DETERMINATION OF THE CORRELATION BETWEEN TURBULENCE INTENSITY AND ACOUSTIC NOISE LEVEL – TWO CLOCKWISE TURNING ROTORS CASE

Student: Omar Vidal Garcia
Supervisor: Jaroslaw Blaszczak, Ph.D.
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1. Introduction

The aim of this project is to find the correlation between turbulence intensity and acoustic noise level.

To achieve this aim a test was performed on the 5th of April 2012 in the acoustic laboratory of the Technical University of Łódź. In this project we are going to explain, how the test was performed, which data we obtained, and which results we got after having analyzed them.

The project has been divided into six parts. The first one is a little introduction. The second one is about sound and noise fundamentals. In this part we are going to talk about the noise and sound concept and about how to do the appropriate measurements.

Afterwards, at the third part, turbulence concept will be developed. This part has been divided into nine parts including the turbulence definition or how to measure turbulent flows.

In the fourth part, it is possible to find a general description about the test performed and the characteristics of the all equipment required.

Next part is the data analysis. In this part the data have been treated for finding the correlation later, in the next part. So, in the part sixth, the correlation between turbulence intensity and acoustic noise level has been found.

Finally, in the last part, the results have been interpreted.
2. Noise and sound fundamentals

2.1 Sound and noise

Sounds are characterized by their magnitude (loudness) and frequency. There can be loud low-frequency sounds, soft high-frequency sounds and loud sounds that include a range of frequencies. The human ear can detect a very wide range of both sound levels and frequencies, but it is more sensitive to some frequencies than others.

Sound is generated by numerous mechanisms and is always associated with rapid small scale pressure fluctuations, which produce sensations in the human ear. Sound waves are characterized in terms of their amplitude or magnitude, wavelength (\( \lambda \)), frequency (\( f \)) and velocity (\( v \)), where \( v \) is found from:

\[
v = f \lambda
\]

The velocity of sound is a function of the medium through which it travels, and it generally travels faster in more dense mediums. The velocity of sound is about 340 m/s in air at standard pressures.

A reference monogram relating wavelength to frequency for the speed of sound under normal conditions is plotted in figure 1

![Figure 1](image.png)

**Figure 1** Wavelength in air versus frequency under normal conditions

This is the simplest of all radiated waves, the plane, progressive wave, so called because it propagates away from the source in one direction only, the wave fronts always remaining parallel to each other. Because it cannot spread
out into the medium, the only attenuation which is experienced is due to transmission losses and dispersion caused by turbulence and temperature gradients within the medium itself. Although the magnitude of a sound wave can be determined in a number of different ways, it is usually more convenient to measure acoustic pressure rather than parameters such as particle displacement or velocity which are extremely different to measure in practice. These parameters are normally only required when measurements are to be made very close to the source in its near field.

The particle velocity here is not necessarily in the direction of travel of the wave, and the sound pressure may vary appreciably at short intervals along the direction of propagation. Under these conditions the acoustic intensity is not simply related to the mean square of the sound pressure. In the far field, however, this relationship is true, and, because sound pressure level is an easy parameter to measure in practice, it has become the most common way of expressing the magnitude of an acoustic field.

2.2 Sound power, energy density and intensity

2.2.1 Sound power

Any source of noise has a characteristic sound power, a basic measure of its acoustic output, but the sound pressure levels it gives rise to depend on many external factors, which include the distance and orientation of the receiver, the temperature and velocity gradients in the medium, and the environment. Sound power on the other hand is a fundamental physical property of the source alone, and is therefore an important absolute parameter which is widely used for rating and comparing sound sources.

2.2.2 Sound-energy density

The acoustic energy contained in a unit volume of the medium is a fundamental parameter of any type of acoustic field. It is termed the energy density and is related to the acoustic pressure by the following equation.
where $D$ is the energy density, $p$ is the root mean square of sound pressure and $\rho c$ is the characteristic acoustic impedance of the medium, which for air at 20 °C is 407 $\frac{kg}{m^2s}$.

### 2.2.3 Intensity

However, the intensity, the acoustic energy flowing through unit area (perpendicular to the direction of propagation of the wave), in unit time, is different for various types of acoustic field. For a diffuse field, such as occurs in a reverberant room, in which there is equal probability of reflected sound waves arriving from any direction, the intensity is

$$I = \frac{p^2}{4\rho c}$$

For a free field in which the sound wave arrives only from the direction of the source

$$I = \frac{p^2}{\rho c}$$

### 2.3 The plane source

Consider an elemental tube of the medium, with unit cross-sectional area and a length equal to the distance travelled by the sound wave in one second, that is, numerically the speed of sound ($c$). If a piston source of power $W$ is constrained by hard walls to radiate all its power into the elemental tube to produce a plane wave, the tube will contain a quantity of energy numerically equal to the power output of the source. Assuming no other losses, the intensity, that is, the acoustic energy flowing through unit area anywhere along the tube in unit time, is independent of the distance from the source and numerically equal to its sound power. Apart from duct systems, plan waves and plan sources are rarely encountered in normal noise measurement situations.
2.4 The point source

Sound sources can be as point sources if their dimensions are small in relation to their distance from the receiver, and many common noise sources, including industrial plant, aircraft, and individual road vehicles can normally treated in this way. As shown in figure 3 the ideal point source can be considered to produce a series of spherical wave fronts resulting from successive disturbances at the point source. For a pure sinusoidal disturbance, the distance between wave fronts representing the successive peak pressures will of course be the wavelength, a fact which is important when considering the effects of reflections within the sound field. As shown in figure 4, the sound energy spreads out equally in all directions so that as it travels further and further from the source its energy is received on an ever larger spherical area. If the medium is assumed to be non-dissipative then the entire power output of the source passes through a spherical shell of radius $r$. The intensity ($I$) is therefore the Power of the source ($W$) divided by the area of this shell

$$I = \frac{W}{4\pi r^2}$$
Figure 3 The propagation of spherical wave fronts from a point source

Figure 4 The dispersion of sound from a point source

It can be seen that the intensity is inversely proportional to the square of the distance between source and receiver, so it attenuates 6 dB per doubling of distance.
2.5 The line source

A line source may be continuous radiation, such as from a pipe carrying a turbulent fluid, or may be composed of a large number of point sources so closely spaced that their emissions may be considered as emanating continuously from a notional line connecting them. In this category are included such factory sources as closely-spaced machines and conveyors, and two extremely important sources of environmental noise, namely roads and railways. A road which has a traffic flow high enough to be a noise nuisance can usually be considered as a line source rather than a succession of single events. Railways are often treated as line sources at the distance from the annoyance.

Very close to or very far from the track, the field is rather more complex. Consider the diagram in figure 5 of part of an infinite line source which has a constant power per unit length. The wave front spreads out from the line in only one dimension perpendicular to its direction of travel, so that any two points at the same distance from the line are on the same wave front and have the same properties. The wave fronts therefore form concentric cylindrical surfaces about the line source as axis. The energy released from a unit length of the source in

![Diagram of a line source](image-url)
unit time passes through the same length of cylindrical surface at all radii. The intensity at given radius is therefore the power emitted by this element \( W \), divided by the area of the cylindrical element surface

\[
I = \frac{W}{2\pi r \cdot 1}
\]

The intensity is therefore inversely proportional to the distance from the source, that is, it attenuates 3 dB per doubling of distance.

2.6 Propagation of sound in air

In addition to the reduction in intensity by distance, there are many others factors which can significantly affect the propagation of sound in a real medium like atmosphere. Velocity and temperature gradients alter the direction of the wave, turbulence distorts it, and viscosity causes absorption. This later effect is far greater for high than for low frequencies, so the atmosphere tends to act as a low pass filter, attenuating high frequencies, and thus distorting the frequency spectrum of a noise, as well as reducing its strength and changing its propagation path. In addition, most measurements are made near ground level where people live and work and where noise is invariably received and, with the notable exception of aircraft noise, produced. For this reason the reflection and absorption of the ground under the path between source and receiver is very important, and must be taken into account as a matter of course whenever studying the transmission of outdoor noise.

Sound frequency denotes the “pitch” of the sound and, in many cases, corresponds to notes on the musical scale (Middle C is 262 Hz). An octave is a frequency range between a sound with one frequency and one with twice that frequency, a concept often used to define ranges of sound frequency values. The frequency range of human hearing is quite wide, generally ranging from about 20 to 20 kHz (about 10 octaves). Finally, sounds experienced in daily life are usually not a single frequency, but are formed from a mixture of numerous frequencies, from numerous sources.
Sound turns into noise when it is unwanted. Whether sound is perceived as a noise depends on subjective factors such as the amplitude and duration of the sound. There are numerous physical quantities that have been defined which enable sounds to be compared and classified, and which also give indications for the human perception of sound.

2.7 Measurement scales

It is important to distinguish between the various measures of the magnitude of sounds: sound power level and sound pressure level. Sound power level is the power per unit area of the sound pressure wave; it is a property of the source of the sound and it gives the total acoustic power emitted by the source. Sound pressure is a property of sound at a given observer location and can be measured there by a single microphone.

Because of the wide range of sound pressures to which the ear responds (a ratio of 105 or more for a normal person), sound pressure is an inconvenient quantity to use in graphs and tables. In addition, the human ear does not respond linearly to the amplitude of sound pressure, and, to approximate it, the scale used to characterize the sound power or pressure amplitude of sound is logarithmic. Whenever the magnitude of an acoustical quantity is given in a logarithmic form, it is said to be a level in decibels (dB) above or below a zero reference level.

Sound intensity, $I$, is defined as the power of the sound per unit area, and so can be measured in watts/m$^2$, but is more commonly measured in units of decibels, as:

$$I = \log_{10}(-I/I_o)$$

where the reference intensity, $I_o$, is often the threshold of hearing at 1000 Hz: $I_o = 10^{-12}$ W/m$^2$.

Because audible sound consists of pressure waves, sound power is also quantifiable by its relation to a reference pressure. The sound power level of a source, $L_w$, in units of decibels (dB), and is given by:
\[ L_w = 10 \log_{10}(P/P_0) \]

with \( P \) equal to the sound power level in units of power density and \( P_0 \) a reference sound power (often \( P_0 = 2 \times 10^{-5} \) Pa).

The sound pressure level (SPL) of a sound, \( L_p \), in units of decibels (dB), is given by:

\[ L_p = 20 \log_{10}(p/p_0) \]

with \( p \) equal to the effective (or root mean square, RMS) sound pressure and \( p_0 \) a reference RMS sound pressure (usually \( 2 \times 10^{-5} \) Pa).

The human response to sounds measured in decibels has the following characteristics:

• Except under laboratory conditions, a change in sound level of 1 dB cannot be perceived.

• Doubling the energy of a sound source corresponds to a 3 dB increase.

• Outside of the laboratory, a 3 dB change in sound level is considered a barely discernible difference.

• A change in sound level of 5 dB will typically result in a noticeable community response.

• A 6 dB increase is equivalent to moving half the distance towards a sound source.

• A 10 dB increase is subjectively heard as an approximate doubling in loudness.

• The threshold of pain is an SPL of 140 dB.

Figure 6 illustrates the relative magnitude of common sounds on the dB scale. For example, the threshold of pain for the human ear is about 200 Pa, which has an SPL value of 140 dB.
2.7.1 Measurement of sound or noise

Sound pressure levels are measured via the use of sound level meters. These devices make use of a microphone that converts pressure variations into a voltage signal which is then recorded on a meter (calibrated in decibels). As described above, the decibel scale is logarithmic. A sound level measurement that combines all frequencies into a single weighted reading is defined as a

**Figure 6**: Sound Pressure Level (SPL) Examples [Source](http://personal.cityu.edu.hk/~bsapplec/sound.htm)
broadband sound level. For the determination of the human ear's response to changes in sound, sound level meters are generally equipped with filters that give less weight to the lower frequencies. As shown in figure 7, there are a number of filters that accomplish this:

- **A-Weighting**: This is the most common scale for assessing environmental and occupational noise. It approximates the response of the human ear to sounds of medium intensity.

- **B-Weighting**: This weighting is not commonly used. It approximates the ear for medium-loud sounds, around 70 dB.

- **C-Weighting**: Approximates response of human ear to loud sounds. It can be used for low-frequency sound.

- **G-Weighting**: Designed for infrasound.

The weighting is indicated in the unit, e.g. measurements made using A-weighting are expressed in units of dB (A).

![Figure 7: Definition of A, B, and C Frequency Weighing Scales](image_url)
Once the A-weighted sound pressure is measured over a period of time, it is possible to determine a number of statistical descriptions of time-varying sound and to account for the greater community sensitivity to night time sound levels. Terms commonly used in describing environmental sound include:

- $L_{10}$, $L_{50}$, and $L_{90}$: The A-weighted sound levels that are exceeded 10%, 50%, and 90% of the time, respectively. During the measurement period $L_{90}$ is generally taken as the background sound level.

- $L_{eq}$: Equivalent Sound Level: The average A-weighted sound pressure level which gives the same total energy as the varying sound level during the measurement period of time. Also referred to as $L_{A\,eq}$.

- $L_{dn}$: Day-Night Level: The average A-weighted sound level during a 24 hour day, obtained after addition of 10 dB to levels measured in the night between 10 p.m. and 7 a.m.

2.7.2 dB Math

From the comments above it can be seen that decibels do not add numerically as linear measures of other physical things do. Figure 8 shows how to add the decibels of two sound sources that are within 12 dB of each other.

![Graph for Addition of Decibels](image)

**Figure 8**: Addition of two sound levels.
For example, when adding two sound sources together, one being 9.5 dB(A) louder than the second, the resultant is approximately 10 dB(A) louder than the second source. It can be seen that when the sound from two sources more than 10 dB(A) apart are combined, the total sound pressure level in decibels is very close to the louder one, with little or no contribution from the softer sound.

2.8 Infrasound & Low Frequency Sound

Terminology: Low frequency pressure vibrations are typically categorized as low frequency sound when they can be heard near the bottom of human perception (10-200 Hz), and infrasound when they are below the common limit of human perception. Sound below 20 Hz is generally considered infrasound, even though there may be some human perception in that range. Because these ranges overlap in these ranges, it is important to understand how the terms are intended in a given context.

Infrasound is always present in the environment and stems from many sources including ambient air turbulence, ventilation units, waves on the seashore, distant explosions, traffic, aircraft, and other machinery. Infrasound propagates farther (i.e. with lower levels of dissipation) than higher frequencies.

Some characteristics of the human perception of infrasound and low frequency sound are:

- Low frequency sound and infrasound (2-100 Hz) are perceived as a mixture of auditory and tactile sensations.

- Lower frequencies must be of a higher magnitude (dB) to be perceived, e.g. the threshold of hearing at 10 Hz is around 100 dB; see figure 9.

- Tonality can not be perceived below around 18 Hz.

- Infrasound may not appear to be coming from a specific location, because of its long wavelengths.
Figure 9: Typical perception threshold of human ear for low frequency sound as a function of pressure

The primary human response to perceived infrasound is annoyance, with resulting secondary effects. Annoyance levels typically depend on other characteristics of the infrasound, including intensity, variations with time, such as impulses, loudest sound, periodicity, etc. Infrasound has three annoyance mechanisms:

- A feeling of static pressure.
- Periodic masking effects in medium and higher frequencies.
- Rattling of doors, windows, etc. from strong low frequency components.

Human effects vary by the intensity of the perceived infrasound, which can be grouped into these approximate ranges:

- 90 dB and below: No evidence of adverse effects.
- 115 dB: Fatigue, apathy, abdominal symptoms, hypertension in some humans.
- 120 dB: Approximate threshold of pain at 10 Hz.
• 120 – 130 dB and above: Exposure for 24 hours causes physiological damage.

There is no reliable evidence that infrasound below the perception threshold produces physiological or psychological effects.

2.9 Sound Propagation

In order to predict the sound pressure level at a distance from source with a known power level, one must determine how the sound waves propagate. In general, as sound propagates without obstruction from a point source, the sound pressure level decreases. The initial energy in the sound is distributed over a larger and larger area as the distance from the source increases. Thus, assuming spherical propagation, the same energy that is distributed over a square meter at a distance of one meter from a source is distributed over 10,000 m$^2$ at a distance of 100 meters away from the source. With spherical propagation, the sound pressure level is reduced by 6 dB per doubling of distance.

This simple model of spherical propagation must be modified in the presence of reflective surfaces and other disruptive effects. For example, if the source is on a perfectly flat and reflecting surface, then hemispherical spreading has to be assumed, which also leads to a 6 dB reduction per doubling of distance, but the sound level would be 3 dB higher at a given distance than with spherical spreading. The development of an accurate sound propagation model generally must include the following factors:

• Source characteristics (e.g., directivity, height, etc).

• Distance of the source from the observer.

• Air absorption, which depends on frequency.

• Ground effects (i.e., reflection and absorption of sound on the ground, dependent on source height, terrain cover, ground properties, frequency, etc).

• Blocking of sound by obstructions and uneven terrain.
• Weather effects (i.e., wind speed, change of wind speed or temperature with height). The prevailing wind direction can cause differences in sound pressure levels between upwind and downwind positions.

• Shape of the land; certain land forms can focus sound.

For estimation purposes, a simple model based on the more conservative assumption of hemispherical sound propagation over a reflective surface, including air absorption is often used:

\[ L_p = L_w - 10 \log_{10}(2\pi R^2) - \alpha R \]

Here \( L_p \) is the sound pressure level (dB) a distance \( R \) from a sound source radiating at a power level, \( L_w \), (dB) and \( \alpha \) is the frequency-dependent sound absorption coefficient. This equation can be used with either broadband sound power levels and a broadband estimate of the sound absorption coefficient (\( \alpha = 0.005 \) dB per meter) or more preferably in octave bands using octave band power and sound absorption data.

3. Turbulence fundamentals

3.1 Definition of turbulence

Everyday life gives us an intuitive knowledge of turbulence in fluids: the smoke of a cigarette or over a fire exhibits a disordered behaviour characteristic of the motion air which transports it.

Turbulence is rather a familiar notion, yet it is not easy to define in such a way as to cover the detailed characteristics comprehended in it and to make the definition agree with the modern view of it held by professionals in this field of applied science.

In 1937 Taylor and Von Karman gave the following definition: “Turbulence is an irregular motion which in general makes its appearance in fluids, gaseous or liquid, when they flow past solid surfaces or even when neighbouring streams of the same fluid flow past or over one another”. [Turbulence J.O. Hinze]. According to this definition, the flow has to satisfy the condition of irregularity.
Indeed, this irregularity is a very important feature. Because of irregularity, it is impossible to describe the motion in all details as a function of time and space coordinates. But, fortunately, turbulent motion is irregular in the sense that it is possible to describe it by laws of probability. It appears possible to indicate distinct average values of various quantities, such as velocity, pressure, temperature, etc., and this is very important. If turbulent motion were entirely irregular, it would be inaccessible to any mathematical treatment.

Turbulence can be generated by friction forces at fixed walls (flow through conduits, flow past bodies) or by the flow of layers of fluids with different velocities past or over one another. There is a distinct difference between the kinds of turbulence generated in the two ways.

In the case of real viscous fluids, viscosity effects will result in the conversion of kinetic energy of flow into heat. Thus turbulent flow, like all flow of such fluids, is dissipative in the nature. If there is no continuous external source of energy for the continuous generation of the turbulent motion, the motion will decay. Other effects of viscosity are to make the turbulence more homogeneous and to make it less dependent on direction. In the extreme case, the turbulence has quantitatively the same structure in all parts of the flow field; the turbulence is said to be homogeneous. The turbulence is called isotropic if its statistical features have no preference for any direction, so that perfect disorder reigns. In this case, no average shear stress can occur and, consequently, no velocity gradient of the mean velocity. This mean velocity, if it occurs, is constant throughout the field.

In all other cases where the mean velocity shows a gradient, the turbulence will be nonisotropic, or anisotropic.

Furthermore, if different turbulent motions are compared in each of which a distinct pattern can be discerned, it is possible to observe differences, for instance, in the size of the patterns. This means that, to describe a turbulent motion quantitatively, it is necessary to introduce the notion of scale of turbulence: a certain scale in time and a certain scale in space. The magnitude of these scales will be determined by the dimensions of and the velocities within the apparatus in which the turbulent flow occurs. For turbulent flow through a
pipe, for instance, one may expect a time scale of the order of magnitude of the ratio between pipe diameter and flow velocity and a space scale of the order of magnitude of the diameter of the pipe.

It is apparent that it is insufficient to characterize a turbulent motion by its scale alone, since to do so does tell anything about the violence of the motion. One cannot take the average value of the velocity as a measure of this violence, because the violence of fluctuations with respect to this average velocity is just what one wants to know.

Whether a flow is laminar or turbulent depends of the relative importance of fluid friction (viscosity) and flow inertia. The ratio of inertial to viscous forces is the Reynolds number. Given the characteristic velocity scale, $U$, and length scale, $L$, for a system, the Reynolds number is

$$Re = UL/\nu$$

where $\nu$ is the kinematic viscosity of the fluid.

The characteristic length-scale for a channel of width $w$ and depth $h$ is the hydraulic radius

$$Rh = wh/P$$

where $P$ is the wetted perimeter. For an open channel $P = (2h + w)$ and for a closed conduit $P = 2(h + w)$. As a general rule, open channel flow is laminar if the Reynolds number defined by the hydraulic radius

$$Re = URh/\nu$$

is less than 500. As the Reynolds number increases above this limit burst of turbulent appear intermittently in the flow. If the conduit boundary is rough, the transition to fully turbulent flow can occur at lower Reynolds numbers. Alternatively, laminar conditions can persist to higher Reynolds numbers if the conduit is smooth and inlet conditions are carefully designed.
The straight, parallel black lines are streamlines, which are everywhere parallel to the mean flow. In laminar flow the fluid particles follow the streamlines exactly, as shown by the linear dye trace in the laminar region. In turbulent flow eddies of many sizes are superimposed onto the mean flow. When dye enters the turbulent region it traces a path dictated by both the mean flow (streamlines) and the eddies. Larger eddies carry the dye laterally across streamlines. Smaller eddies create smaller scale stirring that causes the dye filament to spread (diffuse).

3.2 Characterizing turbulence

Turbulent eddies create fluctuations in velocity. As an example, the longitudinal ($u$) and vertical ($v$) velocity measured at point A in figure 1 are shown below. Both velocities varying in time due to turbulent fluctuations. If the flow were steady and laminar then $u = \bar{u}$ and $v = \bar{v}$ for all time $t$, where the over-bar denotes a time average. For turbulent flow, however, the velocity record includes both a mean and a turbulent component. We decompose the flow as follows.

**Figure 10** Tracer transport in laminar and turbulent flow.
\[ u(t) = \bar{u} + u'(t) \]
\[ v(t) = \bar{v} + v'(t) \]

This is commonly called a Reynolds’ decomposition.

Figure 11 Velocity recorded at Point A in Figure 10.

Because the turbulent motions associated with the eddies are approximately random, we can characterize them using statistical concepts. In theory the velocity record is continuous and the mean can be evaluated through integration. However, in practice the measured velocity records are a series of discrete points, \( u_i \). Below an overbar is used to denote a time average over the time interval \( t \) to \( t+T \), where \( T \) is much longer than any turbulence time scale, but much shorter than the time-scale for mean flow unsteadiness, for example wave or tidal fluctuation.

Mean velocity:
\[
\bar{u} = \int_t^{t+T} u(t) dt = \frac{1}{N} \sum_{i=1}^N u_i
\]
continuous record  
discrete, equi-spaced pts.

Turbulent Fluctuation:
\[
u'(t) = u(t) - \bar{u} : \text{continuous record} \]
\[
u'_i = u_i - \bar{u} : \text{discrete points}
\]

Turbulence Strength:
\[
u_{rms} = \sqrt{\nu'(t)^2} = \sqrt{\frac{1}{N} \sum_{i=1}^N \nu_i'^2}
\]
continuous record  
discrete, equi-spaced pts.
Turbulence intensity: \[ \frac{u_{rms}}{\bar{u}} \]

The subscript ‘\( \text{rms} \)’ stands for root-mean-square. We should recognize the definition of \( u_{rms} \) given as the standard deviation of the set of ‘random’ velocity fluctuations, \( u_i \).

Similar definitions apply to the lateral and vertical velocities, \( v(t) \) and \( w(t) \). A larger \( u_{rms} \) indicates a higher level turbulence. In the figure below, both records have the same mean velocity, but the record on the left has a higher level of turbulence.

![Figure 12 Turbulence level. (Velocity versus time).](image)

3.3 Mean Velocity Profiles - Turbulent Boundary Layers

Near a solid boundary the flow has a distinct structure, called a boundary layer. The most important aspect of a boundary layer is that the velocity of the fluid goes to zero at the boundary. This is called the "no-slip" condition, that is the fluid velocity matches (has no slip relative to) the boundary velocity. This arises because of viscosity, \( \nu \), which is a fluid's resistance to flowing, that is fluid friction. The fluid literally sticks to the boundary. The higher its viscosity, the more a fluid resists flowing. Honey, for example, has a higher viscosity than water. The kinematic viscosity of water is \( \nu = 0.01 \text{ cm}^2/\text{s} \) \( (T = 20 \text{ °C}) \).
The figure below depicts a typical mean velocity profile, $\bar{u}(y)$, above a solid boundary. The vertical axis ($y$) denotes the distance above the boundary. The fluid velocity at the boundary ($y = 0$) is zero. At some distance above the boundary the velocity reaches a constant value, $U_\infty$, called the free stream velocity. Between the bed and the free stream the velocity varies over the vertical coordinate. The spatial variation of velocity is called shear. The region of velocity shear near a boundary is called the momentum boundary layer. The height of the boundary layer, $\delta$, is typically defined as the distance above the bed at which $\bar{u} = 0.99 \, U_\infty$.

![Mean velocity profile above a solid boundary](image)

**Figure 13** Mean velocity profile above a solid boundary

3.4 Shear produces turbulence

Turbulence is an instability generated by shear. The stronger the shear, the stronger the turbulence. This is evident in profiles of turbulence strength ($u_{rms}$) within a boundary layer (see figure below). The shear in the boundary layer decreases moving away from the bed, $\partial(\bar{u})/\partial y < 0$, and as a result the turbulence intensity also decreases. Very close to the bed, however, the turbulence intensity is diminished, reaching zero at the bed ($y=0$). This is because the no-slip condition applies to the turbulent velocities as well as to the mean velocity. Thus, in a thin region very close to the bed, no turbulence is present.
This region is called the laminar sub-layer, $\delta_s$. Note that the profiles shown below are normalized by the free-stream velocity, $U_\infty$. This is done to emphasize the fact that the mean and turbulent profiles within a boundary layer are self-similar with respect to the free stream velocity, $U_\infty$. This means that both profiles have the same shape regardless of the absolute magnitude of the external flow, $U_\infty$. Because of this self-similarity, we have the general rule of thumb that the turbulence level increases with the free stream velocity, $u_{rms} \sim U_\infty$, where the symbol $\sim$ is read “scales on”. In addition, as the turbulence level increases, the thickness of the laminar sub-layer decreases. In general, $\delta_s \sim (1/ U_\infty)$.

![Figure 14: Turbulence strength within a boundary layer](image)

As a second example, consider the profiles of mean and turbulent velocity measured across a jet. The profiles are self-similar when normalized by the centerline velocity, $U_{CL}$. The maximum turbulence level occurs at the positions of maximum shear. At the centerline the shear is zero ($\partial\bar{u}/\partial y = 0$), and the turbulence strength is diminished.
3.4.1 Friction (Shear) velocity, $U^*$

Physically, we know that the turbulence level scales on the shear, $u_{rms} \sim \partial \bar{u} / \partial y$. But this scale relationship is not dimensionally consistent, so we introduce a velocity scale to represent the shear strength. This velocity scale, $u^*$, is called the shear velocity, or the friction velocity, and it characterizes the shear at the boundary. The definition of $u^*$ is based on the bed stress, $\tau_{bed}$

$$\tau_{bed} = \rho u^*$$

where $\tau_{bed}$ is defined by the stress-strain relation

$$\tau_{bed} = \left[ \rho \nu \frac{\partial \bar{u}}{\partial y} \right]_{y=0}$$

Thus

$$u^* = \sqrt{\tau / \rho} = \sqrt{\left[ \nu \frac{\partial \bar{u}}{\partial y} \right]_{y=0}}$$

The shear velocity characterizes the turbulence strength and laminar sub-layer thickness.

$$u_{rms} \sim u^*$$

$$\delta_s = 5\nu / u^*$$
3.5 Turbulent Velocity Profile: The Logarithmic Velocity Profile

The shape of the velocity profile within a turbulent boundary layer is well-established by theory and experiment. The profile has specific characteristics very close to the bed where viscosity controls the vertical transport of momentum, and different characteristics farther from the bed where turbulence controls the vertical transport of momentum. The region closest to the boundary is called the Laminar Sub-Layer, because within the region turbulence is suppressed by viscosity. In this region the velocity profile is defined by the stress-relation given in (2). We substitute the definition given in (1) into (2) and use the approximation $\partial u / \partial y \approx u / y$ to solve for the velocity profile.

Laminar Sub-Layer [$y < \delta_s = 5 \nu / u^*$]:

$$\bar{u}(y) = (u^*)^2 \frac{y}{\nu}$$  \hspace{1cm} (6)

Above the Laminar Sub-Layer ($y > \delta_s$) the velocity profile is logarithmic. The profile shape depends both on the bed stress (through $u^*$) as well as on the bed texture, described by the characteristics roughness, $y_o$.

Logarithmic Layer [$y > \delta_s$]:

$$\bar{u}(y) = \frac{2.3 u^*}{\kappa} \log_{10} \frac{y}{y_o}$$  \hspace{1cm} (7)

where $\kappa$ is is an empirical constant, known as von Karman’s constant and $\kappa = 0.4$.

Nikuradse studied the influence of boundary texture on velocity profile shape. He glued uniform sand grains of diameter $\varepsilon$, to the bed of a flume and measured the velocity profile over the bed at different flow speeds. He found two different behaviours defined by the roughness Reynolds number, $\varepsilon u^*/\nu$. For conditions with $\varepsilon u^*/\nu < 5$, $y_o = \nu/9u^*$, that is, the characteristic roughness is NOT a function of the real roughness scale. This means that the velocity profile shape, through $y_o$, is not a function of the real roughness scale, or, simply, the logarithmic portion of the velocity profile is independent of the surface roughness under these conditions. To understand why, recall from (5) that the thickness of the laminar sub-layer, $\delta_s = 5 \nu / u^*$. So, Nikuradse’s findings simply
say that when the surface texture is smaller than the laminar sub-layer \( (\varepsilon < 5 \nu/u^*) \), then the flow above the laminar sub-layer does not feel the surface texture. We call this regime Smooth Turbulent Flow. When the roughness becomes larger than the laminar sub-layer, specifically \( \varepsilon > (70 \text{ to } 100)\nu / u^* = 14 \text{ to } 20 \delta_s \), then the flow above the laminar sub-layer does feel the surface texture. Under these conditions \( y_o = \varepsilon/30 \), that is, the characteristic roughness is a function of the real roughness scale, and the logarithmic profile is altered, through \( y_o \), by the surface texture. We call this regime Rough Turbulent Flow.

![Figure 15 ε and δ in smooth and rough turbulent flow](image)

**3.5.1 Example of fitting a Logarithmic Profile**

An example of a mean velocity profile is graphed in two forms on the following page, using logarithmic and linear axes. The linear axes reveal the more familiar boundary layer profile. The logarithmic portion of the profile appears linear on the logarithmic axes. By fitting the logarithmic portion of the profile, we can estimate the characteristic roughness, \( y_o \), and the friction velocity, \( u^* \). The red line is the log-linear fit to the velocity profile. We ignore the two points closest to the bed, as these do not follow the same log-linear trend as the rest of the profile, and we suspect (and will later check) that they lie within the laminar sub-layer. From (7) the slope of the red, fitted line gives us an estimate for \( u^* \). Specifically, we select two points on the red line, \( y_1 = 6 \text{ cm} \) and \( y_2 = 0.05 \text{ cm} \), with velocity \( u_1 = 1.2 \text{ cms}^{-1} \) and \( u_2 = 0 \text{ cms}^{-1} \), respectively. Then,
\[ u_\ast = \frac{\kappa (u_2 - u_1)}{2.3 \log_{10}(y_1) - \log_{10}(y_2)} = \frac{\kappa (1.2 - 0 \text{ cm/s})}{2.3 \log_{10}(6/0.05)} = 0.1 \text{ cm/s} \]

The characteristic roughness, \( y_o \), is the \( y \)-intercept of the red line. That is, from (7), \( u = 0 \) when \( y = y_o \). From the graph, \( y_o = 0.05 \text{ cm} \). Since \( y_o > \nu/9u_\ast = 0.01 \), the flow is not Smooth Turbulent. From (5) the laminar sub-layer thickness is 0.5 cm, which confirms that the two points closest to the boundary lie inside the laminar sub-layer. Finally, if we assume the flow is Fully Rough Turbulent, \( \varepsilon = 30y_o = 1.5 \text{ cm} \). Then the roughness Reynolds’ number is \( \varepsilon u_\ast/\nu = 15 \). Because \( \varepsilon u_\ast/\nu < 70 \), we conclude that the flow is not Rough Turbulent, but in a transition between Smooth and Rough.

![Figure 16 Mean velocity profile (logarithmic axes)](image)
3.6 Turbulent transport in the Equation of Mass Conservation

The presence of turbulence creates fluctuations in concentration. As we did with the velocity field above, we decompose the concentration into a temporal mean and turbulent fluctuations around that mean. As above, the over-bar indicates an average over time-scale $T$, which is long compared to the turbulent fluctuations.

$$C(t) = \bar{C} + C'(t)$$

For simplicity we start with a one-dimensional version of the equation of mass conservation (transport equation),

$$\frac{\partial C}{\partial t} + \frac{\partial (uC)}{\partial x} = \frac{D}{\partial x} \frac{\partial C}{\partial x}$$

into which we substitute the decomposition of velocity and concentration.
\[
\frac{\partial (\bar{C} + C')}{\partial t} + \frac{\partial ((\bar{u} + u')(\bar{C} + C'))}{\partial x} = \frac{\partial}{\partial x} \bar{D}_x \frac{\partial (\bar{C} + C')}{\partial x}
\]

Now, we time average each term. By definition, \(\bar{a}' = 0\), and \(\bar{a} = \bar{a}\).

\[
\frac{\partial \bar{C}}{\partial t} + \frac{\partial (\bar{u}\bar{C} + u'C')}{\partial x} = \frac{\partial}{\partial x} \bar{D}_x \frac{\partial \bar{C}}{\partial x}
\]

The term \(u'C'\) represents the net mass flux due to turbulent advection. If we could fully calculate the turbulence field, we could calculate the turbulent flux. Unfortunately this is quite complex and computationally intensive, and for many flows quite prohibitive. Alternatively we can devise a model for the turbulent flux in terms of the mean velocity and concentration, which are easily known. A simple mixing-length model is proposed below. It assumes the turbulent motions can be characterized by the length-scale of the eddies.

**3.7 Mixing-Length Model for turbulent flux**

Below is a long narrow tube with linear concentration gradient \(\partial C/\partial x < 0\). There is no mean current in the tube, \(\bar{u} = 0\). Consider the transport achieved by a single eddy with length-scale \(l_x\). At the top of the eddy \(u' > 0\), and the eddy carries forward fluid of higher concentration, such that a probe positioned at the dashed line would momentarily record a concentration greater than the local mean when this eddy is present. That is, at the position of the dashed line, \(C' > 0\) where \(u' > 0\). Similarly, for this eddy, where \(u' < 0\) then \(C' < 0\).
The magnitude of the concentration fluctuations will be of the scale, \( |c'| \sim lx \partial C/\partial x \). The sign of the concentration fluctuation depends on both the sign of the concentration gradient and the sign of the velocity fluctuation. Again we consider the picture above in which \( \partial C/\partial x \) is negative. The part of the eddy for which \( u' \) is also negative produces a negative \( c' \). The part of the eddy for which \( u' \) is positive produces a positive \( C' \). In a region with positive gradient, \( \partial C/\partial x > 0 \), \( u' \) positive produces \( C' \) negative, and \( u' \) negative produces \( C' \) positive.

In general, when \( \partial C/\partial x \) and \( u' \) have the same sign, \( C' < 0 \), and when \( \partial C/\partial x \) have opposite sign, \( C' > 0 \). So, the sign of \( C' \) is -sign \( (u' \partial C/\partial x) \). Using this definition for the sign of \( C' \) we can now write the turbulent advection generated by an isolated velocity fluctuation \( u' \).

The flux is proportional to the mean concentration gradient, and is counter gradient. Following this analogy, we define a turbulent diffusion coefficient, or turbulent diffusivity,

\[
Dt,x \sim u' lx
\]
Such that the turbulent flux can explicitly modelled as an additional diffusion term,

\[ u'C' = -D_{t,x} \frac{\partial C}{\partial x} \]

Simply stated, this model shows that the turbulent flux depends on the strength of the turbulence (\( u' \)) and the scale of the turbulence (\( lx \)). From our previous discussion, the strength of turbulence is characterized by the friction velocity, i.e. \( u' \sim \nu^* \). In fact many length-scales of turbulence co-exist in a turbulent flow, so we must select the length scale that is most important to the turbulent flux. In general, this will be the largest length-scale in the system, because the effective diffusivity increases with eddy scale. Thus, the dominant length-scale of the turbulent transport will depend on the geometric constraints of the domain, which dictates the largest eddy scale in the domain.

\[
\frac{\partial \overline{C}}{\partial t} + \frac{\partial (u\overline{C})}{\partial x} = \frac{\partial}{\partial x} (D_x + D_{t,x}) \frac{\partial \overline{C}}{\partial x}
\]

Thus, we have shown that the effect of turbulence on the transport equation can be modeled simply by increasing the coefficient of diffusion by an amount dictated by the strength and intensity of the turbulence. In general the turbulent diffusivity, \( D_{t,x} \) is much greater than its molecular counterpart, such that the latter is simply ignored. Now, the solutions already devised for the transport equation can be applied in turbulent flow, but with the molecular diffusivity replaced by its turbulent cousin.

Finally, through similar reasoning on can quickly show that the turbulent diffusivity in the vertical and lateral dimension will scale as,

\[ D_{t,y} \sim v'ly \quad \quad D_{t,z} \sim w'lz \]
Because turbulence is often anisotropic in both length-scale ($l_x \neq l_y \neq l_z$) and intensity ($u' \neq v' \neq w'$), we expect that the turbulent diffusivity will also be anisotropic ($D_{t,x} \neq D_{t,y} \neq D_{t,z}$).

3.8 Measurement of turbulent flows

During the experimental investigations of fluid flow a great number of methods, techniques, and instruments have been developed and used; so today a choice among them is available, one being more suitable for a particular kind of measurements than the other. Most of these methods and instruments have been developed and are used for measuring velocities in flows that are either nonturbulent or assumed to be nonturbulent; in fact only a few are suitable for making reliable measurements in turbulent flows or, more specifically, for measuring the turbulence itself.

The main difficulties in measuring turbulence are caused by the fact that that turbulence is a random, fluctuating flow and three-dimensional. Moreover, the high frequencies of the fluctuations occurring in the turbulent flows in which normally it is interesting make it very difficult for a measuring instrument to satisfy in every respect the basic requirement that is made of such an apparatus, namely that recordings of the quantity to be measured must be as free as possible from distortion.

In measuring turbulence flows we have to distinguish between measurements of the mean flow and measurement of the turbulence proper. The problems connected with these two types of measurement are to a certain extend related, yet the requirements which the methods and instruments used must fulfill are different. For instance, the result of a measurement of the mean velocity at a given point is more or less affected by the turbulence present in the flow, and it is necessary to know what correction must be made in the readings of the measuring instrument, but in measuring the turbulence itself it cannot be tolerated any influence of the mean velocity that produces an error in the turbulence velocities recorded.
Moreover, most of the current methods for measuring mean values are affected by turbulence; so corrections have to be applied to the experimental data. These corrections can be made only if the turbulence values are known; so it is necessary to measure these turbulence quantities in addition to the mean values.

It is possible to divide the various methods, techniques, and instruments into two groups.

In the first group, use is made of a tracer or other indicator which is introduced into the fluid to make the flow pattern visible (photographic recording) or observable by a suitable detecting apparatus outside the field of flow.

In the second group, a detecting element is introduced into the flowing fluid, and the turbulence quantities are measured by the changes of a mechanical, physical, or chemical nature that occur in this element.

When the methods of the first group for measuring turbulence quantities are applied, immediately difficulties associated with very rapid changes according to time and place appear, so that practically instantaneous recordings are necessary, frequently at very short intervals. Moreover, the three-dimensionality of the turbulent motions does not make the interpretation of such recordings any simpler, on the contrary.

As regards the second group, there are a number of requirements that must be satisfied by the detecting element and the rest of measuring apparatus before turbulence can be measured reliably:

1. The detecting element introduced into the flowing field must be so small that it causes only the minimum admissible disturbance of the flow pattern.
2. The instantaneous velocity distribution must be uniform in the region occupied by the element. This means that the detecting element must be smaller than the dimensions of the micro scale of turbulence. If we confine our measurements to flows of low or
moderate velocities, the size of the detecting element should not exceed 1 mm.

3. The inertia of the instrument must be low, so that response to even the most rapid fluctuations is practically instantaneous. For flow velocities that are not too high, frequencies up to 5000 sec\(^{-1}\) may be expected.

4. The instrument must be sufficiently sensitive to record small differences in the fluctuations; these differences are often only a few per cent of the mean value.

5. The instrument must be stable, so that no noticeable change occurs in the calibration parameters during at least one test run.

6. The instrument must be sufficiently strong and sufficiently rigid to exclude vibrations or motions caused by the turbulent flow.

Accordingly, the normal Pitot, or total-head, tube, which is employed so successfully in the measurement of nonturbulent flows, is completely ruled out for measuring turbulence (except perhaps turbulence on a very large scale and of a very low frequencies such as might occur, for instance, tiny tube is used in conjunction with a sensitive pressure transducer.

There is one instrument whose development and application for measuring turbulent flow have far outstripped those of other instruments up to now, namely, the hot-wire anemometer. Its popularity for making turbulence measurements will be easily understood if it is realized that this is the only instrument that reasonably satisfies all the above mentioned requirements, although, of course, it has its limitations.

These involve the nonlinear character of the heat transfer with respect to velocity and temperature. Both the nonlinear temperature effect and the effect of the curvature in the velocity-response curve can be reduced considerably by applying the constant-temperature method, especially when a linearization circuit is added, although the difficulty is then shifted to the building of a reliable electronic circuit.

Another limitation is set by the “resolution power” in the direction of the wire, owing to its finite length. The resolution power in the flow direction is
limited by the time constant of the hot wire. We may introduce a resolution length

\[ l_{res} = \frac{U}{2n_{max}} \]

where \( U \) is the flow velocity and \( n_{max} \) is the maximum frequency of the amplifier at which is still no appreciable loss of response. This resolution length must be small compared with the size of the micro eddies. Since \( n_{max} \) is inversely proportional to the time constant of the hot wire, \( l_{res} \) is directly proportional to it.

At very high flow velocities (say, \( U > 100 \text{ m/sec} \)) extremely small eddies may occur. So that the resolution length may remain small compared with these eddies, small values of the time constant must be used; it is desirable, therefore, to use very fine wires. On the other hand, because of the strength required of the wire in such high-velocity flow, the wire should not be too thin. Moreover, the supports and their aerodynamic interference with the wire may causes disturbances in the flow pattern comparable in size to the smallest eddies.

Another difficulty, which becomes more pronounced in the case of thin wires, is that the hot wire is sensitive to deposition and impact values of small particles (dust); consequently, changes may occur in the values of calibration parameters. The wires must then be cleaned and recalibrated. This cleaning and recalibration may become quite a nuisance when the intervals become very short.
4. General description of test performed

4.1 Equipment required

To carry out the experiment, the equipment required was a fan, two microphones, an accelerometer, a thermal anemometry, an analog oscilloscope, a microphone conditioning amplifier, a data acquisition system, two frequency inverters, and a computer. How they work and the main characteristics are described below, putting special emphasis on the thermal anemometry.

4.1.1 Fan

The fan used in the test is a rotor from a real machine with 8 flat blades. The second rotor has 8 cylinders (height 12mm, diameter 3mm). The distance between the cylinders’ exit plane and the fan’s inlet plane is 68mm.

![Fan used in the test](image)
4.1.2 Microphones

To carry out the experiment, it has been necessary to use 2 *Brüel & Kjær* common microphones. One of them (Channel 1) was at 20 cm from the fan and the other one (Channel 2) was at 5 cm.

![Microphone Brüel & Kjær](image)

*Figure 20 Microphone Brüel & Kjær*

4.1.3 Accelerometer

The accelerometer used in the experiment was a *Brüel & Kjær*.

![Accelerometer Brüel & Kjær](image)

*Figure 21 Accelerometer Brüel & Kjær*
4.1.4 Analog oscilloscope

An oscilloscope is a device which graphically displays a repeated waveform in a circuit, such as the signal for a sound. Oscilloscopes display the voltage or amperage of a circuit as a line on a screen. It is possible to study the waveform, to measure how much time passes between pulses in a circuit, detect the frequency of a signal, or do other useful analyses.

In this case, the device used was a Hameg HM 303-6, 35 MHz Analog Oscilloscope. The main characteristics of this oscilloscope are:

- 2 Channels with deflection coefficients of 1 mV/cm – 20 V/cm.
- Time Base: 0.2 s/cm – 100 ns/cm, with X Magnification to 10 ns/cm.
- Low Noise Measuring Amplifiers with high pulse fidelity and minimum overshoot.
- Triggering from 0 to 50 MHz from 5 mm signal level (up to 100 MHz from 8 mm).
- Up to 500,000 signal displays per second in optimum analog quality.
- Yt, XY and component-test modes.

Figure 22 Hameg HM 303-6, 35 MHz Analog Oscilloscope
4.1.5 Microphone conditioning amplifier

The microphone conditioning amplifier used in the experiment was a *Nexus Brüel & Kjær*.

![Microphone conditioning amplifier Nexus Brüel & Kjær.](image)

**Figure 23** Microphone conditioning amplifier Nexus Brüel & Kjær.

4.1.6 Data acquisition system

WaveBooks are high-speed portable data acquisition devices that can be used in a variety of applications, such as testing engine strain, multi-channel acoustics, mechanical integrity, and vibration/shock/strain.

The data acquisition card utilized was an *I0tech WaveBook 516E*.

![I0tech WaveBook 516E](image)

**Figure 24** I0tech WaveBook 516E
4.1.7 Frequency inverters

A frequency inverter is a device for controlling the speed of a rotational or linear alternating current (AC) electric motor by controlling the frequency of the electrical power supplied to the motor. Frequency inverters are used in a wide number of applications to control pumps, fans, hoists, conveyors, and other machinery.

In the test, two different inverters have been used to control the fan and the cylinders rotational speed.

![Image of Omron V1000]

**Figure 25** Omron V1000

To control the rotational speed of cylinders it has been used a *Omron V1000 Compact Vector Control Drive*. The common specifications are in next table.
<table>
<thead>
<tr>
<th>Model number</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>VZ-43</td>
<td></td>
</tr>
</tbody>
</table>

**Control functions**
- **Control methods**: Sine wave PWM (V/F control, sensorless current vector control)
- **Output frequency range**: 0.1 - 400 Hz
- **Frequency tolerance**: ±0.01% (0°C - 10°C), ±0.1% (25°C - 10°C)
- **Resolution of frequency setting value**: Digital set value: ±0.1 Hz (<100 Hz, ±0.1 Hz (<100 Hz), ±0.1 Hz (>100 Hz)
- **Resolution of output frequency**: 0.01 Hz
- **Overload capability**: Heavy duty use: 150% rated output current for one minute, Normal duty use: 120% rated output current for one minute
- **Frequency set value**: 0.1 V/20 Hz, 0.25 mA (255), 0.26 mA (285)
- **Braking torque**: Short-term average deceleration torque: 16% (up to 1.5 kW), 100% (for 1.5 kW, 50% (for 2.2 kW), 20% (for bigger size)
- **Continuous regenerative torque**: Approx. 20% (126% with optional braking resistor, 10% LED, 10 s, braking transistor built-in)

**V/F Characteristics**
- **Pulse train input, frequency setting value (selectable)**
- **Functionality**
  - Inputs signals: Seven of the following input signals are selectable: Forward/reverse run (3-wire sequence), coast reset, external fault (NO/NC contact input), multi-step speed operation, log command, acceleration time setting, external variable motor, speed search command, UP/DOWN command, acceleration/deceleration, command, LOCAL REMOTE selection, communication control terminal selection, emergency stop, emergency stop alarm, safety test
  - Outputs signals: Following output signals are selectable: NO/NC contact output, 2 photo-coupler outputs: Fault running, zero speed, speed limit, frequency detection output (frequency < or = set value), during overvoltage detection, motor error, during base block operation mode, inverter run ready, during fault reset, during undervoltage detection, reverse running, during speed search, data output through communication
  - Standard functions: Open-loop vector control, full-range automatic torque boost, slip compensation, 17-step speed operation (max.), restart after momentary power loss, DC injection braking current at stoppage (50% of inverter rated current), 0.5 sec. or less, frequency reference bias gain, MEMOBUS communications (RS-485/422, 115K bps), fault reset, speed search, frequency upper lower limit setting, overvoltage detection, frequency display, acceleration (time switch), acceleration prohibited, 2-speed acceleration, PID control, energy-saving control, constant output
  - Analogue inputs: 2 analogue inputs, 0.1 V, 10 mA, 10 mA, 0.2 mA
  - Braking/acceleration times: 0.01 - 5000 s
  - Display: Optionally frequency, current, or set value, timer and status LED
  - Motor overload protection: Electronic thermal overload relay
  - Instantaneous overcurrent: Motor coast to a stop at approx. 250% of inverter rated current
  - Overload: Heavy Duty: Motor coast to a stop after 1 minute at 150% of inverter rated output current, Normal Duty: Motor coast to a stop after 1 minute at 120% of inverter rated output current
  - Undervoltage: Motor coast to a stop if DC bus voltage exceed 110 V (double for 400 V class)
  - Momentary power loss: Following items are selectable: not provided (stop if power loss is 15 ms or longer), continuous operation if power loss is approx. 0.5 sec. or shorter, continuous operation
  - Cooling fan: Yes, provided for 200V, 0.75 kW (3-phase), 0.15 kW (2-phase), 0.06 kW (1-phase)
  - Ambient conditions: Temperature: 5% RH or less (without condensation), Storage temperature: -20°C to +60°C (short-term temperature during transportation)
  - Installation height: Max. 1500 m

**Figure 26** Omron V1000 specifications

Otherwise, to control the rotational speed of the fan it has been used a **Hitachi SJ200-015HFEF**.
The class specifications are described below:

<table>
<thead>
<tr>
<th>Item</th>
<th>400V Class Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>SJ200 inverters, 400V models</td>
<td>EU version</td>
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</tr>
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<td>Applicable motor size *2</td>
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</tr>
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<td></td>
<td>HP</td>
</tr>
<tr>
<td>Rated capacity (460V) kVA</td>
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<td>Rated input voltage *6</td>
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</tr>
<tr>
<td>Integrated EMC filter</td>
<td>EU version</td>
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<tr>
<td>USA version</td>
<td></td>
</tr>
<tr>
<td>Rated input current (A)</td>
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<td>Rated output voltage *3</td>
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<td>Rated output current (A)</td>
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<td>Starting torque *7</td>
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<td>Dynamic braking approx. % torque, short time stop *8</td>
<td>without resistor, from 50/60 Hz</td>
</tr>
<tr>
<td></td>
<td>with resistor</td>
</tr>
<tr>
<td>DC braking</td>
<td>Variable operating frequency, time, and braking force</td>
</tr>
<tr>
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<td>EU version (-HFEF)</td>
</tr>
<tr>
<td></td>
<td>lb</td>
</tr>
<tr>
<td></td>
<td>US version (-HFU)</td>
</tr>
<tr>
<td></td>
<td>lb</td>
</tr>
</tbody>
</table>

*Figure 27* Hitachi SJ200-015HFEF

*Figure 28* Hitachi SJ200-015HFEF specifications
4.1.8 Thermal anemometry

Thermal anemometers measure fluid velocity by sensing the changes in heat transfer from a small, electrically-heated element exposed to the fluid. In the “constant temperature anemometer,” the cooling effect caused by the flow passing the element is balanced by the electrical current to the element, so the element is held at a constant temperature. The change in current due to a change in flow velocity shows up as a voltage at the anemometer output.

A key feature of the thermal anemometer is its ability to measure very rapid changes in velocity. This is accomplished by coupling a very fine sensing element (typically a wire four to six microns in diameter or a platinum thin film deposited on a quartz substrate) with a fast feedback circuit which compensates for the drop in the natural sensor response. Time response to flow fluctuations as short as a few microseconds can be achieved. For this reason, the thermal anemometer has become a standard tool for researchers studying turbulence. The small sensor size, normally only a millimetre in length, also makes the technique valuable in applications where access is difficult or larger sensors obstruct the flow.

Since the actual measurement is of heat transfer between the sensor and its environment, the thermal anemometer will respond to changes in parameters other than velocity, such as temperature, pressure, and fluid composition. While this adds to versatility, it also means that when more than one parameter is changing, special techniques must be used to extract velocity. Modern systems will automatically correct the velocity reading for temperature changes.

When selecting a thermal anemometry probe, the user must choose between film and wire sensors. The choice is based on the fluid characteristics, the velocity range, the number of velocity components, contamination in the flow, and access to the flow.

The traditional sensor for research thermal anemometry has been a fine wire. For very low turbulence intensities, the wire sensor is still superior and the smaller the wire, the better the results. For those applications that require a wire sensor, the 4 micrometer-diameter platinum-coated tungsten wire is almost a standard for measurements at normal room temperatures and below. Tungsten
is very strong and has a high temperature coefficient of resistance. It will, however, deteriorate at high temperatures in oxidizing atmospheres (such as air). Platinum wires, though weaker, can also be made very small and will withstand high temperature in an oxidizing atmosphere. If more strength is needed at high temperatures, an alloy such as platinum iridium should be selected.

The rigidity and strength of cylindrical film sensors, relative to wire sensors, make them the preferred choice in a wide range of thermal anemometry applications. Rigidity is especially important for multi-sensor measurements where the algorithms used for data reduction assume a straight sensor. Also, film sensors are less susceptible to damage or coating by particles in the flow than are wire sensors.

In this case, the thermal anemometry used has been a *TSI IFA-100 Constant-Temperature Anemometry*.

![Figure 29 TSI IFA-100 Constant-Temperature Anemometry](image)

### 4.1.9 Hot wire probe

The hot wire probe used at test was a *Dantec 1 wire* with the next specifications:
<table>
<thead>
<tr>
<th>DISA probe type 55</th>
<th>R13</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensor resistance at 20 °C</td>
<td>$R_{20} = 6.72 , \Omega$</td>
</tr>
<tr>
<td>Leads resistance</td>
<td>$R_c = 0.6 , \Omega$</td>
</tr>
<tr>
<td>Sensor TCR</td>
<td>$\alpha_{20} = 0.42 , % / , \text{C}$</td>
</tr>
</tbody>
</table>

**Figure 30** Hot wire probe specifications

Measure total probe resistance, $R_{\text{tot}}$ at ambient temperature, $T_o$, select $T_{\text{sensor}} < 120 \, ^\circ\text{C}$ and calculate operating resistance, $R$

$$R = R_{\text{tot}} + \alpha_{20} \cdot R_{20} \cdot (T_{\text{sensor}} - T_o)$$

**Figure 31** CTA Anemometry diagram [Source http://www.dantecdynamics.com]
4.2 Description of the test performed

The test was performed with all the equipment described above and with a computer. Throughout the test measurements were taken four channels. The first channel signal was one of the two microphones. The second channel was the signal of the other microphone. The third channel recorded the accelerometer signal. Finally, the fourth channel recorded the thermal anemometry signal.

Notably, the units of each of these channels were in Volts. Each measurement was 30 seconds and the sampling frequency \( f_s \) was 50 kHz, so each simulation generated 1.500.000 of data for each canal.

As mentioned before, the fan had blades. However, the test consisted in making several measurements varying the rotor velocity and the rotational speed of the cylinders, through inverters. That's why the test was divided into two parts.

In the first part of the test, the cylinders stood. The only variable velocity was the speed of the rotor, controlled by the inverter Hitachi SJ-200. The first simulation was performed using a rotor speed of 25 Hz. Over 30 seconds of simulation data were stored on the computer. The following simulation was performed using a rotor speed of 25.2 Hz. That is, each simulation was increased 0.2 Hz. This same step was repeated to reach 35 Hz.

In the second part of the test the cylinders and rotor were functioning. The speed of the cylinders was set at 30 Hz and remained constant over this part. Instead, the rotor speed in the first simulation was also 25 Hz, but in this case was 0.1 Hz increasing each step to reach 35 Hz.

Therefore, in the first stage, 51 simulations were performed and in the second we had 101. In the following section we can see the data analysis.
5. Data Analysis

This section shows how the data provided were collected in the test and how they have been processed to achieve the final result.

The data from each simulation were recorded in a text file. As mentioned before, four channels were available and the units of each of these channels were Volts.

Before starting the test, the background was calculated to be sure that there were no problems. The results were:

<table>
<thead>
<tr>
<th></th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
<th>G</th>
<th>H</th>
<th>I</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>BG</td>
<td>0.00123058</td>
<td>0.0003819</td>
<td>0.00102009</td>
<td>0.0148114</td>
<td>1.48265</td>
<td>0.00389424</td>
<td>0.001208547</td>
<td>0.03228133</td>
</tr>
</tbody>
</table>

So, all the results expected has to be over it.

The text file that we got was like the following:
Using the software Grapher, standard deviations were calculated for each of the four channels and, moreover, the mean for the thermal anemometry channel. These data were passed to an excel file (one for each part), with which we worked throughout the project.
Figure 33 Excel file 1

For further work, it was necessary to change the units of each of the channels. In the first and the second channel were converted to Volt into Pascal using the following conversion factor:

\[
\text{Pa} = 316 \text{ mV in Channel 1}
\]

\[
\text{Pa} = 31.6 \text{ mV in Channel 2}
\]

In the third channel to the conversion of volts into mm/s was used as follows:

\[
\text{mm/s} = 31.6 \text{ mV}
\]

In this part of the project the excel file was like the following:
Afterwards, data from the fourth channel were processed, that is, the standard deviation and the mean. To change the units it was necessary to get the calibration curve to relate Volt with m/s. This calibration curve was found from the data provided by the laboratory staff:

<table>
<thead>
<tr>
<th>R [Hz]</th>
<th>C1 (St. ch)</th>
<th>C2 (St. ch)</th>
<th>C3 (St. ch)</th>
<th>C4 (St. ch)</th>
<th>mean C1</th>
<th>ch1 [Pa]</th>
<th>ch2 [Pa]</th>
<th>ch3 [units]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.004547</td>
<td>0.240722</td>
<td>0.19613</td>
<td>0.142089</td>
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<td>6.32196203</td>
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</tr>
<tr>
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<td>2.68917</td>
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<td>6.32196203</td>
<td></td>
</tr>
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<td>6.32196203</td>
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<td>2.68917</td>
<td>20.42364568</td>
<td>7.11721919</td>
<td>6.32196203</td>
<td></td>
</tr>
</tbody>
</table>

Figure 34 Excel file 2

Afterwards, data from the fourth channel were processed, that is, the standard deviation and the mean.

To change the units it was necessary to get the calibration curve to relate Volt with m/s. This calibration curve was found from the data provided by the laboratory staff:
<table>
<thead>
<tr>
<th>$U$ [V]</th>
<th>$pt$ [mm C2H5OH]</th>
<th>$pot$ [mm C2H5OH]</th>
<th>$p_{dyn}$ [Pa]</th>
<th>$c$ [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1,633</td>
<td>159</td>
<td>159</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>3,150</td>
<td>149</td>
<td>160</td>
<td>88,4862</td>
<td>12,19903058</td>
</tr>
<tr>
<td>3,464</td>
<td>136</td>
<td>161</td>
<td>201,1050</td>
<td>18,39073057</td>
</tr>
<tr>
<td>3,593</td>
<td>123</td>
<td>162</td>
<td>313,7238</td>
<td>22,97001511</td>
</tr>
<tr>
<td>3,696</td>
<td>110</td>
<td>163</td>
<td>426,3426</td>
<td>26,77730789</td>
</tr>
<tr>
<td>3,781</td>
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<td>4,043</td>
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<td>35,85013373</td>
</tr>
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<td>876,8178</td>
<td>38,40097281</td>
</tr>
<tr>
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<td>52</td>
<td>174</td>
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<td>40,62645169</td>
</tr>
<tr>
<td>4,303</td>
<td>42</td>
<td>178</td>
<td>1094,0112</td>
<td>42,89418610</td>
</tr>
</tbody>
</table>

**Figure 35** Calibration table

**Figure 36** Calibration curve

$$c \, [m/s] = 3.942878006 \times \ln(U) - 1.939501103$$
Therefore, the conversion factor used was:

\[ c \text{ [m/s]} = 3.942878006 \cdot \ln(U) - 1.939501103 \]

where \( U \) is the mean in channel four and it is in Volts.

Afterwards, turbulence intensity was found dividing the standard deviation between the velocity.

Finally, we converted the units from channels 1 and 2 by the follow factor conversion:

\[ \frac{P}{P_{\text{ref}}} = 10^{\frac{L}{20 \log_{10}(P/P_{\text{ref}})}} \]

where \( P \) is the value of the pressure in Pa, \( P_{\text{ref}} = 2 \cdot 10^{5} \) Pa is the Pressure reference and \( L \) is the noise level, measured in dB.

---

**Figure 37 Excel file 3**

Finally, we converted the units from channels 1 and 2 by the follow factor conversion:
Figure 38 Excel file 4
6. Correlation between turbulence intensity and acoustic noise level

Once we have the data as we showed before, we can find the correlation between the turbulence intensity and the acoustic noise level. Before find this correlation, it would be interesting to analyze some graphs.

6.1 Part I

We will start with the first part (cylinders stood). The first graph (Fig. 39) shows the relation between the noise level in channel 1 and the rotor velocity

![Figure 39 L1 [dB] versus n [Hz]. Part I](image)

In the second graph (Fig. 40) it is possible to observe the relation between the noise level in channel 2 and the rotor velocity.
The third graph (Fig. 41) shows the relation between the air velocity and the rotor velocity

Figure 40 L2 [dB] versus n [Hz]. Part I

Figure 41 V [m/s] versus n [Hz]. Part I

Another important graph, before find the correlation between the turbulence intensity and the acoustic noise level, is the relation between the intensity turbulence and the rotor velocity (Fig. 42)
Finally, before finding the correlation, we can draw a scatter plot (Fig. 43) between the turbulence intensity and the values from the first channel (in Pascals).

**Figure 42** Turbulence intensity versus $n$ [Hz], Part I

**Figure 43** Scatter plot Turbulence intensity versus Noise level in Ch1. Part I
It is possible to observe the equation of the regression line,

\[ y = -0.0199x + 0.0798 \]

and the \( R^2 \) coefficient,

\[ R^2 = 0.1658 \]

Now it is possible to find the correlation between the turbulence intensity and the acoustic noise level (channel 1). We also can find the next correlations:

<table>
<thead>
<tr>
<th></th>
<th>ch1 [Pa]</th>
<th>ch2 [Pa]</th>
<th>ch3 [mm/s]</th>
<th>Tu [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>ch1 [Pa]</td>
<td>1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ch2 [Pa]</td>
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<td></td>
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<tr>
<td>ch3 [mm/s]</td>
<td>-0.83136418</td>
<td>-0.819524</td>
<td>1</td>
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</tr>
<tr>
<td>Tu [-]</td>
<td>-0.40712734</td>
<td>-0.358301603</td>
<td>0.313645361</td>
<td>1</td>
</tr>
</tbody>
</table>

6.2 Part II

For the second part we can find similar graphs. The first one (Fig. 44) is the relation between the noise level in channel 1 and the rotor velocity.
In the second graph (Fig. 45) it is possible to observe between the noise level in channel 2 and the rotor velocity.

**Figure 45** $L_2$ [dB] versus $n$ [Hz]. Part II

The third graph (Fig. 46) shows the relation between the air velocity and the rotor velocity.

**Figure 46** $V$ [m/s] versus $n$ [Hz]. Part II
And finally, here we have the relation between the intensity turbulence and the rotor velocity (Fig. 47)

Figure 47 Turbulence intensity versus n [Hz]. Part II

Afterwards, we can draw a scatter plot (Fig. 48) between the turbulence intensity and the values from the first channel (in Pascals).

Figure 48 Scatter plot Turbulence intensity versus Noise level in Ch1. Part II
As before, we can observe the equation of the regression line,

\[ y = -0.0759x + 0.1387 \]

and the \( R^2 \) coefficient,

\[ R^2 = 0.378 \]

To conclude this part of the project, we are going to find the correlation between the turbulence intensity and the acoustic noise level (channel 1). We also can find the next correlations:

<table>
<thead>
<tr>
<th></th>
<th>ch1 [Pa]</th>
<th>ch2 [Pa]</th>
<th>ch3 [mm/s]</th>
<th>Tu [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>ch1 [Pa]</td>
<td>1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ch2 [Pa]</td>
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<td></td>
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<tr>
<td>ch3 [mm/s]</td>
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</table>

7. Conclusions

Along the project it had been showed how we got the data and how we should modify it to achieve the correlation between intensity turbulence and acoustic noise level.

In this part this results are going to be interpreted. At first we are going to comment the first part and afterwards the second one.

As we can see, the correlation between turbulence intensity and acoustic noise level when the cylinders were stood is

\[ -0.40712734 = -40.712734 \% \]

The first important aspect to emphasize is the negative sign. It indicates that the relation between turbulence intensity and acoustic noise level is inversely proportional.

Moreover, it is possible to observe that this correlation is not so important since it doesn’t achieve even 50%.
On the other hand, for the second part of the test, where the cylinders were rotating, the correlation between turbulence intensity and acoustic noise level is

\[-0.614856784 = -61.4856784\%\]

As in the first part, the correlation remains negative. However, the value has increased more than a 20%. So, it is possible to say that the correlation is higher when the cylinders stay in movement. It indicates that we must take into account the cylinders.
8. Bibliography


