)^n = A_2 + B_2 U^n$$

(2.34)

Where $\rho$, the air density, $\mu$ the air dynamic viscosity, $A_1, A_2, B_1, B_2$ are constants

By substituting (2.56) into (2.54):

$$RI^2 = \frac{A}{d} Nu k_f \left(T_W - T_f\right) = \frac{A}{d} (A_2 + B_2 U^n)k_f \left(T_W - T_f\right)$$

(2.35)

In general, we used a CTA Hot-wire (a Constant Temperature Anemometer Hot-wire). Another possibility is the constant current anemometer.

In the case of the CTA Hot-wire, we can continue by writing:

$$T_w - T_f = Constant$$

(2.36)

And then (2.56) becomes:

$$RI^2 = A_3 + B_3 U^n$$

(2.37)

where $A_3 = \frac{A}{d} k_f A_2 \left(T_w - T_f\right) = Constant$, $B_3 = \frac{A}{d} k_f B_2 \left(T_w - T_f\right) = Constant$

We can finally write:

$$E^2 = R^2 I^2 = A_3 R + B_3 R U^n = A_4 + B_4 U^n$$

(2.38)

where $E$ is the voltage, and $n \geq 0.45$ according to the King’s Law

Then the velocity can be deducted from the Voltage between the leads and can be written as a polynomial relationship as below:

$$U = \alpha_1 + \alpha_2 E + \alpha_3 E^2 + \alpha_4 E^3 + \alpha_5 E^5$$

(2.39)

The coefficients of the last equation can be determined by using the least root mean square method on experimental results obtained from a probe calibration.

### 2.2.3 Calibration

We understand now, how a hotwire probe works. Calibration also needed to be able to measure the velocity with accuracy.

After the hotwire probe, a CTA module is used to maintain the temperature constant and also to amplify the voltage at the output of the system as explained later. An A/D is used to convert the analogic signal to a digital one which is transmitted to a computer for further analysis.

The acquisition chain can be drawn as on the following picture:
FIGURE 14: ACQUISITION CHAIN FOR A HOTWIRE MEASUREMENT SYSTEM BY (DANTEC, 2012).

We used a CTA modulus from Dantec. To set-up the parameters, we used the Streamware software given by Dantec where we need to enter the probe characteristics.

- The resistance of the probe is written \( R_w \).

- The gain and the voltage offset are used to optimize the output voltage from the hotwire system which can be written as:

\[
E_G(t) = G(E(t) - E_{off})
\]

(2.40)

Where \( G \) is the Gain, \( E(t) \), the voltage generated by the fluctuations in temperature, \( E_{off} \), the offset voltage for centering the measurement on zero, and \( E_G \), the output voltage.

- The overheat coefficient is equal to:

\[
\alpha_0 = \frac{T_w - T_0}{T_0}
\]

(2.41)

where \( T_0 \) is the fluid temperature.

In the software parameters, the temperature of the hotwire probe is chosen by the user, depending on the velocity range the user will measure and the flow temperature. The hotwire users usually choose a high hotwire temperature in the software so as to get more accurate results, but they need to take care of the current intensity which could cause damages in the probe. It is always a good idea to choose the parameters delivered by the company which made the probe.

The CTA modulus is used to maintain a constant temperature (for a CTA probe) and to add a gain and an offset voltage.
The CTA modulus is constituted of an amplifier and a Wheastone’s Bridge(Figure 15). The Wheastone’s bridge has two resistances $R_1, R_2$, an accurate adjustable resistance $R_3$, and the resistance of the probe. $R_L$ is the resistance of the cable and the leads and $R_w$ the resistance of the hot wire.

The bridge is equilibrated so that the potentials at the positions A and B are the same when we remove the probe and we connect the two leads (where the probes is connected). In that case $R_w = 0$.

By using Millman’s theorem in the position A and B, we can easily find the following equations:

$$U_A \left( \frac{1}{R_1} + \frac{1}{R_L + R_w} \right) = \frac{U_D}{R_L + R_w} + \frac{U_C}{R_1} & \quad U_B \left( \frac{1}{R_2} + \frac{1}{R_3} \right) = \frac{U_D}{R_3} + \frac{U_C}{R_2}$$

And $U_D = 0, U_C = E_G(t)$ and $R_w = 0$ then:

$$U_A \left( \frac{R_L + R_1}{R_1 R_L} \right) = \frac{E_G}{R_1} & \quad U_B \left( \frac{R_3 + R_2}{R_2 R_3} \right) = \frac{E_G}{R_2}$$

$$U_A = \frac{E_G R_L}{R_1 + R_L} & \quad U_B = \frac{E_G R_3}{R_2 + R_3}$$

We have: $U_A = U_B$. Then:

$$\frac{R_L}{R_1 + R_L} = \frac{R_3}{R_2 + R_3} \text{ or } R_3 = \frac{R_L R_2}{R_1}$$

FIGURE 15: ELECTRICAL SCHEMA OF THE CTA MODULUS
We can determine in that case $R_3$ at the equilibrium point and also $R_L$.

The software does it in the calibration part in order to determine the resistance of the probe cables $R_L$.

The software adjusts $R_3$ in order to have an equilibrated bridge when the temperature of the hotwire is the wanted one. $R_3$ is easily calculated with:

$$R_3 = \frac{(R_L + R_W(T = T_W)R_2}{R_1} \quad (2.46)$$

Where $R_2, R_3$ are known characteristics of the bridge, $R_L$ is determined by the calibration part explained above, $T_W$ is a value chosen by the user, $R_W(T = T_W)$ is calculated by using the formula below.

The hotwire resistance depends of the temperature:

$$R_W(T = T_W) = R_W(T = T_0) + \alpha (T_W - T_0)R_W(T = T_0) \quad (2.47)$$

Where $\alpha$ is the Temperature Coefficient of Electrical Resistivity Per °C, according to (http://www.engineeringtoolbox.com/resistivity-conductivity-d_418.html)

$$a_{tungsten} = 3.93 \times 10^{-3}K^{-1}$$

$$a_{platinum} = 3.93 \times 10^{-3}K^{-1}$$

When the hotwire probe is exposed to a flow, the resistance $R_W$ fluctuates depending on the velocity. The bridge is not equilibrated anymore, and an error “$e_2 - e_1$” appears and the amplifier delivers a current intensity $I$ to increase the temperature of the hotwire so that this error decreases.

The temperature is then controlled by setting the temperature we want to have in the hotwire probe and by using the formula of equilibrium $R_3 = \frac{(R_L + R_W(T = T_W)R_2}{R_1}$. The temperature is almost constant. Only the adjustable resistance needs to be set up and also the gain and the offset voltage.

2.3 SOME ACOUSTICS

In this section we introduce the concept of multipoles to describe the noise generation at the duct outlets.

The starting point is the classical wave equation with a source term $s(x, t)$, (Åbom, 2006):

$$\frac{1}{c_0^2} \frac{\partial^2 p'}{\partial t^2} - \nabla^2 p' = s(x, t) \quad (2.48)$$

2.3.1 SOLUTION OF THE INHOMOGENEOUS EQUATION
The solution can be found by studying the point source solution. The solution is called a Green’s function \( G \), defined by:

\[
\frac{1}{c_0^2} \frac{\partial^2 G(x,y,t,\tau)}{\partial t^2} - \nabla^2 G(x,y,t,\tau) = s(x,t,\tau) = \delta(t-\tau)\delta(x-y)
\]

The global solution can be obtained by doing an integral of the last equation.

The problem is then to find the source term related to how the sound is produced by unsteady flows. This is the Lighthill’s theory.

In the Lighthill’s theory, the space is split into two spaces: the sound field and the source field. The first space satisfies the homogeneous equation and the second one the inhomogeneous equation.

Lighthill rewrote the fluid mechanics equation for obtaining an equation in the form of a wave equation governing the source field space. The starting point is then the classical fluid mechanics equations of conservation of mass and momentum:

\[
\frac{\partial\rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = m
\]

\[
\frac{\partial\rho u_i}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = \frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + f_{vi}
\]

By doing the following manipulation:

\[
\frac{\partial (2.71)}{\partial t} - \frac{\partial (2.72)}{\partial x_i}
\]

We obtain:

\[
\frac{\partial^2\rho}{\partial t^2} - \frac{\partial^2\rho}{\partial x_i \partial x_i} = \frac{\partial m}{\partial t} - \frac{\partial f_{vi}}{\partial t} + \frac{\partial^2 (\rho u_i u_j - \tau_{ij})}{\partial x_i \partial x_j}
\]

If we assume that we are in an homogeneous fluid with no mean flow, characterized by \( p_0, \rho_0 \) and \( c_0 \), we can write:

\[
p = p_0 + p' \quad \rho = \rho_0 + \rho'
\]

We can finally find the wave equation in function of \( p' \) or \( \rho' \):

\[
\frac{1}{c_0^2} \frac{\partial^2 p'}{\partial t^2} - \frac{\partial^2 p'}{\partial x_i \partial x_i} = \frac{\partial}{\partial t} \left( m + \frac{1}{c_0^2} \frac{\partial}{\partial t} \left( p' - c_0^2 \rho' \right) \right) - \frac{\partial}{\partial x_i} \left( f_{vi} \right)
\]

\[
+ \frac{\partial^2}{\partial x_i \partial x_j} \left( \rho u_i u_j - \tau_{ij} \right)
\]

According to the Lighthill’s theory, we assume that the acoustics field does not interact with the source field. The source term in Lighthill’s acoustics analogy can be split as following:
\[ s_1 = \frac{\partial}{\partial t} \left( m + \frac{1}{c_0^2} \frac{\partial}{\partial t}(p' - c_0^2 \rho') \right) \]  
\[ s_2 = -\frac{\partial}{\partial x_i} (f_{v,i}) \]  
\[ s_3 = \frac{\partial^2}{\partial x_i \partial x_j} (\rho u_i u_j - \tau_{ij}) \]  

(2.55)  
(2.56)  
(2.57)

As proved in the book “An introduction to flow Acoustics” (Åbom, 2006), the general solution for multi poles source can be written:

\[ p_n(x,t) = \iiint_V \frac{\partial^n \left( s \left( y, t - \frac{r}{c_0} \right) \right)}{4\pi r} dV_y \]  

(2.58)

n is the multipole order. n=0 corresponds to a monopole, n = 1 corresponds to a dipole and n = 2 to a quadrupole.

If the source is small enough compared to the wavelength, the source distribution can be reduced to a single point.

2.3.2 THE DIFFERENT KIND MULTI-POLES

The monopole source corresponds to \( s_1 \) and can be used for the mass injection or volume flow source as loudspeakers with box or the outlet or inlet of piston machine. The resulting sound field is the superposition of monopoles:

\[ p'(x, t) = \iiint_V \frac{\dot{m}_t \left( y, t - \frac{r}{c_0} \right)}{4\pi r} dV_y \]  

(2.59)

Where \( \dot{m}_t = \frac{\partial m_t}{\partial t} \) and \( m_t = m + \left( \frac{1}{c_0^2} \right) \frac{\partial (p' - c_0^2 \rho')}{{\partial t}} \)

We can also write: \( m' = \rho_0 q' \) where \( q' \) is the unsteady volume flow per \( m^3 \). Then we obtain:

\[ p'(x, t) = \frac{\rho_0 Q \left( t - \frac{x}{c_0} \right)}{4\pi x} \]  

(2.60)

Where \( Q(t) = \iiint_V q'(y, t) dV_y \)

The sound power radiated from a monopole source, when it is stationary or periodic can be written as:
\[
\overline{W_m} = \iiint_{\text{source}} p'u_r' \, dS = \lim_{r \to \infty} \iiint_{\text{source}} \frac{p'}{\rho_0 c_0} \, dS = \frac{\rho_0 \overline{Q^2}}{4\pi c_0}
\]

where \(u_r'\) is the radial flow velocity.

The dipole source corresponds to \(s_2\) and can be used to model loudspeaker without box or time varying or unsteady forces on solid bodies subject to a mean flow. If we write \(F_i(t) = \iiint V f_{v,i}(y, t) \, dV_y\), the total force acting on the fluid, then we can find the sound field generated by a compact dipole:

\[
p'(x, t) = -\iiint \frac{\partial}{\partial x_i} \left( \frac{f_{v,i} \left( y, t - \frac{r}{c_0} \right)}{4\pi r} \right) \, dV_y
\]

\[
= \{\text{compact assumption}\} = -\frac{\partial}{\partial x_i} \left( \frac{F_i \left( t - \frac{x}{c_0} \right)}{4\pi x} \right)
\]

\[
= F \left( t - \frac{x}{c_0} \right) \cdot \frac{e_x}{4\pi xc_0}
\]

Where the bold characters are vectors and \(e_x = x/x\).

The sound power radiated for a dipole can be written:

\[
\overline{W_d} = \frac{F^2}{12\pi \rho_0 c_0^3}
\]

where \(F = |F|\).

Finally, the quadrupole source corresponds to \(s_3\) and is used when sound is generated due to a momentum transport in a flow. It is the main source in a high speed jet.

If we define the Lighthill’s tensor as: \(T_{ij} = \rho u_i u_j - \tau_{ij}\), we can express the sound field for a compact quadrupole, as following:

\[
p'(x, t) = \iiint \frac{\partial^2}{\partial x_i \partial x_j} \left( \frac{T_{ij} \left( y, t - \frac{r}{c_0} \right)}{4\pi r} \right) \, dV_y
\]

\[
= \{\text{compact assumption}\} = \frac{\partial^2}{\partial x_i \partial x_j} \left( \frac{Q_{ij} \left( t - \frac{x}{c_0} \right)}{4\pi x} \right)
\]

where \(Q_{ij} = \iiint V T_{ij} (y, t) \, dV_y\), is the source strength.

If we assume to be far enough from the source, the far field component is given as:

\[
\overline{W_d} = \frac{F^2}{12\pi \rho_0 c_0^3}
\]
\[
p'(x, t) = \frac{\tilde{Q}_{ij} \left(t - \frac{x}{c_0}\right) e_{x,i} e_{x,j}}{4\pi x c_0^2}
\]  
\( (2.64) \)

where \( e_{x,j} = \frac{x_j}{x} \)

The sound power radiated can be written as:

\[
\bar{W}_q = \frac{\varepsilon_{ij} Q_{ij}}{\rho_0 c_0^2}
\]

where \( \varepsilon_{ij} = \begin{cases} 
\frac{1}{20} \pi & \text{when } i = j \\
\frac{1}{60} \pi & \text{when } i \neq j 
\end{cases} \)  

\( (2.65) \)

\[2.3.3 \text{ The sound generation from the nozzles} \]

Nozzles or holes can be drilled in ducts, in order to let air out for, e.g., ventilation purposes. It is interesting to try to figure out which kind of model for the sound generation we could use.

We will assume that the holes have a diameter \( D \). The velocity through the holes are noted as \( U_h \). \( f \) is the frequency of the excitation of the source (the nozzles) and \( T \) can be seen as the inverse of \( f \). \( \rho_0 \) the mass density. We can scale the sound power generation of the different models seen previously and easily write the following relations:

\[
\begin{align*}
Q &\propto U_h d^2 \\
F &\propto \rho_0 U_h^2 d^2 \\
\rho v_i u_j &\propto \rho_0 U_h^2 \\
 f &\propto U_h/d
\end{align*}
\]

\( (2.66) \)

Then we can express the sound power radiation for the different kinds of multi-poles:

\[
\begin{align*}
\bar{W}_m &\propto \rho_0 U_h^2 d^4 c_0 T^2 \\
\bar{W}_d &\propto \frac{\rho_0 U_h^6 d^2}{c_0^3} \\
\bar{W}_q &\propto \frac{\rho_0 U_h^8 d^2}{c_0^5}
\end{align*}
\]

\( (2.67, 2.68, 2.69) \)

The Mach number is defined as:

\[
M = \frac{U_h}{c_0}
\]

\( (2.70) \)

Then we can conclude by writing:

\[
\begin{align*}
\frac{\bar{W}_m}{\bar{W}_d} &\propto \frac{\bar{W}_d}{\bar{W}_q} \propto \frac{1}{M^2}
\end{align*}
\]

\( (2.71) \)
For our application, we have low speed ventilation ducts (<30m/s) and Mach-numbers less than 0.1. Also for sound from small holes the unsteady jet will produce dipoles and quadrupoles. But in the small Mach number range the dipole will dominate and the quadrupole can be neglected. The monopole can be shown to be of equal strength with the dipole for a jet.

3 THE MEASUREMENT METHOD

In this chapter we introduce the method we used in our work. It gives an overview of the hotwire system, the used sound measurement method, the test setup and the calibration setup.

3.1 HOTWIRE SYSTEM

3.1.1 HOTWIRE PROBES

During the measurement sessions we used different 1D homemade probes and also Dantec probes, specially the 1D Dantec probe 55R05, which is a Boundary layer type with 0.5 μm coating with the following specifications delivered by Dantec in (Dantec, 2012):

![55R05 probe illustration](image)

**FIGURE 16: THE USED PROBES ILLUSTRATED BY (DANTEC, 2012).**

This probe is a Fiber-film probe. Nickel film deposited on 70 μm diameter quartz fiber. The wire length is 3 mm with a sensitive film length of 1.25 mm. Copper and gold plated at the ends and the film is protected with a quartz coating of approximately 0.5μm in thickness. The total diameter is then equal to 71 μm. These probes have a thicker wire compared to the traditional one with Pt-plated tungsten wire or Pt-plated tungsten with a copper and gold plated at the ends of the wire.

The following table (Dantec, 2012) gives us the different characteristics of the used probe (in red) and other possible solutions (in blue). In all the cases, the probes can detect very small velocity and be used at high frequencies (until 90-150 kHz).

<table>
<thead>
<tr>
<th>SENSOR TYPE</th>
<th>Sensor material</th>
<th>Sensor dimensions</th>
<th>Thickness of quartz coating</th>
<th>Sensor resistance R unst (approx.)</th>
<th>Temperature coefficient of resistance (TCR unst) Approx</th>
<th>Max. sensor temperature</th>
<th>Max. ambient temperature</th>
<th>Max. ambient pressure</th>
<th>Min. velocity</th>
<th>Max. velocity</th>
<th>Frequency limit fmax (63% response)</th>
<th>Medium</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gold-plated wire sensors</td>
<td>Platinum-plated tungsten</td>
<td>5μm dia. 1.25mm long*</td>
<td>-</td>
<td>3.5 Ω</td>
<td>0.36% /K</td>
<td>300°C</td>
<td>150°C</td>
<td>-**</td>
<td>0.20m /s</td>
<td>200m /s</td>
<td>90kHz ****</td>
<td>Air</td>
</tr>
</tbody>
</table>
### Miniature wire sensors
- **Platinum-plated tungsten**
  - 5μm dia.
  - 1.25mm long
  - 3.5 Ω
  - 0.36% /K
  - 300°C
  - 150°C
  - 0.20m/s
  - 500m/s
  - 150kHz

### Fiber-film sensors
- **Nickel**
  - 70μm dia.
  - 1.25mm long
  - 0.5μm dia.
  - 6 Ω
  - 0.40% /K
  - 300°C
  - 100°C
  - 0.20m/s
  - 350m/s
  - 90kHz

- **Welt**
  - 2μm dia.
  - 6 Ω
  - 0.40% /K
  - 60°C
  - 100°C
  - 0.01m/s
  - 10m/s
  - 3kHz

*Overall tungsten wire length is 3 mm. Wire ends are gold-plated to 25-30 μm dia., limiting sensor length.

** Depending on type of mounting.

***Overall fiber length is 3 mm. Fiber ends are gold-plated, limiting sensor length.

****At 100 m/s.

The probe is connected to a 4 mm dia. probe supports for single-sensor probes of reference 55H20:

![FIGURE 17: ILLUSTRATION OF A 4 MM DIA. PROBE SUPPORT, (DANTEC, 2012).](image)

The support is then connected to another wire to the CTA anemometer system.

#### 3.1.2- The CTA anemometer

The CTA anemometer is a Dantec anemometer of reference number 90N10 connected to the acquisition computer with a serial connection in order to set up the parameters of the Wheatstone bridge. The Streamline anemometer 90N10 could also be connected to a temperature probe. In our measurements, we do not use the temperature probe and the air flow is assumed to be at the fixed temperature (21 °C) of the pumped air. The Streamline anemometer works with added modules inside the anemometer of reference 90C10 which are connected to the hotwire probe connection and this module is constituted of the Wheatstone bridge. The streamline anemometer is the interface between the module 90C10 and the computer. Moreover, the software of Dantec for configuring the anemometer is set-up depending of the kind of probe used, we can choose the library or to create a new probe especially if it is a homemade probe. The characteristics defined are the resistance of the probe, the overheat ratio, the heat coefficient, the gain, the offset voltage, the wire diameter and the length of the wire.

The anemometer has also an output which is directly plugged to the acquisition card and then to the computer, for acquiring the signal.

#### 3.1.3- The national instrument acquisition card
The output signal from the anemometer is a voltage. The hardware part of the acquisition is done by using an A/D national instrument card NI 9215 with 4 anagoric inputs +/- 10 V and with BNC plugged to NI CompactDAQ chassi NI USB-9162.

The national instrument (NI) card has 4 simultaneously sampled analog inputs, 100kS/s and 16-bit resolution. The frequency acquisition has to be at least two times higher than the highest frequency in the input signal (Nyquist theorem). The card frequency is corrected according to the frequency probe of utilization.

The signal acquisition and the conversion from the voltage to the velocity are done with our own code using the measurement software Labview.

3.2 Sound Measurement Method

The reverberation room at FläktWoods, Jönköping, has the following characteristics:

<table>
<thead>
<tr>
<th>Volume:</th>
<th>8 x 5.6 x 4.7 = 210 m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight:</td>
<td>240 tons</td>
</tr>
<tr>
<td>Absorption:</td>
<td>~4 m²S</td>
</tr>
<tr>
<td>Background level:</td>
<td>20 dB [A]</td>
</tr>
<tr>
<td>Airflow:</td>
<td>0 – 2000L/s</td>
</tr>
</tbody>
</table>

This section was written according to (Feng, 2011) and (ISO3741). More specification is given in Annex 1.

The method for measuring the sound power level, ISO 3741, is considered as a precision method which is used in a reverberation room (with or without a reference source).

The first step consists of measuring the averaged Sound Pressure Level, in the reverberation room, for the reference source $L_{p(RSS)}$ with several microphones. This reference source is delivered by a certificated company specialized in calibration tools and give us the accurate value corresponding of the sound power level for the reference source $L_{W(RSS)}$. After this measurement, the reference source can be removed.

The second step is the measurement of the source under test (ST) (this source should be at the same position than the reference source of the first step). The averaged Sound Pressure Level is measured in the reverberation room for the source under test ($L_{p(ST)}$).

Then we have $L_{p(ST)}$, $L_{p(RSS)}$, and $L_{W(RSS)}$. Sound power levels in third-octave bands for the source under test ($L_{W(ST)}$) can be found by using the following relation:

$$L_{W(ST)} = L_{W(RSS)} + L_{p(ST)} - L_{p(RSS)}$$

where $L_W$ and $L_p$ are respectively the sound power level and the sound pressure level.

Some precision of the location and the number of the microphones are needed like the characteristics of the test room.

The absorption of the test room affects the minimum distance between the noise source and the microphones. Another effect, it influences the sound radiation and the frequency response
characteristics. That is why a reverberation room used for measuring the sound power level needs to be neither too large nor extremely small. Moreover, the source has to be placed at least 1.5 m from any wall of the room. Other requirements are needed concerning the microphone positions. The minimum distance between source and microphones $d_{\text{min}}(m)$

$$d_{\text{min}} = C \left( \frac{L_{\text{WR}} - L_{\text{pr}}}{20} \right)$$ (3.2)

where $C = 0.4$ and preferable $= 0.8$. $L_{\text{WR}}$ and $L_{\text{pr}}$ are the sound power and the sound pressure level of the reverberation sound source.

For a standard reverberation room (200 m$^3$), the minimum distance is about 0.5 m. The distance between the microphones has to be at least $\lambda/2$ from each other and 1 m from the room surface.

Furthermore, we use a reverberation room in order to have a diffuse field but it is not a perfect diffuse field and that is why we average over many microphone positions to get the average sound pressure level. When fixed microphones positions are used, at least 6 microphones position at different heights are needed. For a precise standard deviation, more microphones could be needed as described in the ISO standard.

Finally, the sound power level can be calculated from these averages:

$$L_{w} = (L_{p}) - 10 \log \left( \frac{A}{A_0} \right) - 6 + \frac{4.34A}{S} + 10 \log \left( 1 + \frac{S_c}{BVf} \right)$$

$$- 25 \log \left( \frac{427}{400} \sqrt{\frac{273}{273 + \theta}} \frac{B}{B_0} \right)$$ (3.3)

where $\frac{4.34A}{S}$, the correction of air absorption, $10 \log \left( 1 + \frac{S_c}{BVf} \right)$ is the term for wave interference, and $25 \log \left( \frac{427}{400} \sqrt{\frac{273}{273 + \theta}} \frac{B}{B_0} \right)$ is the term for correcting the ambient pressure and the temperature.

3.3 Test setup

The prototype is done in order to study the influence of an expansion on the sound generation of nozzles here small holes, especially due to the turbulence created by the expansion.

The following picture illustrates the different turbulence generation cases involved in the test set up:
All these cases have to be investigated. From here, two main cases appear: with and without an expansion. That is why two prototypes are made. The length between the holes and the expansion is also important and has to be neither too big nor small. We choose to use a length $L$ of 10cm and 20 cm.

The first prototype considers a straight duct with a flow coming through the duct at different velocities $V$ (0m/s included). After a silencer (to diminish the noise) and several meters of straight duct (to get a fully developed flow profile), needed for getting the lower turbulence as possible, ten nozzles are made in a duct portion around the circumference. The pressure in the duct has to be easily modified for investigate the influence of the pressure. A damper is used at the end of the duct for changing the pressure in the duct. The used duct has a diameter of 320mm. Figure 17, where the dimensions are according to scales, illustrates the first case in the reverberation room.
Some pictures are also added in Annex 2, where we can see the straight duct in the reverberation room.

FIGURE 19: HATCH AND THE NOZZLES IN THE NORMAL CASE.

10 Nozzles around the duct

FIGURE 20: THE OUTLET OF THE PROTOTYPES.

The second prototype under test is globally the same than the first one except of the increase in section before the nozzles. The small duct has a diameter equal to 125 mm and the large duct is also equal to 315mm. The length L between the nozzles and the increase is evaluated according to the theoretical part and we choose L=20cm and 10cm. The simulated overview of the second prototype is below.

FIGURE 21: REVERBERATION ROOM WITH THE PROTOTYPE WITH THE INCREASE IN SECTION.
In both prototypes, we can modify different parameters. The different possible configurations are summed up in the following table:

<table>
<thead>
<tr>
<th>Prototype</th>
<th>Straight duct</th>
<th>Expanded duct</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air flow velocity V(m/s)</td>
<td>0/2/3/4</td>
<td>0/2/3/4</td>
</tr>
<tr>
<td>Holes diameter D(cm)</td>
<td>0.5 or 1</td>
<td>0.5 or 1</td>
</tr>
<tr>
<td>Length between the nozzles and the expansion L(cm)</td>
<td>10 or 20</td>
<td>10 or 20</td>
</tr>
<tr>
<td>Static pressure Ps(Pa)</td>
<td>100/200/300</td>
<td>100/200/300</td>
</tr>
</tbody>
</table>

The probe position is described in chapter 4, in the introduction by the variable Y.

### 3.4 Calibration and Measurement Setup

For positioning the probe inside the duct, a probe support can be built for this special use. The manufactured probe support is described below (Figure 23).
The probe support is positioned inside the duct. The support is stable in the duct and does not disturb the upstream flow. According to the arrows (Figure 23), we can easily position the probe where it is needed with accuracy.

The hatch guarantees a good access to the probe support for positioning as seen below without disturbing the upstream flow under test.

![Image](image-url)

**FIGURE 24: THE PROTOTYPE WITH THE AREA INCREASE AND THE PROBE SUPPORT.**

During the measurements session, the sound measurement was also done without the probes support. The probe support does not disturb the upstream flow for the turbulence measurements but for a high speed air flow velocity, the probe support generates an extra noise which disturbs the sound measurement. That is the reason we did the sound measurements without probes. We have compared the sound measurements with and without the probes support and the influence of the probes support can be neglected for the duct expansion case.

4 **Turbulence and Sound Analysis**

The temperature of the reverberation room is approximately equal to 21 °C and the pressure is equal to the atmospheric pressure (10^5 Pa).

Two main cases were introduced previously, and two main kinds of measurements are done: with and without an expansion before the nozzles for different probe position, different grazing flow velocities, and different static pressures.

In the analysis section, different explanations need to be given. We use in general to describe the position of the probe the coordinates x and y. If we use only y it means that x=0. Otherwise the coordinate system is defined as following, in the P-Plane:
The different variables, used during the measurement, are introduced on Figure 26.

\[ q(\text{l/s}) \text{, represents the air flow rate going through the duct (measured with a calibrated flange). } \]

\[ V \text{ is then calculated as being the mean flow velocity going through the large duct } (V = \frac{q}{S}, S \text{ the area of a duct section}). \]

\[ \text{We have to take care to not mistake } V \text{ for the mean flow measured with the probe.} \]

\[ \text{The coordinate system used is described on Figure 25, and } Y \text{ corresponds in the following explanations to the length between the probe and the holes. If the probe is not centered to the hole we use the variable } x \text{ to describe the shift in the X-direction.} \]

In the analysis, we compared measurements which were not made always in the same time and with the same probes. A lot of uncertainty appears. The fact that we use different probes should not influence our measurements so much. The calibration is made carefully. The time of response of the probe is very low and the fluctuations seen by different probes have to be the same. Only the calibrations change. The only approximation is that the measured velocity has an error of less of 5 %. Otherwise, the fluctuations, acquired by different probes with different calibrations, are then comparable. It was checked for measurements with the same setup.

In this chapter, we develop the analysis for the straight duct from 4.1 to 4.3. In 4.4, we introduce for the first time, the measurements for the expanded duct without air flow velocity in the duct. In 4.5, we continue with the turbulence analysis with an expanded duct with an air flow velocity.
through the duct and in 4.6, we do the sound analysis of the expanded duct with a grazing flow. Finally, in 4.7, we analyze the influence of the expansion on both the sound and the turbulence. And 4.8 can be seen as an extra analysis on the influence of the length $L$ between the holes and the expansion.

### 4.1 A STRAIGHT DUCT WITHOUT GRAZING FLOW ($V=0\text{m/s}$)

For this section, the damper (see Figure 20) is totally closed and no grazing flow is going through the duct ($V = 0\text{m/s}$).

**Concerning the turbulence analysis**, in Figure 27, we can observe the Power Spectrum of the flow velocity (PSv) without the grazing flow ($V = 0\text{m/s}$) for different static pressures, positions of the probe ($y = 5 \text{ or } 23 \text{mm}$) and holes diameters ($D = 0.5 \text{ or } 1\text{cm}$). The PSv curves show that no high amplitude of power can be detected. The peaks that we observe between 100 Hz and over 1 kHz are included in the background noise (the background or the white noise is obtained when the probe is measuring in a closed space without pressure and velocity. The probe is unexcited). It means that the anemometer is the cause of the peaks that we have on Figure 27, between 100 Hz and 1000 Hz. Nevertheless, we can see different kinds of amplitudes. For instance, we have more turbulence for 300 Pa ($D = 1\text{cm and } y = 5\text{mm}$, red dash line curve) than for 100Pa (blue dash line curve). The reason is that we have a higher velocity through the nozzles and then the velocity fluctuations are amplified as well.

![PSV for a straight duct without air flow](image)

**FIGURE 27**: PSV for a straight duct with no air flow, different static pressures (PS) and measured at different distances from the hole ($y$) with different holes diameters ($D$).

In Figure 28, we select different case from Figure 27 in order to be able to make conclusion.
An explanation for the small amplitudes of the turbulence level can be formulated. The static pressure creates an ordered air flow (closed to the nozzles) and it reduces the natural convection. The PSv graphics (Figure 27 and Figure 28), which quantified the turbulence, are only high at low frequency and decreases very quickly with the frequency. In Figure 29, we can see that the Turbulence intensities (defined by equation (2.26)), for different static pressures are rather low.

Moreover when the probe is further from the nozzles, \( Y \) is higher and the turbulence level is stronger (by comparing Figure 29 with \( y = 5 \text{ mm} \) to Figure 30 with \( y = 23 \text{ mm} \)). The relative order
of the air flow is lower (and TU is higher) when Y is higher and the order increases when the air flow is converging to the nozzles.

![Graph of TU (%) and Flow velocity (m/s) vs. Static pressure (Ps)](image)

**FIGURE 30**: MEASUREMENT IN A DUCT WITHOUT EXPANSION WITH NO GRAZING FLOW FOR DIFFERENT DIAMETERS (D) AND WITH Y=23MM.

Concerning the sound analysis, Figure 31 gives the sound measurement we obtained without grazing flow for the straight duct case and for different holes diameters and static pressures. It seems that, for low pressure, the background noise is too high (green curve vs. blue curve).

![Graph of SPL vs. Frequency (Hz)](image)

**FIGURE 31**: SPL FOR A STRAIGHT DUCT WITH NO GRAZING FLOW, DIFFERENT STATIC PRESSURES (Ps) AND DIFFERENT HOLES DIAMETERS D.
We observe that the higher is the pressure, the higher the flow velocity through the nozzles is and the stronger the sound generation is (the purple curve compared to the red one, Figure 31). We can also say that the sound generation is always reduced for the small holes.

Furthermore, we can explain the reason of the increase of the sound with an increase of the pressure. The velocity $V_h$ of the air flow ejected from the nozzles is proportional to the sound generated. Equation (2.89) (see. dipole theory in 2.3.3) gives the sound power generated by dipoles:

$$\overline{W_d} \propto \frac{\rho_0 V_h^6 D^2}{c_0^3}$$

The fact that the air flow velocity seen by the probe (called $V_p$ in Figure 29 and Figure 30) increases with the static pressure means that, $V_h$ (equal to $V_p$, without grazing) is also increasing and it leads that the sound power level is increased when the statics pressure is higher (Figure 31).

Moreover the sound generated by dipoles is also proportional to $D^2$ and the higher the holes diameter $D$ is, the higher the SPL will be (see Figure 31). According to (2.89), if we double $D$, $\overline{W_d}$ is multiplied by 4, and it corresponds to an increase of 6 dB for the sound power level. According to the measurements made, we can calculate the total sound power level $L_{wtot}$ for each measurement.

<table>
<thead>
<tr>
<th>$P_s$</th>
<th>$L_{wtot1}$ [dB] for D=0.5 cm</th>
<th>$L_{wtot2}$ [dB] for D=1 cm</th>
<th>$L_{wtot2} - L_{wtot1}$ [dB]</th>
</tr>
</thead>
<tbody>
<tr>
<td>100Pa</td>
<td>33.8</td>
<td>38.6</td>
<td>4.8</td>
</tr>
<tr>
<td>200Pa</td>
<td>35.1</td>
<td>41.5</td>
<td>6.4</td>
</tr>
<tr>
<td>300Pa</td>
<td>38.6</td>
<td>45.8</td>
<td>7.2</td>
</tr>
</tbody>
</table>

**Figure 32: Difference of the total sound power level for different pressures in a straight duct and no grazing flow.**

For 100 Pa, we have only 4.8 dB and we know that the background level is too high compared to the sound measurement (Figure 31). It is certainly a reason of the low increase. For 200 Pa, the measurements suit with the theory, and for 300 Pa, we have an increase of 7.2 dB instead of 6 dB. The source power $\overline{W_d}$ is multiplied by 5.2 instead of 4. An uncertainty comes from the static pressure (which is approximately fixed at 300 Pa) and the holes geometry (all the holes have defaults).

To conclude, the turbulence measurements give only low power amplitude but we can observe differences on the turbulence level in function of the case. For the sound measurement, we found that the background level is relatively high compared to the sound measurements at 100 Pa. For 200 Pa and 300 Pa, the background noise is low enough. We have also seen that the sound power level depends of the velocity going through the holes and also of the holes diameters. We will continue our investigation with a positive velocity in a straight duct in the next section.

### 4.2 Turbulence Analysis with a Grazing Flow ($v>0$ M/s) in the Straight Duct

For this section and the next one, the measurements were made for a specific static pressure ($P_s$), a certain grazing flow velocity $V$ and a known diameter $D$. The damper is not totally closed but partially, in order to reach the needed static pressure ($P_s$) and the mean flow velocity through the duct ($V$).
We start in this section with the turbulence analysis in a straight duct with a grazing flow. Figure 33 gives the turbulence intensity for different static pressures for $V = 2m/s$ and $0m/s$ in order to compare the influence of the grazing flow on the turbulence measurements. The power spectrum of the velocity is much higher for frequencies between 10 Hz and 1 kHz when we have a grazing flow ($V > 0$), as Figure 33 shows.

The grazing flow disorders the order created by the static pressure in the closed surrounding of the nozzles and it explains what we see in Figure 33. Then two phenomena are involved. First one, the increase of the static pressure involves a decrease of the turbulence level (red, green and light blue curves in Figure 33). On the other hand, the grazing flow globally increases the turbulence level for frequencies until 1 kHz. Our guess is we have a lot of turbulences with the grazing flow due to two different flows, one going through the duct and the other one trying to go outside by the nozzles due the static pressure. These two different flows make eddies.

Moreover, the global pattern of the turbulence curves obtained for a straight duct and with a grazing flow can be predicted by the theoretical part. For the following explanation, we can plot a representative case (for instance, the PSv curve for $v = 4m/s$ and 100 Pa measured at 5mm from the hole) with a modified scale. We choose in abscise $\log(f)$ which is equal to $\log(\kappa) + constant$ described in the theoretical part 2.1, more specifically in section “2.1.2 The energy spectrum”, where the energy-spectrum equation (2.12) is defined in function of $\kappa$. 

**FIGURE 33 : PSV FOR A STRAIGHT DUCT WITH AND WITHOUT A GRAZING FLOW, WITH DIFFERENT PS, Y=5MM AND D=1CM**
The Kolmogorov -5/3 spectrum, described by equation (2.13) is found. We can identify the different subranges from Figure 6: the **inertial subrange** (where the red curve has a -5/3 slope) and the **dissipation subrange** with a higher negative slope (in blue in Figure 34). Nevertheless the **energy containing range** is not really viewable in the figure and the graphics cannot be plotted for smaller frequencies. But the curve starts also to be round as it appears in Figure 6.

Furthermore, for all the measurements done by the probe in a straight duct and with a grazing flow, the curves are gathered in different groups, one for each grazing flow velocity as we observe in Figure 35, where we measured the turbulence level for different static pressures and velocities measured at 5mm from the hole. We add in Annex 5 (Figure 86 and Figure 87) the same measurement for \( D = 0.5 \text{cm}. \)
The different parameters involved and analyzed in this paragraph are the position of the probes ($y$), the holes diameters ($D$), the flow velocity ($V$) and the static pressure in the duct ($P_s$).

In Figure 35, PSv’s curves are plotted with different grazing flow velocities (2m/s, 3m/s and 4 m/s) and different static pressures (100 Pa, 200 Pa and 300 Pa) for $y = 5mm$ and $D = 1cm$. The probe is then close to the hole and three groups of curves, one for each velocity, appear. It means that the grazing flow velocity through the duct is the main variable of influence for the turbulence level. Furthermore, for each velocity, we notice that the PSv is in general stronger for low static pressure. The idea, that the static pressure forces the air to go through the nozzles and consequently reduces the turbulence, seems a good explanation at least in the close surrounding of the nozzles.

When the probe is further from the holes ($y = 23mm$ in Figure 36), we can observe that the three groups of curves, one for each velocity, is still viewable, but the curves are more close each other’s (Figure 36). The influence of static pressure seems to be negligible (compared to the grazing flow velocity). It means, in other terms, that the relative order generated by the influence of pressure is negligible compared to the turbulence generated by the geometry and the grazing flow. This conclusion is valid when the probe is far enough from the hole.

![Figure 36: PSV for a straight duct with different $V$, static pressure ($P_s$) and measured at 23 mm from a hole with $D=1CM$.](image)

Finally, Figure 37 illustrates the fact that the turbulence level increases with the distance from the hole. That is why there is a shift between the curves plotted for $y = 5$ and 23 mm. On the other hand, as we explained in the last paragraph, the turbulence curves are very close to each other for a same velocity. We prefer for this reason to be closer to the hole ($y = 5mm$) to be able to observe the influence of the static pressure on the turbulence measurements and we assume that it is always better to be closer of the sound sources (in our case, the holes). The measurement done for $y = 5mm$ will be used in priority for this reason.
4.3 Sound analysis with a grazing flow (\(\nu>0\) m/s) in the straight duct

In this section, we do the sound analysis for the straight duct with a grazing flow. We will use what is already said about the turbulence level in the last section.

We start the sound analysis with the comparison with and without a grazing flow in the straight duct. Figure 38 gives the sound power level in third octave bands for two different static pressures (100 and 300 Pa) and with \(\nu = 0\) m/s and 2 m/s. The holes diameter is 1 cm. Extra measurements for \(D = 0.5\) cm is given in Annex 5.
We notice that over 1 kHz, the grazing flow seems to have a positive effect and less sound is generated. Below 600-800 Hz the SPL increases when we have a grazing flow (Figure 38).

These two subranges, below 800 Hz and over 800 Hz, can be explained if we assume that two phenomena occur.

The first phenomenon is related to the turbulence level. The grazing flow has an influence on the sound source generation. When we increase the grazing flow, there is more turbulence around the nozzles and it leads to a higher PSv until 1 kHz (section 4.2). We can assume that the increase of the SPL until 600-800Hz (Figure 38) can be explained by the higher turbulence level with grazing flow.

The second phenomenon is related to the grazing flow and the static pressure involving an air flow rate going through the holes. Our guess is that the grazing flow breaks the air flow going through the holes. This is a strong assumption and extra measurement could be helpful to show if we have a lower air flow velocity going through the nozzles \( V_h \) when we increase the grazing flow. We could measure, for instance, \( V_h \) by positioning the probe outside the duct, very close to the hole. If we verified this assumption, the dipole equation (2.89) show us that when \( V_h \) decreases, the sound generated decreases as well. It means that when we have a grazing flow, the decreasing of \( V_h \) (if it is verified) explains the lower sound level over 1 kHz with a grazing flow we observed in Figure 38.

The first conclusion is that the grazing flow has different effects on the sound. Before 800 kHz, the grazing flow increases the sound level (and we found the same effect on the turbulence level). Over 1 kHz, the grazing flow makes decrease the sound level and extra measurements are needed to fully explain this observation (the turbulence level seems to have no effect in this subrange).

We continue the sound analysis, by modifying the different variables \( (V, Ps, D) \) of the test setup and we analyze the influence of these changes.

Figure 39 is the sound measurement for \( Ps = 300 \) Pa and different velocities. The higher is the grazing flow velocity \( V \), the stronger the sound level is until 1 kHz. Over 1 kHz, it seems that the sound levels are very close to each other. We see that, over 1 kHz, the highest \( V \) \( (V = 4m/s) \) make less noise than the lowest one \( (V = 2m/s) \). We verified it for other similar experiments and the explanation we provided in the last paragraph (the increase of grazing flow breaks the air flow going through the holes) could be the solution. Extra measurements are needed.
Figure 39: A straight duct with different \( V \), \( P_S = 300 \text{Pa} \) and measured for holes with \( D = 1 \text{cm} \).

Figure 40 gives the measurements for 100 Pa. Below 3 kHz, the higher is the velocity, the stronger the sound level is. Over 3 kHz, the background level is too high and the sound generated is too low.

Figure 41 shows the sound measurements for 3m/s and for different flow velocities. Over 800 kHz, the static pressure seems to be very influential on the SPL. Otherwise, below 800 kHz, the different curves are more gathered. The statics pressure seems to be more influential at high frequency.
FIGURE 41: A STRAIGHT DUCT WITH V=3M/S, AND DIFFERENT STATIC PRESSURES (PS) AND MEASURED FOR HOLES WITH D=1CM.

To do general conclusion on the different variables of influence, we need to plot different velocities and static pressures on the same figure. It makes the figure heavier and more difficult to understand, but we can observe if there are coupled effects between the curves. This is what we do in Figure 42.

FIGURE 42: SPL FOR A STRAIGHT DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED FOR HOLES WITH D=1CM WHICH COMPLETES FIGURE 35 AND FIGURE 36.
The different analysis we did concerning Figure 39, Figure 40 and Figure 41 let us think there are two frequency ranges. The first frequency subrange is between 100 Hz and 1 kHz and the second one between 1 kHz and 10 kHz. Figure 42 highlights the coupled effects we observed in the sound measurement. We will explain these effects in the two next paragraphs.

**Between 100 Hz and 1 kHz, for a fixed pressure**, we have a group of three different curves, one for each static pressure (100, 200 and 300 Pa) (Figure 41 presents the case V=3m/s and all the cases are included in Figure 39). These three curves of different Ps (for a fixed velocity V) are then very close to each other’s and we can observe (with difficulties in Figure 42) that they are all the more close since V is high. It means that the higher is the grazing flow velocity, the smaller the influence of the static pressure is between 100 Hz and 1 kHz.

Another conclusion, between 100 Hz and 1 kHz is that we have always a higher sound power level (SPL) with a higher static pressure. On the other hand, we found, in the turbulence section that the higher was the static pressure, the lower the turbulence level was. It is exactly the inverse for the SPL. From this observation, we could conclude that the turbulence analysis is not enough for predicting the sound measurement. Actually, the sound generation is extremely complex and the turbulence level is only one measurement and cannot fully explain the sound generation. What we investigate on, is the influence of the turbulence level and if it can help in the conception of ventilation systems.

**Between 1 kHz and 10 kHz**, for a fixed grazing flow velocity V, we have three different curves, one for each static pressure Ps (Figure 39 and Figure 40 presents respectively the cases Ps = 300 Pa and Ps = 100 Pa). Over 1 kHz, the curves of the same pressure are converging to each other’s. In this frequency range the higher is the pressure, the stronger the sound power level is. It seems that the static pressure is the variable of the highest influence. The curves of the highest velocity (for a fixed $P_s$) are in general lower compared to the other curves of the same pressure.

Over 1 kHz, the sound generated are caused by $V_h$ (see dipoles equation) and we assumed that the grazing flow makes decrease $V_h$ (need to be check with extra measurements) and then reduces the sound generated at high frequency caused by dipoles.

Figure 43 compares the sound power level over 1 kHz with different velocities (0m/s included). We can see that if we have no grazing flow, the sound is higher at high frequency. It is true for 300 Pa (Figure 43) and also for 200 Pa. For 100 Pa the sound is higher for 4 m/s then for 0m/s. Nevertheless, the sound is higher for 0m/s than 3 m/s.

A general conclusion could be that, the grazing flow increases both turbulence (according to section 4.2) and the sound power level below 1 kHz. Over 1 kHz, it seems that the statics pressure is the most important variable and we assume that the sound power level is decreasing due to lower $V_h$ (which needs to be verified in another works).
Furthermore, Figure 44 compares the sound level with different holes diameters $D$ for $v=3$ m/s, and two different static pressures (100 Pa and 300 Pa). The sound generated is always lower for $D=0.5$ cm (see Figure 44). The sound generation can also be explained by the dependency of the sound generated by dipoles to $D^2$. The fact that the sound is lower for $D=0.5$ cm is the reason we choose to use the large nozzles in the analysis. The difference between the background level and the sound generated is higher and this is a good argument for this choice. Moreover the fact that the holes are smaller implies that the close surrounding is smaller. The influence of the position is more important and the velocity fluctuations detected by the probe are smaller. To make good acquisition seems easier with the larger holes (more noise and a larger turbulent surrounding around the nozzles) and the measurement uncertainty is also lower (for both the turbulence level and the sound). We choose to analyze in priority the measurement for $D=1$ cm.
Extra measurements are added to Annex 5 (for instance Figure 86, Figure 87, Figure 88 and Figure 89).

4.4 **INFLUENCE OF THE DUCT EXPANSION WITHOUT GRAZING FLOW \((V=0 m/s)\)**

For this section, we use the second prototype with a duct expansion before the nozzles. The length between the expansion and the nozzles can be modified and two configurations were used: \(l = 10 \text{ cm} \) and \(20 \text{ cm} \). In the expanded duct analysis, in sections 4.4, 4.5 and 4.6, we decide to use \(l = 20 \text{ cm} \). This choice is justified in the last section of Chapter 4, section 4.8.

The damper (see Figure 20) is totally closed and no grazing flow is going through the duct \((V = 0 m/s)\). We organize this section as we did in 4.1. We start with the turbulence analysis and then we finish with the sound analysis.

**Concerning the turbulence analysis**, Figure 45 compares the turbulence level with and without the expansion with holes diameter of 1 cm. The dash line are for \(P_s = 100 \text{ Pa} \) and the full line, for 300 Pa.

![Figure 45: SPL [dB] for a straight and an expanded duct with \(l=20 \text{ cm} \) and \(D=1\text{cm} \), for different pressures and no air flow.](image)

Figure 45 shows that the turbulence level (between 10 Hz and 1 kHz) is much higher with an expansion. For this measurement, we have no grazing flow (the damper is close). We expected that the influence of the expansion will be low without grazing flow and the first guess should be that the fact we have no air flow, means that the holes (and then the probe) see exactly the same set-up as it was without the expansion (and without air flow). Nevertheless, we observe, this is not the case and the expansion modifies the turbulence level a lot. We underestimate the modification of the turbulence generated by the expansion despite of the low air flow going through the nozzles. Extra measurements, for \(D = 0.5 \text{ cm} \) (Figure 90), are provided in Annex 5.

Furthermore, Figure 46 gives the turbulence level for different static pressures (100 Pa, 200 Pa and 300 Pa) for different holes diameters (0.5 cm and 1 cm) and measured at 5 mm to the holes.
The turbulence levels for the holes of 1 cm (the dash line) are always higher than for the holes of 0.5 cm (the full line). As we said for the straight duct when we increase the holes diameter, there is a higher air flow rate going through the holes and the turbulence amplitude is increased as well. In general we prefer to use the large holes diameter in the analysis (as we did in the straight duct analysis).

**FIGURE 46:** PSV FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURES (PS) AND y=5MM WITH DIFFERENT DIAMETERS D AND L=20 CM.

Figure 47 shows the different turbulence amplitudes for different probe's positions in the duct, y = 5 mm (for the dash line) and 23 mm (for the full line) and for different static pressure (100 Pa, 200 Pa and 300 Pa). In general, the turbulence level is higher when the probe is further from the hole (23 mm). We explained this observation in straight duct by saying that the static pressure makes order in the close surrounding of the hole. Consequently, the further from the holes the probe is, the higher the turbulence level is.

**FIGURE 47:** PSV FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED AT DIFFERENT POSITIONS WITH DIFFERENT DIAMETERS D AND L=20CM.
Concerning the sound analysis, we can start with the comparison of the sound generation with and without the duct expansion. Figure 48 shows the sound power level for the straight duct and the expanded duct and for the holes diameter \( D = 0.5 \text{ cm} \) (the dash line) and \( D = 1 \text{ cm} \) (the full line). The static pressure is fixed at 100 Pa.

**FIGURE 48 : SPL [DB] FOR A STRAIGHT AND AN EXPANDED DUCT WITH L=20 CM, FOR PS=100 PA AND NO AIR FLOW.**

In Figure 48, we see that the sound level is very low for the holes diameters of 0.5cm (as it was the case for the straight duct) compared to the background noise and we can observe a modification of the sound level with the adding of the expansion. Otherwise, between 200 Hz and 3 kHz, we see that the SPL is much higher with the adding of the expansion (comparison of the light blue and the red curve).

In Figure 49 we increase the static pressure (300 Pa) in order to have a higher air flow going through the holes and also a higher sound level. The sound power levels for the small holes are increased and are not spoiled by the background noise. We observe an increase of the sound power level with the adding of the duct expansion and the increasing can be around 5 dB which is a huge difference. We can also see that in Figure 48, the SPL is higher for the larger holes.
FIGURE 49: SPL [DB] FOR A STRAIGHT AND AN EXPANDED DUCT WITH L=20 CM, FOR PS=300 PA AND NO AIR FLOW.

We have also proved that both situations, with and without the expansion, are not totally similar. The turbulence level (PSv) is quite low for high frequency but we can detect a difference of around 5 dB for the sound power level. On an acoustical point of view, we cannot neglect such a difference. A deeper investigation need to be done to explain why we have big differences around 1 kHz for the sound measurement and no difference for the power spectrum of the velocity.

Figure 50 can be used to compare the influence of the static pressure (taken equal to 100 Pa and 200 Pa) with the large holes. The static pressure makes increase the velocity through the holes (Vh) and make increase the sound level.

FIGURE 50: SPL FOR AN EXPANDED DUCT WITH NO GRAZING FLOW, DIFFERENT STATICPressures (PS) AND MEASURED FOR DIFFERENT HOLES DIAMETERS WITH L=20CM.
Extra measurement are also provide for L=10 cm in Annex 5, Figure 92 for instance.

To conclude on the sound measurements, the adding of the duct expansion (without grazing flow) increases significantly the sound power level. The influence of the holes diameter D and the static pressure is similar to the straight duct. The increasing of D or Ps increases the SPL and it is predictable by the dipole theory, which is developed in the theory chapter.

We can finally say that in this section, both the turbulence level and the sound power level are increased by the adding of the duct expansion. The adding of the expansion increases the turbulence level between 10 Hz and 1 kHz and the sound power level, between 200 Hz and 3 kHz/5 kHz. This increase of the sound cannot be neglected and can be easily heard during the measurement session.

We will now investigate on the influence of the grazing flow (the velocity V through the duct) in the duct with an expansion.

4.5 TURBULENCE ANALYSIS WITH A GRAZING FLOW (V>0M/S) IN THE EXPANDED DUCT

For this section and the next one, the measurements were made for a specific static pressure (Ps), a certain grazing flow velocity (V) and a known diameter (D). The damper is not totally closed but partially, in order to reach the needed static pressure (Ps) and the mean flow velocity (V) through the duct.

We start in this section with the turbulence analysis in the duct with an expansion and with a grazing flow. In Figure 51, we use the turbulence intensity TU to compare the turbulence level with and without an expansion, with different holes diameters. We think that the turbulence TU is a good way to compare different situations on the same graphics.

![Figure 51: Comparison between the turbulence intensities, TU (%), with and without an expansion with L=20cm, PS=100 PA for different velocities through the duct.](image)

In Figure 51, we see that the adding of the expansion, when we have a grazing flow in the duct, increases the turbulence intensity (TU) significantly (bleu & red vs. green & purple). Moreover when we increase V, TU does not increase a lot in the straight duct. In the expanded duct, TU increases with the grazing flow velocity V (the green and the purple graphs). It means that the
expansion generates more eddies since the grazing flow velocity increases. (Extra diagram are added in Annex 5, Figure 109 and Figure 110).

Moreover, Figure 52 gives the mean flow measured by the probe $V_p$ (measured at 5mm from the hole). We can see than when we add the expansion, $V_p$ decreases. The reason is the air flow pattern is not fully developed (2.1.4 Turbulence generation of an expansion). The detected air flow $V_p$ is lower when we add an expansion before the nozzles and it is verified for the large and the small holes (0.5cm and 1cm).

Moreover, Figure 52 gives the mean flow measured by the probe $V_p$ (measured at 5mm from the hole). We can see than when we add the expansion, $V_p$ decreases. The reason is the air flow pattern is not fully developed (2.1.4 Turbulence generation of an expansion). The detected air flow $V_p$ is lower when we add an expansion before the nozzles and it is verified for the large and the small holes (0.5cm and 1cm).

Moreover, the Mean flow $V_p$, detected by the probe, has to be seen as a combination of the velocity components parallel to the duct direction and another perpendicular to the duct direction and going through the nozzle $V_h$. The expansion reduces the measured mean flow $V_p$. But we do not know how the velocity $V_h$ is affected. As we have already said in other sections, we should measure $V_h$ with a probe outside the duct, closed to the hole. With this method, we could figure out how the expansion reduces $V_h$ and we could conclude about the sound generation with the dipole theory.

Figure 53 gives the turbulence intensity $TU$, for a fixed grazing flow velocity (3m/s) and different static pressures (100 Pa, 200 Pa and 300 Pa).
In Figure 53, the diagrams show that for different static pressures (Ps) the expansion makes increase the turbulence intensity (TU) as we observed at 100 Pa, in Figure 51. Otherwise, the influence of the expansion at different static pressures is similar.

We will now investigate on the power spectrum of the velocity we polled for different setups. Figure 54 gives the PSv for an expanded duct with D=1 cm and L=20 cm for different static pressures (Ps) and velocities (V'). The PSv curves are similar to the curves for the straight case. The Kolmogorov -5/3 slope is also verified (as we did in 4.2). In the turbulence analysis of the straight duct, we said that different groups of curves were identified. We find in Figure 54, that we have also three groups of curves (the dash lines, the full lines and the dot lines), one for each velocity. Concerning the influence of the static pressure, we reach the same conclusion than in the straight duct with a grazing flow. The higher the static pressure is, the smaller the turbulence level is. It is explained by the ordering effect of the static pressure as we explained.

![Figure 54: PSV for an expanded duct with different V, static pressure (Ps) and measured at 5 mm from the hole with D=1 cm and L=20 cm.](image)

In Figure 55, we can observe the influence of the expansion on the PSVs. It completes what we said on the turbulence intensity TU. We see that the turbulence level is much higher when we have an expansion (the dash lines vs. the full lines) and more turbulence is detected for higher frequencies. For instance the dash lines are bottoming out between 1 and 3 kHz and the full lines still continue to decrease until 5 kHz/10 kHz. It means that the probe detects, in the duct expansion, more fluctuations at high frequency and it also means that it detects smaller eddied.
Finally, to complete the turbulence analysis, we can compare the influence of the probe position to the hole. Figure 56 shows that the further from the hole the probe is, the higher the turbulence level we have (the dash line corresponds to y=5mm and the full to y=23mm). In our analysis we choose in general y=5mm. The reason is the same than in the straight duct case. The turbulence level is more dependent of the static pressure for y=5mm (for a fixed velocity, the curves for different static pressures are closer to each other when we are further from the hole). And we assume that it is better to be closer of the sound sources: the holes.

FIGURE 55 : PSV OF A STRAIGHT AND AN EXPANDED DUCT FOR DIFFERENT STATIC PRESSURES, VELOCITIES WITH Y=5MM, D=1CM AND L=20 CM

FIGURE 56 : PSV FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED AT DIFFERENT DISTANCES FROM THE HOLE WITH D=1 CM AND L=20 CM.
Different Figures are added in Annex 5. For instance we can change the holes diameter, the length L and make extra comparison.

4.6 **SOUND ANALYSIS WITH A GRAZING FLOW \( V>0 \text{m/s} \) IN THE EXPANDED DUCT**

In this section, we do the sound analysis for the duct expansion with a grazing flow. We will use what is already said about the turbulence level in the last section.

To start the sound analysis, we can analyze the influence of the grazing flow on the sound measurements. Figure 57 shows the sound power level for different velocities (0m/s, 2m/s and 4 m/s), different static pressures (100 Pa and 300 Pa) for L=20 cm, D=1cm and y=5mm.

![Figure 57: SPL for an expanded duct with different grazing flow velocities (V=0m/s included), different static pressures (PS) and with D=1cm and L=20cm.](image)

In Figure 57, the SPLs for 4m/s (the dot curves) are always much higher than without grazing flow. Nevertheless, over 1 kHz, we have a less sound level for V=2m/s than 0m/s. In the sound analysis for the straight duct (section 4.3), the sound power level was always higher over 1kHz without grazing flow. It changes with the expansion.

On the other hand, we saw that, in a straight duct, the turbulence level was bottoming out around 2 kHz. For the duct expansion case, we observed that the turbulence level was bottoming out between 5 kHz and 10 kHz. Our guess is the difference of turbulence level that we observed for the expanded duct is the reason we got a higher SPL, with the grazing flow than without, at higher frequencies.
We continue the sound analysis, by modifying the different variables (V, Ps and D) of the test setup and we analyze the influence of these changes.

Concerning the hole diameter influence, we can introduce Figure 58. In this figure, the Sound power level is plotted for an expanded duct with different holes diameters. The main conclusion is that the larger holes generate more sound (and it can be explained with the dipole theory as we did for the straight duct). We have around 5 dB more sound for each third band. That is the reason we used in general cases the large holes to avoid the low sound level which could be spoiled by the background noise.

![Figure 58](image-url)

**FIGURE 58**: SPL FOR AN EXPANDED DUCT WITH DIFFERENT GRAZING FLOW SPEED, DIFFERENT STATIC PRESSURE (PS), DIFFERENT HOLES DIAMETERS AND L=20CM.

In the next comparison, we decide to use D=1cm and to investigate on the influence of the grazing flow and the static pressure on the sound level.

Figure 39 is the sound measurement for Ps=300 Pa and different grazing flow velocities V. The higher is the grazing flow velocity V, the stronger the sound level is. In the straight duct case, we found that at 1 kHz, the curves was converging each other’s. Over 1 kHz, the grazing flow was not really influencing the SPL. With an expansion, we cannot observe this phenomenon. We assume that the turbulence is modifying the sound at high frequency and it was not the case in the straight duct.
In Figure 60, we plot the SPL for different velocities and a fixed static pressure \( (Ps=100 \text{ Pa}) \). We have exactly the same observation than we did at 300 Pa. The SPL increases with the grazing flow and we assume than we cannot observe the convergence of the three curves as it is in the straight duct because of the turbulence level at high frequency.

Figure 61 gives the sound power level for different static pressures (100 Pa, 200 Pa and 300 Pa) and for a fixed grazing flow velocity \( (V=3\text{m/s}) \) (the other parameters are \( D=1\text{cm} \) and \( L=20 \text{ cm} \)). We see that, the three different curves, one for each static pressure, are close to each other’s between 50 Hz.
and 500 Hz. It means that static pressure has not a huge influence of the sound. After 800 Hz, the curves are not close to each other’s.

![Figure 61: SPL for an expanded duct with V=3m/s and different static pressures (Ps) and measured for holes with D=1cm and L=20cm.](image)

Different conclusions can be made with the last observations we did. Firstly, we can refresh the conclusion we found in 4.2 and 4.3 concerning respectively the turbulence and the sound analysis in the straight duct. For the straight duct, there was a limit at 1 kHz between two subranges:

- Before 1 kHz, the grazing flow velocity \( (V) \) was the main variable of influence for the sound level. Concerning the turbulence level, the turbulence Level was bottoming out around 1 kHz and the velocity was also the main variable of influence.
- Over 1 kHz, the static pressure \( (Ps) \) was the main variable of influence for the sound level and we tried to explain the different observations with the dipole model, concluding that extra measurements are required to fully explain the observation.

The turbulence level was too low for this frequency range.

The general conclusion for the straight case was that the sound generated before 1 kHz was generated by the high level of turbulence. Over 1 kHz, we assumed that the static pressure increases \( V_h \) and increases the sound generation according to the dipoles theory.

For the duct expansion, we observed that both the static pressure \( Ps \) and the grazing flow velocity \( V \) have a high influence on the sound generation for the whole frequency range.

Furthermore, according to 4.5, the turbulence level is highly and mainly dependent of the grazing flow \( V \) (the static pressure does not modify a lot the turbulent level). It means that in Figure 61, plotted for different pressures \( Ps \) and a fixed \( V \), suggests that for a similar turbulence pattern in the duct, the sound power levels are quite similar between 50 Hz and 800 Hz. It leads that until 800 Hz, the turbulent eddies are mainly responsible of the sound generation. Over 800 Hz, both \( V \) and \( Ps \) are very influent and there is the superposition of different effects (the turbulence generates noise and the static pressure generate sound due to \( V_h \)).
Figure 62 gives a global overview of the situation in the expanded duct for different statics pressures $P_s$ and grazing flow velocities. This figure is not easily readable. The reason is that we cannot watch the coupled effect we explained in section 4.3. And this is finally a good way to conclude the analysis of the expanded case by explaining that the flow pattern in the expansion is really complicated to describe. Different variables ($L$, $P_s$, $D$ and $V$) are involved and different phenomena are superposed. No general conclusion can be stated.

It does not mean that we cannot correlate the turbulence to the sound generated. We know now that the turbulence in the expansion duct is much higher than in the straight duct and we feel that the increase of both the turbulence and the sound with the adding of the expansion, are correlated.

The purpose of the next section aims to clarify this matter.

![FIGURE 62: SPL FOR AN EXPANDED DUCT WITH DIFFERENT GRAZING FLOW SPEED, DIFFERENT STATIC PRESSURE (PS) AND WITH D=1CM AND L=20CM.](image)

4.7 **Comparison between Turbulence and Sound Generation with and without an Expansion**

The expansion creates turbulences as we described in the last sections. We will clearly develop and quantify how the turbulence level influences the measured sound power level when we have a grazing flow.

In that section the Power spectrum of the velocities ($P_{Sv}$) are often converted into third octave band to be able to compare them with the sound power level ($SPL$).

In the last section, we reach different conclusions concerning the turbulence level and the sound power level. One of them was that both the turbulence level and the sound increase with the adding of the duct expansion (dash lines vs. full lines).
In Figure 63, the PSv is plotted for 100 Pa and 300 Pa for 2m/s. The turbulence level (PSv) is much higher with an expansion.

In Figure 64, we plot the sound power level for 100 Pa, and for both configurations, with and without the duct expansion. The sound power level are much higher (5 and 15 dB of increase) when we have the duct expansion. (Extra measurements are available in Annex 5 for instance Figure 111)

In the next figures, both PSv and SPL are plotted in third octave band on the same graphics to simplify the comparison. We take care to scale correctly the different axis (the left axis for the SPL [dB] and the right axis for the PSv [dB]). In that way, we can evaluate the influence of the grazing flow...
V (Figure 112) on the PSV and the SPL, on the same time. Figure 65 compares the PSV and the SPL with and without an expansion for a low grazing flow (2m/s) and 100 Pa. We can easily see the consequences of adding an expansion on the turbulence level and on the sound. The turbulence level seems to increase with an expansion much more than the sound power level. If the SPL increases of 10 dB with the adding of the expansion, the PSV can increase of 50dB. The phenomenon is not linear but we have a kind of correlation between the sound and the turbulence.

![Figure 65](image)

**FIGURE 65**: SPL and PSV in third octave band for a straight duct and an expanded one with $D=1\text{cm}$, $Y=5\text{mm}$ and $L=20\text{cm}$ and for $V=2\text{m/s}$.

We know that for 100 Pa and a low grazing flow, the background level is too close to the sound level. Figure 66 is the same figure than Figure 65 but we modified the grazing flow velocity ($V=4\text{m/s}$) to have better sound acquisition (a higher SPL compared to the background noise). We can observe that the increasing of sound level seems to be correlated to the turbulence level as well.

![Figure 66](image)

**FIGURE 66**: SPL and PSV in third octave band for a straight duct and an expanded one with $D=1\text{cm}$, $Y=5\text{mm}$ and $L=20\text{cm}$ and for $V=4\text{m/s}$.
Finally to conclude on the correlation between the increasing of sound and turbulence, due to the expansion, we can compute the difference of the SPLs (respectively SPs) between the expanded duct and the straight one:

\[
\Delta SPL = SPL_{\text{Expanded duct}} - SPL_{\text{Straight duct}}
\]

\[
\Delta PSv = SPv_{\text{Expanded duct}} - SPv_{\text{Straight duct}}
\]

In Figure 67, we computed the differences $\Delta SPL$ and $\Delta PSv$ for a static pressure of 100 Pa and two different velocities 2m/s and 4m/s.

![Figure 67: Difference between SPL (or of the PSv) for an expanded duct and a straight one (measured with Y=5MM, D=1CM, L=20CM, PS=100PA and V=2-3 M/S) in third oCave band.](image)

In Figure 67, the blue curve ($\Delta SPL$) and the red curve ($\Delta PSv$) are more or less correlated (for 2m/s) with a little shift. The green curve ($\Delta SPL$) and the light blue curve ($\Delta PSv$), corresponding to 3m/s, have a peak at the same position, which means that the increasing of the SPL with the expansion is correlated to the increasing of the PSv. We can also conclude with this figure, that the turbulence modify the sound generation especially around 1 kHz.

Figure 68 shows $\Delta SPL$ and $\Delta PSv$ when we have 4m/s (and 100 Pa). The increasing of the turbulence level (with the adding of the expansion) is especially high around 3kHz. The sound level also increases around 3 kHz. Nevertheless, the sound power level has an increasing around 100 Hz which cannot be explained by an increase of turbulence level. As we have already concluded, the sound generation is a complex phenomena and the turbulence level is not the only sound source. The sound source is the consequence of a flow pattern in a complex geometry and the only conclusion is the turbulence makes increase the sound level and need to be avoided.
Finally, we wanted to show that the correlation observed on the two last figures is not by chance, and we also have to check this correlation for the smaller holes diameter ($D = 0.5 \text{ cm}$) for $2 \text{ m/s}$ and $3 \text{ m/s}$ at $100 \text{ Pa}$ (Figure 69). The correlation is even better than in the previous figures and the different curves for $\Delta SPL$ and $\Delta PSV$ have peaks at the same frequency (we compare the blue with the red curve and the green with the light blue curve).
Figure 70 gives the correlation for the small holes for 4m/s. As we have already noticed, for 4m/s, the sound level increase with the expansion adding, is not fully explained around 100 Hz. Otherwise, both the turbulence level and the sound level are correlated around 3 kHz.

![Figure 70: Difference between SPL [DB] (or of the PSV) for an expanded duct and a straight one (measured with Y=5mm, D=0.5CM and L=20CM, PS=100PA and V=4M/S) in third octave band](image)

Other graphics are added in Annex 5, for instance Figure 113, Figure 114, Figure 115 and Figure 116. In 70 %, we found a correlation between the sound increase and the turbulence increase.

We can conclude this section by saying we have a correlation between sound and turbulence but we also need to add that the turbulence cannot fully explain the sound generation due to the complexity of the different phenomena involved in the sound process.

### 4.8 Influence of the length L from the holes to the expansion

This extra section is written to explain the reason we choose L=20 cm when we analyzed the measurements for duct expansion.

In the prototype setup, the length between the holes and the expansion L, can be taken as 20 cm and 10 cm. In both cases, according to the theory, the air flow is not fully developed. Then, we are sure to have more turbulence than in the straight duct case.

Firstly, we analyze the turbulence level in both cases (with L=10 cm and L=20 cm). Figure 71 gives the difference of the PSvs (between the cases L=20cm and L=10cm) measured at 5 mm from the hole for different static pressures Ps and grazing flows (0m/s included). In this graphic, we plot, on the ordinate axis, the value calculated with:

\[
\Delta SPL = SPL_{Expanded\ duct\ with\ L=20cm} - SPL_{Expanded\ duct\ with\ L=10cm}
\]
We find that without grazing flow, we have more turbulences when L=10 cm. This is not fully explained but we guess that the wedge of the expansion disturbs the flow pattern and generates turbulence. Otherwise we have in general more turbulence in L=20 cm. It seems that the higher the static pressure is, the larger this difference will be as the light blue curve shows.

To explain this increase of turbulence we can easily understand that when the nozzles are at 10 cm from the expansion wedge, the nozzles are too close to the expansion. The turbulent air flow is disturbed by the wedge and cannot circulate correctly. The eddies have more difficulties to be formed. A deeper investigation is needed if we want exactly to describe the flow pattern in wedge.

Figure 72 shows the difference of PSv as we did in the last figure, which we measured at 23 mm of the wall. The probe is further from the hole and we observe that the differences are lower certainly due to the fact that in this position, the probe detects similar amount of eddies at 10 cm and 20 cm.
Concerning the sound analysis, Figure 73 gives the sound power measurements for different velocities and pressures and different lengths L. The dot curves correspond to \( V = 4 \text{ m/s} \) and the full curves to \( 2 \text{ m/s} \). We have always a higher sound level when L=20 cm. There are different reasons. The higher turbulence level for \( L = 20 \text{ cm} \) are certainly one of them. We could also imagine that the air flow pattern coming from the small duct upstream, makes decrease the static pressure in the wedge when the air flow go through the expansion. If we verify it, it could explain that there are less sound for \( L = 10 \text{ cm} \), because we are too close to the wedge where the static pressure is lower (which leads a lower \( V_p \) and then a lower sound generation).

![Figure 73](image)

**FIGURE 73**: SPL FOR AN EXPANDED DUCT WITH DIFFERENT \( V \), STATIC PRESSURE (PS), DIFFERENT \( L \) AND WITH HOLES OF DIAMETER OF 1CM. **FIGURE 73 COMPLETES THE SOUND MEASUREMENT ALREADY INCLUDED IN FIGURE 62 AND FIGURE 58.**

Extra measurements are provided in Annex 5 (for instance Figure 99, Figure 100, Figure 101 and Figure 105).

To conclude this section, we have both higher turbulence level and sound level when L=20cm. We use for this reason the measurements done for L=20 cm. To explain this difference between L=10 cm and L=20 cm, we advice to use another process to observe exactly where the eddies can be formed after an expansion (for instance a smoke analysis with Plexiglas ducts). We could investigate, in detail, where the nozzles are exposed to more turbulence and will generate certainly more sound.
5 CONCLUSION

The designing of ventilation systems needs always to use trial measurements to verify that the final product will not generate too much noise. In industry, this is the normal way of proceeding and a company expects to finalize a project with concrete results and not only simulations and theoretical results.

This thesis aimed to combine aeroacoustic models with systematic measurements.

The measurements for different configurations were analyzed for different configurations (especially with and without expansion with different parameters) and general conclusions are stated on the sound generation and the influence of the turbulence in the sound mechanisms.

The statics pressure makes decrease the turbulence level in the close surrounding of the holes and increase the sound generation. For a straight duct, the grazing flow can make decrease the sound generation but it is not verified in the expanded duct, where the turbulence and the sound level are increased when there is a grazing flow. Finally, a correlation between the sound and turbulence level exists but the turbulence does not fully explain the sound generation. The turbulence level is only one measure and it describes partially a complex sound mechanism.

To conclude, the thesis combines as much as possible the theoretical knowledge with the measurements analysis. It is showed for instance that the pattern of the turbulence level with a grazing flow corresponds to the Kolmogorov prediction and it shows that the used method is consistent. The dipole model was also often used during the analysis and it is proved that without grazing flow, the increase of noise is predictable using the dipoles model.

The feeling that the turbulence is involved in the sound generation, is become definitely a scientific fact and it suggests to take it into account when engineers design the ventilation units.
6 FURTHER WORKS

During this thesis project different ideas of investigation were set aside.

Firstly, a hotwire probe could be positioned outside the duct, very close to the holes in order to measure the flow velocity going through the holes. This measurement is useful for improving the sound analysis (with the dipole theory). The velocity through the holes indicates precisely how much air flow is going through the holes and the sound power generated could be estimated and compared to the sound measurement.

Another idea is to use others tools to investigate on the global air flow pattern in the close surrounding of the nozzles and in the wedge of the expansion. A smoke analysis (with Plexiglas duct) could be a good idea to watch directly where eddies are formed and where the flow pattern is fully expanded after an expansion. The smoke analysis is also a solution to improve our understanding of the boundary layer and estimate the boundary layer thickness which could be compared to the theory about the boundary layers we developed in Annex 4.

Finally, another investigation should be done on the sound and the turbulence analysis for different geometries of holes (for instance rounds holes, rectangular nozzles...). An ambitious project could be also done to adapt our analysis in a real product, for instance the chilled beam, which is a special ventilation outlet, developed by FläktWoods and introduced in Annex 6.
REFERENCES


Annex 1  SOUND MEASUREMENT STANDARD

Specification of the method ISO 3741:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>ISO 3741 Standard</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test environment</td>
<td>Reverberation room</td>
</tr>
<tr>
<td>Criterion for suitability of test environment</td>
<td>Room volume and reverberation time to be qualified</td>
</tr>
<tr>
<td>Volume of sound source</td>
<td>Preferably less than 2% of test room volume</td>
</tr>
<tr>
<td>Character of noise from the source</td>
<td>Steady, broad-band, narrow-band or discrete frequency</td>
</tr>
<tr>
<td>Limitation for background noise</td>
<td>$K_1 \leq 0.5 \text{dB}$, $\Delta L \geq 10\text{dB}$</td>
</tr>
<tr>
<td>Instrumentation:</td>
<td></td>
</tr>
<tr>
<td>a) sound level meter</td>
<td>a) class 1</td>
</tr>
<tr>
<td>b) integrating sound level meter</td>
<td>b) class 1</td>
</tr>
<tr>
<td>c) frequency band filter</td>
<td>c) class 2</td>
</tr>
<tr>
<td>d) calibrator</td>
<td>d) class 1</td>
</tr>
<tr>
<td>e) sound intensity instrument</td>
<td></td>
</tr>
<tr>
<td>Sound power levels obtainable</td>
<td>A-weighted and in one-third-octave or octave bands</td>
</tr>
<tr>
<td>Optional information available</td>
<td>Other frequency-weighted sound power levels</td>
</tr>
</tbody>
</table>

^ grade of accuracy: precision = grade 1; engineering = grade 2; survey = grade 3.
^c $K_1$ is the correction for background noise.
dAt least class ** of:
  a) IEC 61672-1, b) IEC 61672-1, IEC 61260, d) IEC 60942, e) IEC 61043
Annex 2  

PICTURES OF THE TEST BENCH IN THE REVERBERATION ROOM IN THE FLÄKTWOODS PRODUCTION PLANT IN JÖNKÖPING (SWEDEN)

In this annex, we add different picture which could be very interesting, especially for repeating the measurements.

FIGURE 74: DANTEC PROBE SUPPORT AND A PROBE BOX.

FIGURE 75: THE DANTEC ANEMOMETER.
FIGURE 76: THE STRAIGHT DUCT PROTOTYPE IN THE REVERBERATION ROOM.

FIGURE 77: THE COMPUTER WITH THE DANTEC ANEMOMETER.
FIGURE 78: THE PROBE SUPPORT MOUNTED IN THE DUCT SEEN BY THE HATCH.
Annex 3  INDEPENDENCE OF NOZZLES

This annex shows that the ten nozzles around a section is independent and do not interact each others.

To prove it, we moved the probe on the median plane between two holes as illustrated on Figure 79. The measurements were done during the first session of measurement and are stored in the appendices book, in appendix 1.

FIGURE 79 : ILLUSTRATION OF THE MEDIAN PLANE BETWEEN TWO HOLES.

In this experiment we can see that the mean velocities registered by the probes are very low as expected. The turbulences are very high compared to the mean flow and the turbulence density is also high consequently. Moreover the mean flow is globally constant. It means that the distance from
the holes to the probes does not influence the measurement. It means that in that plane the probe does not see the holes and another consequence is the two holes doesn’t interact each other.

It means that a hole is an independent source of noise. We can also try to verify it in the sound generation.

For a plugged straight duct we measured the sound power level with a different numbers of nozzles open. If we assume that the background noise is not too high (at least 5-10 dB below the nozzles noise) the fact that we multiply by 2 the number of opened nozzles, the global power of independent sources is multiply by 2 and the sound power level is 3 dB higher (or $10 \log(2)$).

By analyzing the registered sound power level in third octave band, the comparison of the two cases can be made with the next figure.

![Difference of the SPLs when all the nozzles are opened and half of the nozzles, with an increase in section, for different velocities, with $Pr=10^{3}Pa$](image)

The 3 decibels expected are observed but only between 100 Hz and 500 Hz. This is due to the fact that all the nozzles are not exactly the same and the sound generations is highly dependent of the geometry. Furthermore, the holes are not totally independent and it can also dependent of the setup of the experiment ($Ps$, velocity...).

Finally, the background noise is too high for these measurements. The following graphic give an overview of the influence of the background noise on the measurement by measuring the SPL when all the nozzles are opened or closed.
The background noise is 5-10 decibels lower than the noise of the Nozzles only for the frequency range 100Hz to 1000 Hz, where we found the 3 decibels of difference when we doubled the number of nozzles. The background noise is definitely involved if the 3 decibels are not always reached.
Annex 4  THE BOUNDARY LAYER

In this section, we introduce the boundary layer theory which is needed to start an investigation on the boundary layer effect. In our measurement, we ignored if specially for the calibration, where we assumed that the velocity profile was uniform. An investigation in the velocity profile in the FläktWoods duct could be interesting if we need to create a specific model for FläktWoods.

The boundary layer theory is considered like an entire part of the fluid mechanics with a lot of application especially in aeronautics. Two different cases appear in the theory: the laminar boundary layer and the turbulent boundary layer. The transition between the two kinds of boundary layer is more like a mix between the two kinds.

In this part, we will explain the basic case of a flat plate exposed to a homogeneous flow of mean flow U, illustrated with Figure 81. A boundary layer appears close to the flat plat and the laminar boundary layer will be transformed to a turbulent boundary if the Reynold’s number, $Re$ is high enough.

All the issue is to determine the velocity profile $u(x,y)$ in the boundary layer. After that we can easily determine the layer thickness $\delta$ depending of $x$ which is defined as:

$$\delta = y \text{ where } u(x, \delta) = 0.99V$$  \hspace{1cm} (1)

where $V$ is the mean velocity outside the boundary layer (or $U$ in Figure 81).

Before going further, we need to specify that we are in the basic case of a flat plate with an incompressible flow. The theory explained in this section is synthesized and the books “Fundamentals of Fluid mechanics” (Munson, Young, & Okiishi, 1997) and “The handbook of Fluid dynamics” (Johnson, 1998) give a better overview.
The starting point is the Navier-Stokes equations which give the velocity profile. For a steady laminar and a 2D flow, with negligible gravitational effects, the reduced following equations are enough for our study and express Newton’s second law:

\[
\begin{align*}
\frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} &= -\frac{1}{\rho} \frac{\partial p}{\partial x} + v \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \\
\frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} &= -\frac{1}{\rho} \frac{\partial p}{\partial y} + v \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right)
\end{align*}
\]

The conversion of mass equation is also expressed as:

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0
\]

We have the mathematical equations describing the problem, but no analytical solution has been still found.

Prandtl find a solution, considering some approximations, valid for large Reynold’s number flows. One of Prandtl’s student, Blasius, solves these simplified equations for the flat plate case.

The assumption made was:

The normal component to the plate is much smaller than the parallel component and the change of velocity is much higher for the normal component than for the parallel one. It is summed-up with the following equations:

\[
v \ll u \; \text{and} \; \frac{\partial}{\partial x} \ll \frac{\partial}{\partial y}
\]

With this assumption the equations (2.38) and (2.39) can be reduced to:

\[
\begin{align*}
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} &= 0 \\
\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y} &= v \frac{\partial^2 u}{\partial y^2}
\end{align*}
\]

In the last equations, the momentum on the y-axis and the pressure have been eliminated. Only the velocity in function of x and y are still in the equation.

The plate imposed that there is no velocity at y=0. Furthermore when y tends to infinity, the velocity tends to U. The boundary conditions are written as following:

\[
\begin{align*}
u = v &= 0 \; \text{on} \; y = 0 \\
u &\rightarrow U \; \text{as} \; y \rightarrow \infty
\end{align*}
\]

The solution is an approximation and we can notice that the Navier-Stokes equation imposes an elliptical solution (because of the derivatives of order 2 on both directions) whereas the reduced equation should give us a parabolic solution.

The good idea of Blasius was to use a clever change of variable. A function is also defined with dimensionless variables describing the velocity profile.
\[
\frac{u}{U} = g \left( \frac{Y}{\delta} \right) \tag{10}
\]

The \( g \) function has to be determined. First of all, it can be shown that the boundary layer thickness is proportional as seen as following:

\[
\delta \sim \left( \frac{v_x U}{\nu} \right)^{\frac{1}{2}} \tag{11}
\]

The last equation illustrates two different contributions: the viscosity, with \( \nu \) and the inertial forces with \( U \). Furthermore, we introduce dimensionless similarity variable \( \eta = \left( \frac{U}{v_x} \right)^{\frac{1}{2}} Y \) and the stream function \( \psi = \left( v_x U \right)^{\frac{1}{2}} f(\eta) \), where \( f \) is an unknown function. The velocity components of a 2D flow can be also express in function of \( \psi \):

\[
u = \frac{\partial \psi}{\partial y} \quad \text{and} \quad v = -\frac{\partial \psi}{\partial x} \tag{12}
\]

We can rewrite the velocity component as:

\[
u = U f'(\eta) \quad \text{and} \quad v = \left( \frac{v U}{\nu} \right)^{\frac{1}{2}} \left( \eta f' - f \right) \tag{13}
\]

with \( f' = \frac{\partial f}{\partial \eta} \)

The boundary layer equation becomes by using the \( f \) function the differential equation:

\[
2 f''' + f f'' = 0 \tag{14}
\]

And the boundary condition can be rewritten as:

\[
f = f' = 0 \quad \text{at} \quad \eta = 0 \quad \text{and} \quad f' \to 1 \quad \text{as} \quad \eta \to \infty \tag{15}
\]

The differential equation of \( f \) can be integrated with a computer and gives us a solution of \( f \) in function of \( \eta \) which is a variable defined in function of \( x \) and \( y \).

The boundary layer thickness can be also determined from the computed solution introduced with the figures below. We have \( \frac{u}{U} \approx 0.99 \) at the boundary layer limit, and it corresponds to \( \eta = 5.0 \). Thus,

\[
\delta = 5 \sqrt{\frac{v_x}{U}} \quad \text{or} \quad \frac{\delta}{x} = \frac{5}{\sqrt{Re_x}} \tag{16}
\]

with \( Re_x = U x / \nu \).

The general solution found numerically is illustrated with Figure 82:
The approximation of Blasius is one among others. We can use other dimensionless constants and define other models of approximation as the graphics and the table illustrate below. The linear, parabolic cubic and sine wave models are compared to the Blasius solution.

The solutions above are calculated using the flat plate momentum-integral results assuming the laminar flow velocity profiles. This method is described in more detail in (Munson, Young, & Okiishi, 1997), in “Fundamentals of Fluid mechanics”.

If the geometry is more complex, there are other solutions and techniques to solve it. In first approximation we could say that the used pipes have a radius of curvature large enough to
approximate it with a plate. In reality we can’t. The ducts used in ventilation system are very specific and the walls are not smooth. Furthermore the order of Reynold’s number is quite high and we are in the turbulent case.

Furthermore, we made our own investigation on the boundary layer and we try to delimitate the boundary layer in the FläktWoods duct.

For the boundary layer measurement we used the measurement of the first measurement session. The prototype had been improve between the first and the second session but the measurements remain enough trustworthy.

According to the measurements, the boundary layer seems to be very thin and the positioning of the probe needs to be done with a higher accuracy (for instance, we cannot see the velocity decreasing because we are not close enough). The numerical simulation of the duct could be a solution but we should certainly investigate a long time on the non linearity of the fluid equation and how to find a numerical solution in a proper way.
Annex 5  ADDITIONAL TURBULENCE AND SOUND GRAPHICS

The following figures represent extra graphics from the measurement session 2.

Graphics for a straight duct:

FIGURE 86: PSV FOR A STRAIGHT DUCT WITH DIFFERENT V, STATIC PRESSURES (PS) AND MEASURED AT 5 MM FROM A HOLE WITH D=0.5CM.

FIGURE 87: PSV FOR A STRAIGHT DUCT WITH DIFFERENT V, STATIC PRESSURES (PS) AND MEASURED AT 23 MM FROM A HOLE WITH D=0.5CM.
FIGURE 88: SPL FOR A STRAIGHT DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND WITH D=1CM.

FIGURE 89: SPL FOR A STRAIGHT DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND WITH D=0.5CM.

Graphics for an expanded duct:
FIGURE 90: SPL (dB) for a straight and an expanded duct with $L=20$ cm and $D=0.5$ cm, for different pressures and no air flow.

FIGURE 91: PSV for an expanded duct with different $V$, static pressure (PS) and $Y=5$ mm with different diameter $D$ and $L=10$ cm.
FIGURE 92: SPL FOR AN EXPANDED DUCT WITH NO GRAZING FLOW, DIFFERENT STATIC PRESSURES (PS) AND MEASURED FOR DIFFERENT HOLES DIAMETERS WITH L=10CM.

FIGURE 93: PSV FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED AT 5 MM FROM THE HOLE WITH D=1 CM AND L=10 CM.
FIGURE 94: PSV FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED AT 5 MM FROM THE HOLE WITH D=0.5 CM AND L=10 CM.

FIGURE 95: PSV FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED AT 23 MM FROM A HOLE WITH D=1 CM AND L=10 CM.
FIGURE 96: PSV FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED AT 23 MM FROM A HOLE WITH D=0.5 CM AND L=10 CM.

FIGURE 97: PSV FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED AT 5 MM FROM THE HOLE WITH D=0.5 CM AND L=20 CM.
FIGURE 98: COMPARISON OF THE PSV MEASURED AT 5 MM AND 23 MM FROM THE HOLE FOR A STRAIGHT DUCT WITH DIFFERENT V AND STATIC PRESSURE (PS). THE HOLES DIAMETER D IS EQUAL TO 0.5 CM.

FIGURE 99: FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED AT DIFFERENT DISTANCES FROM THE HOLE WITH D=0.5 CM AND L=10 CM.
FIGURE 100: FOR AN EXPANDED DUCT WITH DIFFERENT $V$, STATIC PRESSURE ($PS$), LENGTH ($L$) AND MEASURED AT DIFFERENT DISTANCES FROM THE HOLE WITH $D=1$ CM.

FIGURE 101: PSV FOR AN EXPANDED DUCT WITH DIFFERENT $V$, STATIC PRESSURE ($PS$) AND MEASURED AT DIFFERENT DISTANCES FROM THE HOLE WITH $D=1$ CM AND $L=10$ CM.
We have also measured the turbulence further from the nozzle as we can see on Figure 103.

We compare on the following graphics the measures for $y = 5\, mm$ and $y = 23\, mm$. 

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FIGURE 102: PSV FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED AT 23 MM FROM THE HOLE WITH D=1 CM AND L=20 CM.

FIGURE 103: PSV FOR AN EXPANDED DUCT WITH DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED AT 23 MM FROM THE HOLE WITH D=0.5 CM AND L=20 CM.
FIGURE 104: PSV FOR AN EXPANDED DUCT WITH DIFFERENT $V$, STATIC PRESSURE ($P_S$) AND MEASURED AT DIFFERENT DISTANCES FROM THE HOLE WITH $D=0.5$ CM AND $L=20$ CM.

FIGURE 105: DIFFERENCE OF THE PSV FOR AN EXPANDED DUCT FOR $L=20$ CM AND $L=10$ CM FOR DIFFERENT $V$, STATIC PRESSURES AND DIFFERENT POSITIONS IN THE DUCT.
FIGURE 106: INFLUENCE OF THE EXPANSION IN THE DUCT ON THE PSV FOR DIFFERENT V, STATIC PRESSURE (PS) AND MEASURED 5 MM FROM THE HOLE WITH D=0.5 CM AND L=20 CM.

FIGURE 107: PSV OF A STRAIGHT AND AN EXPANDED DUCT FOR DIFFERENT STATIC PRESSURES, VELOCITY WITH Y=5MM, D=1CM AND L=10 CM.
FIGURE 108: PSV OF A STRAIGHT AND AN EXPANDED DUCT FOR DIFFERENT STATIC PRESSURES, VELOCITY WITH Y=5MM, D=0.5CM AND L=10 CM.

FIGURE 109: COMPARISON BETWEEN THE TURBULENCE INTENSITIES, TU (%), WITH AND WITHOUT AN EXPANSION WITH L=20CM, PS=100 PA FOR DIFFERENT VELOCITIES THROUGH THE DUCT.
FIGURE 110: COMPARISON OF THE MEAN FLOW VELOCITIES DETECTED BY THE PROBE, $V(\text{m/s})$, WITH AND WITHOUT AN EXPANSION WITH $L=20$CM, $PS=100$ PA FOR DIFFERENT VELOCITIES THROUGH THE DUCT.

FIGURE 111: SOUND POWER LEVEL FOR A STRAIGHT DUCT AND AN EXPANDED ONE WITH Y= 5 MM, D=1CM AND $PS=300$ FOR DIFFERENT GRAZING FLOWS $V$. 
FIGURE 112: SPL AND PSV IN THIRD OCTAVE BAND FOR AN EXPANDED ONE WITH D=1CM AND Y=5MM AND FOR DIFFERENT VELOCITIES.

FIGURE 113: DIFFERENCE BETWEEN SPL[DB] (OR OF THE PSV) FOR AN EXPANDED DUCT AND A STRAIGHT ONE (MEASURED WITH Y=5MM, D=0.5 AND L=20CM, PS=300PA AND V=2-3 M/S) IN THIRD OCTAVE BAND.
FIGURE 114: DIFFERENCE BETWEEN SPL [DB] (OR OF THE PSV) FOR AN EXPANDED DUCT AND A STRAIGHT ONE (MEASURED WITH Y=5MM, D=0.5 AND L=20CM, PS=300PA AND V=4M/S) IN THIRD OCTAVE BAND.

FIGURE 115: DIFFERENCE BETWEEN SPL [DB] (OR OF THE PSV) FOR AN EXPANDED DUCT AND A STRAIGHT ONE (MEASURED WITH Y=23MM, D=1CM, L=20CM, PS=100PA AND V=2.35 M/S) IN THIRD OCTAVE BAND.
FIGURE 116: DIFFERENCE BETWEEN SPL[DB] (OR OF THE PSV) FOR AN EXPANDED DUCT AND A STRAIGHT ONE (MEASURED WITH Y=23MM, D=1CM, L=20CM, PS=100PA AND V=4 M/S) IN THIRD OCAVE BAND.
Annex 6  CHILLED BEAM

The “chilled beam” is a product developed by FläktWoods and the master thesis can be used to study this product and the sound generation due to the turbulence.

FIGURE 117: FLÄKTWOODS PRODUCT CALLED “CHILLED BEAM”