ON THE GENERATION OF UNSTEADY MEAN AND TURBULENT FLOWS IN A WIND TUNNEL USING AN ACTIVE GRID

by

Abdullah Azzam

A thesis submitted in conformity with the requirements for the degree of Master of Applied Science
University of Toronto Institute for Aerospace Studies
University of Toronto

© Copyright 2018 by Abdullah Azzam
Abstract

On the Generation of Unsteady Mean and Turbulent Flows in a Wind Tunnel using an Active Grid

Abdullah Azzam
Master of Applied Science
University of Toronto Institute for Aerospace Studies
University of Toronto
2018

Unsteady flows are pervasive in nature and engineering applications such as UAVs. Simulating these flows in an experimental setting is key to avoiding detrimental changes in the performance of such applications. This study explores the capabilities of an active grid in producing unsteady flows in a recirculating wind tunnel. The grid is operated in different modes that allow control over the flow frequency, amplitude and turbulence intensity. Hot-wire measurements demonstrate the grid's capabilities in generating flows typical of those used in experiments of unsteady aerodynamics. Mean grid blockage, freestream speed and tunnel size were all found to be factors influencing the resulting flow. The response of turbulence to the oscillating flow was also investigated and results showed a modulation of the turbulence intensity and dissipation rate that is dependent on the frequency of the imposed oscillations. A change in the scaling behaviour of the dissipation rate was also noted.
Acknowledgements

Doing research in this field has been a dream that I’ve had for quite some time. To see it realized through this thesis makes me appreciate the exceptional people around me who helped make it happen.

I am very thankful to my supervisor Professor Philippe Lavoie, for granting me the opportunity to join his team and for his constant support. His extensive knowledge, attention to detail and relentless attitude in problem-solving have been inspiring and pushed me to give my best. The support provided by the Natural Sciences and Engineering Research Council for this work is greatly appreciated.

The support of all my lab mates at the Flow Control and Experimental Turbulence lab has been invaluable. Your technical guidance, physical abilities (installing the active grid was not an easy task!) and our conversations are all things I am grateful for.

Thank you to my childhood friends, or rather family, Ryan, Haya, Raed, Rawad, and Wajeb for making life easier in so many ways, to my friend Stephanie, who constantly provided encouragement, and to Rawad and Chantal who have always been so supportive and caring.

I’m grateful for my aunt, uncle and cousins who have been my home away from home for the past two years.

No acknowledgement section would be complete without expressing the immense gratitude that I have for my family. Thank you to my parents and brothers for your constant support. Each one of you has always led by example and I am lucky to have you. This wouldn't be possible without you.
# Contents

Acknowledgements ........................................ iii

Table of Contents ........................................ iv
List of Tables ............................................. vi
List of Figures ............................................. vii

1 Introduction ........................................ 1
   1.1 Motivation and Objectives ................................. 4
   1.2 Thesis Outline ........................................ 5

2 Literature Review ........................................ 6
   2.1 Unsteady Flows ......................................... 6
   2.2 Active Grids ........................................... 10
   2.3 Unsteady Turbulence ..................................... 13

3 Experimental Setup ....................................... 17
   3.1 Wind Tunnel ........................................... 17
   3.2 Active Grid ............................................ 19
   3.3 Instrumentation ........................................ 20
   3.4 Uncertainty Analysis ................................... 22

4 Unsteady Flow Analysis .................................... 24
   4.1 Triple Decomposition .................................... 24
   4.2 Unsteady Flow Capabilities ............................... 27
   4.3 Dynamic Model .......................................... 32
   4.4 Turbulence Considerations ............................... 38
List of Tables

1.1 Wind speeds compared to cruising speeds of insects, birds, UAVs and other aircraft. Modified and adapted from Tennekes (2009).................................................. 2

2.1 Typical amplitude and reduced frequency values used in unsteady aerodynamics experiments................................................................. 7

3.1 Tunnel section lengths and areas................................................................. 18
List of Figures

1.1 Applications of fractal grid elements in (a) micro-channel and (b) aircraft spoilers (Vassilicos et al., 2007). ............................................. 3

2.1 Louver system used by Greenblatt (2016) to generate velocity oscillations. 8

2.2 Passive grids with varying mesh lengths (Zilli, 2017). ....................... 10

3.1 Recirculating wind tunnel schematic, adapted and modified from Hearst (2015). Section numbers correspond to Table 3.1. ....................... 18

3.2 Active grid mounted to test section, stepper motors are shown around the perimeter of the grid in black. ............................. 19

3.3 Active grid flapping angle variation as a function of time for different operating modes. .......................................................... 20

3.4 Downstream instrumentation. .................................................. 21

4.1 Sample velocity measurements taken upstream (red) and downstream (blue) of the grid at a mean flow velocity of 4 m/s and frequency of 1 Hz with $\beta = 90^\circ$ and 50% open times for Modes 2 and 3. .............................. 24

4.2 Autocorrelation of the measured flow velocity using Mode 0 at a grid frequency of 0.1 Hz. Inset shows the oscillatory behaviour of the autocorrelation. ................................................................. 26

4.3 (a) Upstream and (b) downstream comparison of original velocity signal with phase averaged data points and Fourier least squares fit. .......... 27

4.4 Triple decomposition of velocity signal into mean, periodic and fluctuating components. ......................................................... 27

4.5 Power spectral density for upstream and downstream measurements. . . 27

4.6 Variation of upstream flow amplitude with freestream velocity for Mode 0 and a grid frequency of 1 Hz. ........................................ 29

4.7 Variation of upstream flow amplitude with grid frequency for Mode 0 and a mean velocity of 4 m/s .................................................. 29
4.8 Variation of upstream flow amplitude with flapping angle for Mode 1 at a
grid frequency of 1 Hz and a mean velocity of 4 m/s.  

4.9 Comparison of velocity response for Mode 2 at a grid frequency of (a) 0.1
Hz, (b) 0.5 Hz and (c) 1 Hz.  

4.10 Effect of open time on velocity time series for Mode 2, presented here for
a grid frequency of 1 Hz.  

4.11 Sinusoidal flow velocity approximation using Mode 3 with a frequency of
0.09 Hz.  

4.12 Comparison of unsteady flow conditions used in literature to those pro-
duced by the active grid.  

4.13 Unfolded tunnel schematic.  

4.14 Variation of time constant and cutoff frequency, $f_c$, with blockage, ex-
pressed in terms of the average grid area and flapping angle.  

4.15 Comparison of experimental magnitude ratio with that predicted from
experiments for different operating modes and flow frequencies, plotted as
$\omega = 2\pi f$. Maximum error on $M_e$ was 6%.  

4.16 Collapse of data from all experimental runs when plotted against nondi-
mensional flow frequency $\omega \tau$. Maximum error on $M_e$ was 6%.  

4.17 Static amplitude map for different test conditions.  

4.18 Comparison of the power spectral density for $u(t)$ and $u'(t)$, under dif-
f erent grid wing rotation frequencies. PSDs for the different cases are
intentionally offset by 6 decades for clarity.  

4.19 Comparison between the premultiplied spectra of the total velocity com-
ponent and periodic components at flow frequencies of (a) 1 Hz and (b) 4
Hz.  

4.20 Comparison between the dissipation spectra of the total velocity compo-
nent and periodic components at flow frequencies of (a) 1 Hz and (b) 4
Hz.  

4.21 Comparison of upstream power spectral density spectrum at a mean speed
of 4 m/s and grid frequency of 4 Hz for $u(t)$, $u'(t)$ and the case when the
grid is open and not moving.  

4.22 Comparison between first three flow cycles (blue), middle cycles (red) and
last three cycles (green) of an experiment run for (a) mode 1 at $\beta = 30^\circ$
and 1 Hz and (b) mode 0 at 4 Hz.  

4.23 Variation of upstream turbulence intensity with grid Reynolds number.  

4.24 Variation of downstream turbulence intensity with grid Reynolds number.
4.25 Variation of downstream turbulence intensity with Rossby number.

4.26 Variation of (a) streamwise integral length scale, (b) Taylor microscale and (c) dissipation scale with grid Reynolds number.

4.27 Variation of turbulence intensity with flapping angle at a mean speed of 4 m/s.

4.28 Turbulence intensity homogeneity profiles at $x/M = 26$ with FR at $3 \pm 2$ Hz.

5.1 Phase variation of the periodic velocity component and turbulence intensity for (a) a step input and (b) Mode 0 at 0.1 Hz for a mean velocity of 4 m/s.

5.2 Phase variation of velocity, turbulence intensity and dissipation rate at 4 m/s and for grid frequencies of (a) 0.1 Hz (b) 0.5 Hz (c) 1 Hz and (d) 2 Hz. The maximum error on the phase averaged dissipation rate was 6.4%.

5.3 Amplitude and phase difference variation of the turbulent kinetic energy dissipation rate with respect to frequency and mean velocity.

5.4 Power spectral density spectra of $u'(t)$ for the cases where a change in dissipation response takes place.

5.5 Variation of integral time scale for each flow period at 4 m/s for different flow frequencies.

5.6 Phase variation of $C_\epsilon$ at 4 m/s for different flow frequencies.

5.7 Phase variation of $C_\epsilon/\sqrt{Re_M}$ with $Re_\lambda$ at 4 m/s and 7 m/s for different flow frequencies. Error bars were omitted for clarity, but are similar to those shown in Figure 5.6.

5.8 (a) Variation of $C_\epsilon/\sqrt{Re_M}$ with $Re_\lambda$ at 7 m/s and 0.1 Hz with phases identified in accordance with (b), the phase variation of the velocity. Red symbols correspond to regions of constant $C_\epsilon$, while magenta symbols indicate scaling according to (5.13).
Chapter 1

Introduction

The work presented in this thesis centers around the production of unsteady flows in a recirculating wind tunnel using an active grid. This has two primary applications. First, generating tailored unsteady flows enables experiments in unsteady aerodynamics on par with those conducted by Miller & Fejer (1964), Grandlund et al. (2014), Greenblatt (2016) and Strangfeld et al. (2016) using dedicated facilities. Second, the produced unsteady flows allow for a novel experimental investigation of unsteady turbulence and its dynamics, which is a currently growing field of study as evidenced by the review of Vassilicos (2015) on the matter.

Unsteady, or time-varying, flows are pervasive in nature and many practical engineering applications. Wind gusts are present both in the free, inviscid atmosphere as well as in the atmospheric boundary layer, whose thickness can range anywhere from 0.5 km to 2 km above ground level (Vrankova & Palko, 2016). Closer to the ground, surface features such as buildings or trees, dictate the local flow field as a result of the wakes they generate. In fact, the unsteadiness found in nature is interwoven into the challenges faced by many engineering applications. For instance, wind turbines are subject to conditions of the atmospheric boundary layer in addition to more local factors pertaining to the surrounding environment (Hearst & Ganapathisubramani, 2017). As a result, it is imperative that the design of wind turbines takes into consideration the unsteady environment within which they are operating.

Similar challenges are also faced with the advent of the rapid commercialization of unmanned aerial vehicles (UAVs). The use of these vehicles spans several industries such as surveillance, agriculture and entertainment. As a result, UAVs are becoming smaller, more lightweight and are operating in the so-called roughness zone of the atmospheric boundary layer near buildings and hilltops (Watkins et al., 2006). Therefore, as pointed out by Smith et al. (2016), UAVs have become increasingly susceptible to wind gusts and
unsteadiness due to their smaller size and the fact that they operate at an airspeed that is on the same order of the typically encountered gusts. Most commercial UAVs usually have cruising speeds ranging from 8 m/s to 20 m/s. This makes them sensitive to most breeze strengths, as well as gales as shown in Table 1.1.

Table 1.1: Wind speeds compared to cruising speeds of insects, birds, UAVs and other aircraft. Modified and adapted from Tennekes (2009).

<table>
<thead>
<tr>
<th>Beaufort no.</th>
<th>Gust</th>
<th>Windspeed (m/s)</th>
<th>Cruising speed of</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Light air</td>
<td>0.5 - 1</td>
<td>Butterflies</td>
</tr>
<tr>
<td>2</td>
<td>Light breeze</td>
<td>2 - 3</td>
<td>Dragonflies</td>
</tr>
<tr>
<td>3</td>
<td>Gentle breeze</td>
<td>4 - 5</td>
<td>Human-powered airplanes</td>
</tr>
<tr>
<td>4</td>
<td>Moderate breeze</td>
<td>6 - 8</td>
<td>UAVs, MAVs, bees, crows, swans</td>
</tr>
<tr>
<td>5</td>
<td>Fresh breeze</td>
<td>9 - 11</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Strong breeze</td>
<td>11 - 13</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Near gale</td>
<td>14 - 17</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Gale</td>
<td>18 - 21</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Strong gale</td>
<td>21 - 24</td>
<td>Sailplanes</td>
</tr>
<tr>
<td>10</td>
<td>Storm</td>
<td>25 - 28</td>
<td>Home-built aircraft</td>
</tr>
<tr>
<td>11</td>
<td>Violent storm</td>
<td>29 - 32</td>
<td>Light aircraft, diving hawks</td>
</tr>
<tr>
<td>12</td>
<td>Hurricane</td>
<td>&gt; 32</td>
<td>Diving falcons</td>
</tr>
</tbody>
</table>

It is then important to develop control schemes that make UAVs resilient to the wind gusts or unsteadiness that is encountered during their missions. Part of that effort involves understanding the unsteady loading incurred by such conditions and systematically testing proposed designs under those conditions. Research in unsteady aerodynamics played a crucial role in that effort by creating a theoretical framework for understanding the generated unsteady forces (Theodorsen (1935), Isaacs (1945) and Greenberg (1947)) and experimentally testing the proposed theories under simulated unsteady conditions (e.g. Strangfeld et al. (2016)).

The first part of this work fits into the experimental portion of the effort outlined above, and involves creating a comprehensive test environment for experiments in unsteady aerodynamics in low Reynolds number regimes. This enables the extension of the work done by Grandlund et al. (2014) and Strangfeld et al. (2016) to include the effect of freestream turbulence intensity on the generated unsteady lift, and also the creation of a physics-based model that describes the unsteady loading during flow separation, which hasn’t been accounted for in the classical theories. Furthermore, the aim is to provide the experimentalist with the knowledge and tools needed to customize the generated
unsteady flows to suit a particular application.

On the other hand, understanding the fundamentals of turbulence, such as its production, dissipation, decay and scaling has many important practical applications. Our current understanding of turbulence stems from the theories developed by Taylor (1935) governing the scaling of the dissipation rate, and the Richardson-Kolmogorov energy cascade. However, recent work in this area exhibited a deviation from the classical Taylor (1935) scaling in certain regions of the flow prior to being fully developed and homogeneous. As a result, this allows the development of theories that enable the design of devices that operate in conditions where the flow might not be fully homogeneous (such as directly after mixers or pipe fittings) and hence make use of the newly found scaling to improve their efficiency (Mazellier & Vassilicos, 2010). Applications include enhancing the mixing process in pipes and combustion chambers, facilitating heat transfer, microchip cooling and reducing aircraft spoiler noise (Nagata et al. (2013), Vassilicos et al. (2007), Nicolleau et al. (2011) and Nedi et al. (2012)). For instance, mixing in micro-channels can be enhanced by using a fractal element as shown in Figure 1.1a. Additionally, Vassilicos et al. (2007) and Nedi et al. (2012) have shown that using fractal grids in lieu of the traditional spoilers or airbrakes causes a reduction in the noise generated by these devices. An example of this is shown in Figure 1.1b.

Therefore, the second part of this thesis will build upon recent research in this field by studying the fundamental nature of turbulence during freestream velocity oscillations. Previous efforts investigated turbulence generated by regular passive grids or fractal
grids to identify regions where the classical scaling laws do not hold, or used direct numerical simulations (DNSs) to investigate the response of turbulence to a spatially periodic forcing of the flow (Vassilicos, 2015). In contrast, this study uses an active grid to periodically force the flow and provide a novel insight into the dynamic response of unsteady turbulence. This allows the experimental extension of the results from DNS studies, and the investigation of the phase-averaged modulation of various turbulence quantities and their scaling behaviour, while drawing parallels between the effects of the spatial forcing used in DNSs and the forcing used here.

1.1 Motivation and Objectives

The motivation for this project stems directly from the two applications discussed earlier. First, is the need to produce customizable flows with varying degrees of mean flow unsteadiness and turbulence intensities to enable further advances in unsteady aerodynamics research. Second, is gaining a deeper understanding of the dynamics of unsteady turbulence through a new experimental approach. Therefore, the aim of this project will be realized through completing the following objectives:

- Develop the capability of producing unsteady flows in the recirculating wind tunnel facility at the University of Toronto Institute for Aerospace Studies.
- Determine active grid capabilities in producing unsteady flows through testing a number of grid operation modes and identifying key grid parameters that allow for the customization of the produced flow.
- Assess the effectiveness of the produced unsteady flows by comparison with flows typically used in experiments of unsteady aerodynamics.
- Characterize the dynamic response of the wind tunnel and active grid system by understanding the impact that certain tunnel and active grid parameters have on that response.
- Characterize the resulting turbulence during unsteady flow production and propose methodologies that allow for customizing various turbulence characteristics.
- Investigate the dynamics of the produced turbulence and assess its response under the action of the oscillating flow.
- Investigate the scaling behaviour of turbulence in response to the imposed temporal forcing.


\section*{1.2 Thesis Outline}

This thesis is organized as follows: chapter 2 provides a review of the currently available means to produce unsteady flows, applications of active grids, and results from recent studies that investigated unsteady turbulence. Chapter 3 details the experimental setup, chapter 4 explores the range and characteristics of the produced unsteady flows, chapter 5 provides insight into the dynamics of turbulence under the action of an oscillating flow, and chapter 6 summarizes the work done with conclusions and future recommendations.
Chapter 2

Literature Review

2.1 Unsteady Flows

The fundamental theory governing the prediction of unsteady forces on airfoils was put forth by Theodorsen (1935). In that work, the potential flow theory and the Kutta condition were employed in order to estimate the circulation over an airfoil undergoing purely sinusoidal motion about an equilibrium position in a steady freestream velocity. Much of the theory governing unsteady aerodynamics was motivated by estimating the loads of rotating helicopter blades during flight. Therefore, Wall & Leishman (1992) pointed out that Theodorsen’s theory presented a limitation since it did not take into account any fluctuations in the freestream velocity, which typically arise from the lead-lag motion of the blades in addition to the presence of wind gusts. As reported by Strangfeld et al. (2016), Theodorsen’s work was later extended by Isaacs (1945) and Greenberg (1947) to include the effect of an oscillating freestream velocity and account for the previously neglected freestream variations.

Furthermore, the presence of such theories raises the question of how should unsteady airfoil behaviour be characterized experimentally, and if there is a difference in loading between an oscillating freestream with a static model, and an oscillating model in a steady freestream velocity. In fact, this issue was investigated by Grandlund et al. (2014), who showed that in an unsteady freestream, the loads on a static model include the unsteady aerodynamic forces, in addition to an added pressure gradient effect that is needed to sustain the oscillatory flow. This effect is comparable to the inertial force experienced by an oscillating model due to the model’s acceleration with respect to the steady flow. The two methods were then found to be equivalent in evaluating the gust response of aerodynamic bodies, but only after the necessary corrections have been made in order to account for the employed testing method.
In this work, emphasis is placed on creating unsteady oscillations of the freestream velocity. These oscillating velocities, such as those produced by Strangfeld et al. (2016) and Grandlund et al. (2014) typically have the form,

\[ u(t) = \bar{u}(1 + \sigma \sin(2\pi ft)) \],

(2.1)

where \( \bar{u} \) is the mean freestream speed, \( f \) is the flow frequency, and \( \sigma \) is the dimensionless velocity amplitude given by

\[ \sigma = \frac{u_p}{\bar{u}} \],

(2.2)

where \( u_p \) is the peak velocity. The frequency of the flow is often expressed in its dimensionless form, called reduced frequency, viz.

\[ k = \frac{2\pi fc}{\bar{u}} \],

(2.3)

where \( c \) is the chord length of the model under consideration. Table 2.1 shows typical values of these quantities that are usually used in experiments.

Table 2.1: Typical amplitude and reduced frequency values used in unsteady aerodynamics experiments.

<table>
<thead>
<tr>
<th>Study</th>
<th>( \sigma )</th>
<th>( k )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Grandlund et al. (2014)</td>
<td>0.1</td>
<td>0.1 to 3</td>
</tr>
<tr>
<td>Strangfeld et al. (2016)</td>
<td>0.34, 0.5</td>
<td>0.08, 0.98, 0.074, 0.097</td>
</tr>
<tr>
<td>Yang et al. (2017)</td>
<td>0.2</td>
<td>0.094, 0.19, 0.28, 0.38</td>
</tr>
</tbody>
</table>

Multiple techniques have been used to generate oscillating flows, including the implementation of a movable tunnel wall through a cam system by Holmes (1973) to generate temporally and spatially varying gusts. More recently, Yang et al. (2017) used a multi-fan array to create oscillations in the freestream velocity. However, the most common way of producing oscillations is by using rotating shutter systems that are installed inside the wind tunnel. The rotation of the shutters causes a change in the static pressure through the variation of the cross sectional area of the tunnel, thereby inducing flow oscillations (Al-Asmi & Castro, 1993). A recent example of such a system is that designed by Greenblatt (2016), and is shown here in Figure 2.1. The system consists of a servomotor that drives the rotation of the shutters by means of cogwheels connected to the shutters. The frequency of the oscillations is achieved by varying the tunnel blockage at the desired frequency through commands to the motor. These systems can either be placed upstream...
or downstream of the test section, however an upstream placement causes large disturbances to the flow and the production of harmonic peaks (Greenblatt, 2016). Variations on the shutter system have also been developed in order to produce temporally and spatially varying gusts. Tang et al. (1996) used a vane system consisting of four airfoils with rotatable slotted cylinders at their trailing edge in order to generate single and multiple harmonic gust waves. The amplitudes and frequencies of the generated waves depended on various factors such as the spacing between the cylinders and their sectional profile. Similarly, Poussot-Vassal et al. (2017) generated vertical gusts through the use of a gust generator that contained two pitching airfoils.

Figure 2.1: Louver system used by Greenblatt (2016) to generate velocity oscillations.

Wind tunnels with such systems are usually characterized by the maximum oscillation amplitude that is achievable (usually given as a percentage of the mean, as in (2.2)), and the amount of attenuation that they impose on the generated flow velocity, which is the result of the dynamic response of the tunnel to the imposed static pressure fluctuations. Specifically, the desired frequency of the flow dictates the amount of amplitude attenuation that occurs, whereby higher flow frequencies cause smaller velocity amplitudes. This attenuation is characterized by the cut-off frequency of the tunnel as shown by Greenblatt (2016), which depends on the imposed blockage, mean velocity and tunnel dimensions. The effect of these parameters will be illustrated in greater depth in chapter 4.

Furthermore, some control of the resulting amplitude may be exercised by varying the width of the used shutters and consequently adjusting blockage, as in Miller & Fejer (1964) who replaced the shutters with louvers of variable widths, which resulted in the production of velocity amplitudes ranging from 8% to 92% of the mean velocity. The motor powering this system was capable of achieving frequencies of 4 to 125 Hz, and
while using the same facility, Grandlund et al. (2014) reported a maximum sinusoidal flow frequency of 6 Hz, with the first harmonic sufficiently away from this dominant frequency.

Another method for controlling the amplitude and general response of the tunnel to louver motions was proposed by Rennie et al. (2017), who investigated the effect of using a breather located downstream of the test section in a recirculating wind tunnel. The breather allows air to enter and exit the wind tunnel circuit during operation of a louver system similar to that shown in Figure 2.1. At the lower louver frequencies, the mass flow entering and exiting through the breather was negligible compared to the freestream velocity oscillations and so the tunnel’s response was similar to the case when the breather was closed. However as the louver frequency is increased, the freestream velocity amplitude drops, as established earlier, and so the mass flow travelling through the breather plays a more dominant role. This allows for the creation of higher amplitudes at the higher frequencies since less effort is required for the flow to enter and exit the breather, than to be oscillated around the entire wind tunnel circuit.

In an effort to tailor the produced unsteady flow, Rennie et al. (2018) sought to reproduce a desired gust spectrum with user tunable parameters, such as velocity and gust length scale. The gust model used was the von Karman model, and the dynamics of the tunnel were modelled using the method of characteristics which tracks the finite speed with which pressure disturbances propagate within the tunnel in order to predict the resulting velocity time series. Using these models, Rennie et al. (2018) implemented an open-loop controller that outputs the needed louver motions by correcting the desired gust spectrum through accounting for the tunnel dynamic response at the prescribed conditions. Results show an accurate production of the desired gust spectrum and its amplitude. Other control strategies, such as the neural network predictive control method proposed by Sutcliffe & Rennie (2016), also have the potential to be used during unsteady flow production as pointed out by Rennie et al. (2017).

Therefore, unsteady flows have been shown to be readily produced using various configurations, however the systems used are often limited in their flexibility of producing customizable flow properties without undergoing some hardware changes. Additionally, imposing desired turbulence characteristics on the flow would require the use of a separate device, such as a passive grid, which also necessitates the availability of multiple grids to achieve different turbulence intensities. A possible solution to this is to make use of the capabilities of an active grid, which will be discussed in the next section.
2.2 Active Grids

Production of turbulence in a wind tunnel has often been accomplished by means of grids due to their ability to produce homogenous and nearly isotropic turbulence (Makita, 1991). The most commonly used grids are passive grids such as those shown in Figure 2.2. Several parameters govern passive grids and have an effect on their performance. These include whether the grid is made up of square or parallel meshes, with either round or square bars, in addition to the distance between meshes and the bar dimensions. Turbulence intensities typically show a power law decay behaviour with distance, while the turbulent energy and the length scales scale with the bar dimensions (Roach, 1987).

A major limitation of passive grid generated turbulence is the absence of an appreciable scaling range that allows for the study of fundamental turbulence flows and the replication of certain desired flow conditions. As a result, Makita (1991) proposed the design of a new device that actively perturbs the flow using a bi-planar mesh of motor controlled rods onto which agitator wings are mounted. It was found that such a configuration produced homogeneous and nearly isotropic turbulence with an appreciable scaling range that spanned approximately two wavenumber orders of magnitude on the energy spectrum. The introduction of this new type of grids significantly increased the parameter space within which they can be operated, in comparison with the traditional passive grids. The additional parameters include variables such as the wing rotation rate, direction and overall mode of operation. This was briefly touched upon by Mydlarski & Warhaft (1996) who operated their grid in a random and synchronous mode. In random mode, the wings were driven at a constant speed but with randomly varying times and directions of rotation, while synchronous mode involved no direction switching. Lower turbulence intensity and integral length scale values were reported using the synchronous mode. Poorte & Biesheuvel (2002) introduced a variation on the forcing protocols used by Mydlarski & Warhaft (1996) through using a double-random asynchronous mode that involved randomly varying rotation times as well as wing rotation rates. They observed
that this mode minimized contaminations of the energy spectra, which usually result from constant wing rotational rates (Mydlarski & Warhaft, 1996).

Mydlarski (2017) also reported a variety of other active grid operating modes that have been used to customize flows. For instance, Cekli et al. (2015) actuated their active grid using random signals to the motors, where they not only controlled the distributions of the sent signal, but also the correlations between different motors. It was found that commanding the motors using time scales that match that of the flow produces isotropic turbulence, while imposing a mismatch between the scales causes significant large-scale intermittency in the flow, which can be helpful in modelling atmospheric flows. On the other hand, Weitemeyer et al. (2013) investigated the effect of varying the local grid solidity by maintaining a fixed phase difference between the grid wings over time, thereby producing a constant long-time solidity. Results showed that altering the local solidity by varying the initial angle at which the wings are set, allows for the customization of the downstream decaying behaviour of the grid.

Therefore, it is apparent that the produced turbulence will heavily depend on the used grid parameters. Two extensive parametric studies were completed by Larssen & Devenport (2011) and Hearst & Lavoie (2015) to identify the dependency of various turbulence statistics on the grid actuation parameters. Both studies concluded that the freestream speed and wing rotational frequency play the most dominant role in the creation of turbulence. The mean velocity is often represented through the grid Reynolds number which is based on the grid mesh length viz.

\[ \text{Re}_M = \frac{\bar{u}M}{\nu}, \]  

where \( M \) is the grid mesh length and \( \nu \) is the kinematic viscosity. Additionally, the grid frequency was nondimensionalized by Larssen & Devenport (2011) using the Rossby number, which is a ratio of the inertial forces to rotational forces, given by

\[ \text{Ro} = \frac{\bar{u}}{Mf}. \]  

Both studies showed that an increase in either \( \text{Re}_M \) or \( \text{Ro} \) caused an increase in the turbulence intensity and the produced integral length scale. Additionally, Hearst & Lavoie (2015) showed that a saturation in the values of the turbulence intensity is reached for values of \( \text{Ro} \) beyond those tested by Larssen & Devenport (2011). An additional parameter tested by Hearst & Lavoie (2015), was the wing shape, and results showed that the turbulence intensity is primarily affected by the resulting blockage rather than the shape.
itself, whereby shapes causing a greater blockage caused higher turbulence intensities. Therefore, active grids are capable of producing a wide array of flow conditions, which can simply be realized by changing the grid actuation parameters that are sent to the driving motors.

While active grids have been shown to provide a great range of turbulent flow conditions, there application has also extended to other studies, beyond that of turbulence fundamentals. For instance, Hearst & Ganapathisubramani (2017) used an active grid in order to simulate the flow fields typically encountered by wind turbines. In their work, they tailored the turbulence of shear flows by flapping the vertical bars of an active grid to create a desired shear flow profile, and then used the horizontal bars to impose turbulence on the generated flow through random motions similar to those discussed earlier. The implemented scheme offered the ability to “mix and match” between turbulence levels and shear profiles in order to generate a specific flow condition. Similarly, Mydlarski (2017) points out the method employed by Good & Warhaft (2011) which entailed removing the center agitator wings from the active grid in an effort to simulate the complex flow field encountered by wind turbines. Cekli et al. (2010) used spatial and temporal forcing protocols of an active grid as a means to investigate the behaviour of the turbulent kinetic energy dissipation rate during flow modulation. It was found that there exists a resonant flow modulation frequency at which the dissipation rate peaks for a given spatial forcing, and that this frequency is equal to the large eddy turnover rate of the flow. Furthermore, Heißelmann et al. (2016) used an active grid to characterize the performance of a wind turbine airfoil section in laminar as well as tailored turbulent conditions. The turbulent conditions were produced by actuating the active grid in one of two modes. The first is a sinusoidal velocity variation that produces a single dominant frequency on the energy spectrum with some harmonics, while the second is an intermittent velocity variation that gives rise to a more broadband response on the spectrum. Results showed that during the sinusoidal velocity profile, there was a reduced slope of the lift polar and a smoother transition to stall when compared with the laminar and intermittent profiles.

Although the versatility of active grids has been readily demonstrated in the literature, there is limited information available on their capabilities to produce periodic flows, especially in comparison with more conventional methods of mean flow modulation, such as shutter systems. Therefore, this provided the motivation to study the capabilities of an active grid in generating periodic flows, and to assess its effectiveness in creating a test environment for experiments in unsteady aerodynamics. Additionally, superimposing periodic variations in velocity on turbulent fluctuations will provide insight into how the
turbulence would react under the action of an unsteady freestream velocity. This links directly to the second portion of this thesis, and so the current research in unsteady turbulence and its relationship to this work will be illustrated in the next section.

2.3 Unsteady Turbulence

In the well-known Richardson-Kolmogorov cascade, turbulent kinetic energy gets transferred from the larger scales of the fluid, or the mean flow, through an intermediate scale (the Taylor microscale), then down to smaller scales where it gets dissipated into heat once it reaches a small enough scale, known as the Kolmogorov scale. During this process, it is assumed that the turbulence is in equilibrium, where the rate at which kinetic energy is supplied by the mean flow is equal to the dissipation rate at the small scales. As a consequence of this equilibrium, the dissipation rate was shown to scale as

$$\epsilon = C_\epsilon \frac{\left(\sqrt{u'^2}\right)^3}{L},$$

by Taylor (1935). In (2.6), $\sqrt{u'^2}$ is the root mean square of the turbulent velocity fluctuations, $\epsilon$ is the turbulent kinetic energy dissipation rate, $L$ is the integral length scale, which is considered to be representative of the largest eddies in the flow, while $C_\epsilon$ is a constant.

As reported by Vassilicos (2015), there has been considerable support for the fact that $C_\epsilon$ is a constant through both direct numerical simulations (DNS) and experiments. The findings compiled by Sreenivasan (1984) from literature showed that $C_\epsilon$ asymptoted to a constant value at local Reynolds numbers, $Re_\lambda$, greater than 100 (defined by the Taylor microscale and the root mean square of velocity fluctuations). This was further confirmed by Sreenivasan (1995) who showed that $C_\epsilon$ remains constant in homogeneous shear flows. Similar results were obtained using DNSs by Sreenivasan (1998) and Kaneda et al. (2003), who implemented a large-scale forcing in their simulations. Sreenivasan (1998) further verified the constant character of $C_\epsilon$ for $Re_\lambda \geq 100$, while Kaneda et al. (2003) showed that $C_\epsilon$ remains constant for $Re_\lambda$ up to 1200.

More recently, Burattini et al. (2005) provided an update on the findings collected by Sreenivasan (1984) by investigating different types of flows, including grid-generated turbulence and wakes of a cylinder and flat plate. The obtained values of $C_\epsilon$ showed appreciable scatter between different test cases (0.5 - 2.5), however were fairly constant for a given flow at $Re_\lambda \geq 60$. Based on these results, and their survey of recent literature,
it was concluded that $C_\epsilon$ does not attain a universal value across different flows and initial conditions.

While the previously discussed studies all agree that $C_\epsilon$ is constant for the higher values of $Re_\lambda$, Sreenivasan (1984) shows that there exists some dependency of $C_\epsilon$ on $Re_\lambda$ for its lower values due to the fact that $C_\epsilon$ is only constant in an asymptotic sense. This was further corroborated by Burattini et al. (2005) in their results. Such variations might imply that $C_\epsilon$ is not a constant as originally postulated by Taylor (1935) in (2.6). Indeed, Seoud & Vassilicos (2007) showed that along the centerline in the turbulence decay region of fractal grids, $C_\epsilon$ shows a dependency on the local Reynolds number given by $C_\epsilon = 98/Re_\lambda$. Mazellier & Vassilicos (2010) also conducted experiments using fractal grids and introduced a wake interaction length $x^*$, which is defined as the distance from the grid at which the wakes from the largest grid bars interact with each other. Based on $x^*$, they defined a production region that extends from the grid to $0.5x^*$, and a decay region beyond $0.5x^*$. Even though the turbulence in the decay region was homogeneous, Mazellier & Vassilicos (2010) found that the scaling in (2.6) was not universal and does not hold in the investigated decay region. Furthermore, $C_\epsilon$ was shown to be a function of the inlet Reynolds number, which is defined by the used grid geometry. Finally, they showed that an increase in Reynolds number causes the $Re_\lambda^{-1}$ scaling effect (Seoud & Vassilicos, 2007) to become more pronounced.

The universality of this new scaling was further investigated by Gomes-Fernandes et al. (2012) who used Particle Image Velocimetry (PIV) to investigate the wake of fractal grids in a water channel. They found that the flow does indeed follow the scaling reported by Seoud & Vassilicos (2007) and Mazellier & Vassilicos (2010), whereby $C_\epsilon$ showed an increase with the inlet Reynolds number, and exhibited clear $Re_\lambda^{-1}$ scaling. Valente & Vassilicos (2012) carried out experiments using three passive grids of varying mesh length and solidity. It was found that within the wake of the passive grids (or more specifically within regions defined by $x^*$), the scaling of $C_\epsilon$ also showed a dependency on the inlet Reynolds number and $Re_\lambda$ similar to that shown previously for fractal grids.

As noted by Vassilicos (2015), several other studies have further confirmed the previously discussed scaling. For instance, Discetti et al. (2013) observed significant variations (up to a factor 4) in the value of $C_\epsilon$ as a function of $Re_\lambda$ using PIV to study the wake of square fractal grids. Additionally, similar results were found by Nagata et al. (2013), who also investigated the turbulent kinetic energy budgeting by producing cross-sectional profiles of advection, transport, production, diffusion and dissipation terms at various streamwise locations in the wake of fractal grids. Furthermore, Isaza et al. (2014) found that increasing the inlet Reynolds number for grid-generated turbulence caused an exten-
sion of the region within which the new $C_\epsilon$ scaling exists. As a result, Vassilicos (2015) goes on to conclude that there has been mounting evidence that the new $C_\epsilon$ scaling is in fact repeatable and does not depend on the type of grid used or on any inhomogeneity or anisotropy incurred by the method of wake production.

Following the presented evidence for the new scaling of $C_\epsilon$, Hearst & Lavoie (2014) designed a fractal grid which allows for a decay region that extends far downstream, in an effort to investigate whether $C_\epsilon$ follows the classical scaling at those locations. It was shown that the decay of turbulence sufficiently downstream of the grid is in agreement with results for regular passive grids. Closer to the grid, $C_\epsilon$ showed a rapid increase in value before becoming constant at distances far from the grid. In fact, this provides evidence that the constancy of $C_\epsilon$ is an asymptotic phenomenon that is associated with the flow reaching an equilibrium. This issue was further investigated for regular passive grids by Isaza et al. (2014) who identified a “near-field” region extending from $x/M = 6$ to $x/M = 12$ where the new $C_\epsilon$ scaling was present. This region corresponds to the initial decay period of turbulence, and is defined by $1.1 < x/x^* < 2.3$. Furthermore, Hearst & Lavoie (2014) and Isaza et al. (2014) both agree that there is no fundamental violation of the classical turbulence theory, instead in regions close to the grid, this new scaling is a transient process that the turbulence goes through until it fully decays and reaches equilibrium in the far downstream regions of the grid.

Furthermore, the dissipation law (2.6) assumes that the proposed energy equilibrium cascade extends to the largest scales, $L$, in the flow. However, Vassilicos (2015) suspects that this assumption might be at fault given the documented departures from (2.6), and that there probably exists some form of energy cascade, other than the Richardson-Kolmogorov cascade, which dictates the transfer of energy within the flow.

In order to further investigate this notion, Goto & Vassilicos (2015) used DNSs of forced and decaying turbulence in order to obtain a scaling for the interscale energy flux. Two cases were simulated, one involving a steady spatially periodic forcing, while the other involved switching off the forcing at the maximum dissipation and letting the turbulence decay. First, results showed that the dissipation rate followed the new scaling discovered by Seoud & Vassilicos (2007) for both cases. Furthermore, it was shown that during the forced turbulence case, values of the energy flux and dissipation are not equal but did fluctuate around an equilibrium state where they would be equal. On the other hand, in the decaying turbulence case, dissipation and energy flux were never equal. Additionally, it was found that in both cases the energy flux scaled in an identical manner to dissipation, which in conjunction with the discussed observations, implies that there is a departure from the equilibrium assumption of (2.6) and hints at the presence
of a different energy cascade, where the turbulence is not at equilibrium.

The departure from classical scaling was further investigated in Goto & Vassilicos (2016a) through a DNS with imposed spatially periodic forcing. It was shown that the flow reacts to the forcing through quasi-periodic fluctuations in dissipation rate, $C_\epsilon$, and all length scales. Furthermore, at high wavenumbers, the dissipation was equal to the energy flux, while a discrepancy between the two was observed at the lower wavenumbers (close to $L^{-1}$). This represented a direct contradiction to the presence of any equilibrium between the dissipation and energy flux at the low wavenumbers, and therefore presents evidence to the claim made by Vassilicos (2015) that the deviation from the classical dissipation law is due to the fact that the energy equilibrium does not extend to the integral length scales of the flow.

In this work, the results from DNSs will be extended in order to theoretically and experimentally investigate the relationship between the unsteady mean flow and turbulence response, while also assessing any deviations from the classical scaling of Taylor (1935) due to the perturbed state of the flow under action of the periodic velocity.
Chapter 3

Experimental Setup

3.1 Wind Tunnel

The wind tunnel used to conduct all experiments is the low-speed recirculating wind tunnel in the Flow Control and Experimental Turbulence (FCET) lab at the University of Toronto Institute for Aerospace Studies. The wind tunnel has a hexagonal cross-sectional area that is 1.2 m wide and 0.8 m high. The total test section length is 5 m and is made up of two modular sections, each 2.5 m in length. The corners of the cross-section are manually adjustable and allow for the approximation of a zero pressure gradient along the length of the test section. The fan speed is set using a variable frequency drive, with the maximum frequency limited to 50 Hz. Without any grids in place, the maximum reported freestream speed is 40 m/s with a turbulence intensity of 0.08% (Zilli, 2017). A schematic of the tunnel is shown in Figure 3.1.

The length and area of each of the tunnel sections are given in Table 3.1, where \( x \) denotes the distance measured in the streamwise flow direction from the beginning of a section. Quantities were either reported from available schematics or measured firsthand.
Figure 3.1: Recirculating wind tunnel schematic, adapted and modified from Hearst (2015). Section numbers correspond to Table 3.1.

Table 3.1: Tunnel section lengths and areas.

<table>
<thead>
<tr>
<th>Number</th>
<th>Section</th>
<th>Length (m)</th>
<th>Area (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Fan Section</td>
<td>5.2</td>
<td>π</td>
</tr>
<tr>
<td>2</td>
<td>Diffuser 1</td>
<td>3.3</td>
<td>1.4x + 3.3</td>
</tr>
<tr>
<td>3</td>
<td>Plenum 1</td>
<td>9.5</td>
<td>7.8</td>
</tr>
<tr>
<td>4</td>
<td>Contraction</td>
<td>3.6</td>
<td>0.02x⁴ + 0.16x³ − 1.56x² + 0.26x + 8.7</td>
</tr>
<tr>
<td>5</td>
<td>Test Section</td>
<td>5</td>
<td>0.9</td>
</tr>
<tr>
<td>6</td>
<td>Vented Junction</td>
<td>1.2</td>
<td>0.9</td>
</tr>
<tr>
<td>7</td>
<td>Diffuser 2</td>
<td>2.2</td>
<td>0.59x + 0.9</td>
</tr>
<tr>
<td>8</td>
<td>Plenum 2</td>
<td>6.2</td>
<td>2.2</td>
</tr>
<tr>
<td>9</td>
<td>Heat Exchanger 1</td>
<td>1.5</td>
<td>1.5x + 2.2</td>
</tr>
<tr>
<td>10</td>
<td>Heat Exchanger 2</td>
<td>2</td>
<td>−0.68x + 4.5</td>
</tr>
</tbody>
</table>
3.2 Active Grid

The active grid used in this study was designed by Hearst & Lavoie (2015), and is shown here in Figure 3.2. It uses a double-mesh design where half of the wings are mounted onto one plane, and the other half are mounted on the other in an alternating pattern. The streamwise separation between the two meshes is 40 mm. A total of 254 solid square wings are mounted onto rods which are 6.35 mm in diameter and spaced 80 mm apart resulting in a mesh length $M = 80$ mm. The total number of rods is 50, with 20 horizontal and 30 vertical rods evenly divided between the two meshes. Each rod is connected to an Applied Motion Products STM23S-3RN stepper motor that is driven through serial commands from 2 RS-485 serial ports. Motors were driven using their “stepping” mode as that proved to be more precise in executing the commanded motions. Each serial port controls 25 motors that are also evenly distributed between the front and aft meshes. In accordance with best practices for the operation of stepper motors, the inertia of the connected rod, wings and coupling was calculated and specified in the motor software.

![Figure 3.2: Active grid mounted to test section, stepper motors are shown around the perimeter of the grid in black.](image)

The grid was operated in one of 4 actuation modes shown in Figure 3.3 and detailed as follows:

- **Mode 0**: The wing frequency is specified and wings complete full rotations at the desired frequency.
- **Mode 1**: Continuous open/close motion at a specified frequency and flapping angle $\beta$, where $\beta = 90^\circ$ corresponds to a “closed” orientation.
• Mode 2: Instantaneous open/close motion at a specified frequency and wing angle, with stop times enforced when the wings are in an open orientation, similar to Reinke et al. (2017).

• Mode 3: User defines opening and closing times, in addition to time that grid remains open or closed.

Modes 0 and 1 were chosen because they represent the basic actuation modes that are typically used in the literature to generate unsteady flows using the previously discussed shutter systems. Specifically, Mode 1 was proposed as a possible active grid alternative to the $\beta$ time profiles used by Greenblatt (2016) and the variable shutter widths used by Miller & Fejer (1964). Furthermore, Mode 2 was chosen in order to approximate a step change in the freestream velocity, while Mode 3 was implemented in order to generate a purely sinusoidal flow profile using the method reported by Greenblatt (2016).

The active grid was placed in the middle of the test section to facilitate taking upstream and downstream measurements. Upstream measurements were recorded at $x/M = 17$ from the grid, while downstream measurements were taken at $x/M = 26$.

![Active grid flapping angle variation](image)

Figure 3.3: Active grid flapping angle variation as a function of time for different operating modes.

### 3.3 Instrumentation

Velocity measurements were taken using constant temperature anemometry. The anemometers were designed and manufactured at the University of Newcastle (Miller et al., 1987)
and an overheat ratio of 1.6 was used to ensure the longevity of the hot wires, as recom-

mended by the manufacturer. Dantec single-wire probes and probe holders were used,
and the hot wires were made in-house with a sensing length of 1 mm and a diameter of
5µm. Calibration of the hot-wires was done in-situ with the grid in a fully open configura-
tion. 12 reference velocities were used for calibration and were obtained using pitot-static
tubes placed upstream and downstream of the grid. The velocities and hot-wire voltages
measured were fit with a fourth order polynomial, but it was also verified that a King
law fit produced acceptable values for the exponent \( n \), which ranged from 0.39 to 0.42.
Calibrations were done before and after each run for most of the runs, and the final
measured velocities were obtained using time-weighted linear interpolation between the
two calibrations in order to minimize any drift.

The pitot-static tubes used in the calibration were connected to a 10 Torr MKS
Instruments pressure transducer to obtain the reference velocities. The static pressure
drop across the grid was also measured from the difference between the static pressure
ports of the pitot-tubes. Temperature throughout the tests was recorded using an Omega
T-type thermocouple. A Velmex XSlide traverse was used for positioning the hot-wire
probe. A picture of the setup taken downstream of the grid is shown in Figure 3.4. A
similar setup was placed simultaneously upstream.

![Figure 3.4: Downstream instrumentation.](image)

Data was acquired through differential sampling (except for thermocouple data which
was acquired using non-refenced single ended sampling) using a National Instruments
(NI) PCIe-6259 data acquisition card and an NI BNC-2110 connector block. The acquired
data was amplified using the “Filter-Amplifier” module of the University of Newcastle
anemometers in order to minimize digitization errors. The appropriate filtering frequency
was determined by conducting sample runs without any filtering and identifying the position of the noise floor on the power spectral density. Consequently, runs at the mean speeds of 4 m/s and 7 m/s were filtered at 2.8 kHz and sampled at 8 kHz, while runs at 10 m/s and 13 m/s were filtered at 5.2 kHz and sampled at 12 kHz.

### 3.4 Uncertainty Analysis

The total uncertainty on the measured quantities was estimated using the quadrature addition of bias and random uncertainties. Bias uncertainties were calculated using the methodology presented by Jørgensen (2002) for hot-wire measurements. On the other hand, the random uncertainty on the mean flow velocity is given by

\[ \delta \bar{u} = 1.96 \frac{\sigma_u}{\sqrt{N}}, \]  

which is a well-documented result derived from the central limit theorem, assuming a 95% confidence interval. In (3.1), \( \sigma_u \) is the standard deviation of the original velocity signal \( u(t) \), and \( N \) is the number of independent samples in the measurement. The calculation of \( N \), as shown by George et al. (1978) is given by

\[ N = \frac{t_s}{2t_x}, \]  

where \( t_s \) is the total sampling time, and \( t_x \) is the integral time scale. Furthermore, the integral time scale may be found by integrating the velocity autocorrelation with respect to time (Benedict & Gould, 1996). Similarly, for phase averaged quantities, random uncertainties are estimated using (3.1), but with the standard deviation taken over a phase averaging window.

Uncertainty on the root mean square (RMS) value of the velocity fluctuations, which is used for calculating turbulence intensity, is given by Benedict & Gould (1996) \textit{viz}.

\[ \delta u_{\text{RMS}} = 1.96 \sqrt{ \frac{u'^2}{2N} }. \]  

Furthermore, the bootstrapping algorithm (Benedict & Gould, 1996) was used to calculate uncertainties on other quantities, such as derivatives. Finally, error propagation rules were employed in order to calculate the uncertainties on quantities that are dependent on the uncertainties identified in this section. Sampling times ranged from 10 to 35 minutes for each test case depending on the desired frequency of the produced flow,
where higher frequencies required a shorter time for the statistics to converge. Similar sampling times were reported by Tardu et al. (1994), who investigated the effect of imposing periodic velocity oscillations on turbulent channel flows using a water channel. They found that 15 to 25 minutes were enough for the statistical convergence of velocity and turbulence intensity phase averages. The maximum total uncertainties corresponding to the slowest converging cases are ±7.2% on the mean velocity, ±7.8% on the RMS of velocity fluctuations and ±8.7% on the peak velocity. Uncertainties are represented here either as error bars on the corresponding figures or mentioned in figure captions to reduce clutter.
Chapter 4

Unsteady Flow Analysis

4.1 Triple Decomposition

Sample velocity measurements taken during the previously discussed grid actuation modes are presented in Figure 4.1. The periodicity of the flow both upstream and downstream of the grid is clearly shown. Furthermore, downstream measurements exhibit the same oscillatory behavior observed upstream but with superimposed stochastic turbulent fluctuations. Therefore, this observation suggests the use of the triple decomposition method as a means to proceed with analyzing the data.

Figure 4.1: Sample velocity measurements taken upstream (red) and downstream (blue) of the grid at a mean flow velocity of 4 m/s and frequency of 1 Hz with $\beta = 90^\circ$ and 50% open times for Modes 2 and 3.
Similar to Reynolds decomposition whereby a signal is broken down into its mean and fluctuating components, the triple decomposition method goes a step further and decomposes an oscillating signal into its mean, fluctuating and periodic components (Hussain & Reynolds, 1970). The velocity is then expressed as

$$u(t) = \bar{u} + \tilde{u}(t) + u'(t), \quad (4.1)$$

where $\tilde{u}$ represents the periodic component and $u'$ represents the fluctuating or turbulent component. $\bar{u}$ is the time average given by

$$\bar{u} = \frac{1}{t_s} \int_0^{t_s} u(t) dt. \quad (4.2)$$

The periodic component $\tilde{u}(t)$ is defined by the phase average viz.

$$\tilde{u}(t) = \frac{1}{N_T} \sum_{n=0}^{N_T} u(t + nT), \quad (4.3)$$

where $T$ represents the period of the signal under consideration, and $N_T$ is the total number of periods within $t_s$. Finally, the fluctuating component is obtained by subtracting the mean and periodic components from the total velocity viz.

$$u'(t) = u(t) - \bar{u} - \tilde{u}(t). \quad (4.4)$$

While the evaluation of $\bar{u}$ is a straightforward averaging process, the calculation of $\tilde{u}(t)$ first requires the identification of the periods of the velocity signal. This may be done using a reference signal that possesses the same frequency content as the original signal. Following the recommendation presented in Ostermann et al. (2015), the autocorrelation of $u(t)$ was used as the reference signal since it takes into account the local mean and root mean square values, thereby making the phase averaging process less sensitive to noise or amplitude variations. Figure 4.2 shows a typical autocorrelation of $u(t)$. Additionally, it was verified that the frequency of the generated autocorrelation matches that of the original velocity signal.

The oscillation periods were defined using the “zero-crossing” method, where each zero crossing signifies the start of a period. Once all the periods are identified, a phase angle is assigned to each time instant according to

$$\phi = 360^\circ \times \frac{t - T_i}{\Delta T}, \quad (4.5)$$
where $t$ refers to a specific instant of the measured velocity, while $T_i$ and $\Delta T$ refer to the starting time and duration of the period containing the time instant $t$, respectively. The measured velocities are then sorted based on their assigned phase angle values and averaged within a window size that adequately captures the velocity variations. Therefore, the described process results in an averaged period of the flow velocity, as shown in Figure 4.3. In order to generate the time series of the full periodic component, a Fourier expansion is fit using the method of least squares, to the points obtained from the phase averaging process.

The final result of the triple decomposition process is shown in Figure 4.4. The amplitude of the velocity signal is determined from $\tilde{u}(t)$, while turbulence statistics can be extracted from $u'(t)$. The dominant flow frequency, which arises due to the rotation of the grid wings at a certain desired frequency, can be determined by examining the power spectral density $E(f)$ of $u(t)$. For instance, varying the grid area at 1 revolution per second causes a peak in the power spectral density (PSD) at 1 Hz, as shown in Figure 4.5. Furthermore, the triple decomposition method attenuates the dominant flow frequency as also shown in the spectrum of $u'(t)$ in Figure 4.5.
4.2 Unsteady Flow Capabilities

Using the triple decomposition method presented earlier, the generated unsteady flow parameters can now be examined. Specifically, the effect of input variables, such as mean flow velocity, grid frequency, and grid actuation mode will be discussed.

Upstream and downstream velocity measurements showed similar trends, and so the discussion here will center on the upstream measurements. However, downstream velocity amplitudes showed higher values than those upstream, for a given grid frequency and mean freestream velocity. This was attributed to the presence of a pressure standing wave as a result of the grid motion. As Rennie et al. (2018) points out, an increase in the
upstream static pressure occurs when the active grid wings close, followed simultaneously with a decrease in static pressure downstream of the grid. This causes the pressure disturbances to travel around the tunnel and give rise to a standing wave through their interaction. Furthermore, Rennie et al. (2018) showed that for a recirculating wind tunnel (which is similar in dimensions and configuration to the tunnel in this study) the standing waves directly upstream and downstream of the grid are 180° out of phase with one another. As a result, the static pressure amplitude resulting from the coalescence of the generated upstream and downstream disturbances varies along the length of the tunnel and attains a maximum directly downstream of the grid, as also shown by Rennie et al. (2018). Therefore, the static pressure amplitude upstream of the grid is lower than that downstream. This causes the observed discrepancy in the velocity amplitudes, since it is the fluctuation of the static pressure that gives rise to fluctuations in velocity (Al-Asmi & Castro (1993) and Rennie et al. (2017)).

The effect of the mean freestream velocity on the flow amplitude is shown in Figure 4.6. It is clear that an increase in the freestream velocity causes a consequent increase in the amplitude, as also shown by Malone (1974) in his design of a gust generator. Grid frequency also plays a role in determining the resulting amplitude as shown in Figure 4.7, where the amplitude drops rapidly from 40% of the mean velocity to 5% upon increasing the frequency from 0.1 Hz to 2 Hz, after which it decreases more gradually to 1% at 10 Hz. Similar trends were reported by Rennie et al. (2017) using oscillating louvers downstream of the test section. Rennie et al. (2018) offer an explanation to this behaviour by first defining the resonant frequencies of the tunnel by the time that it takes pressure disturbances to propagate around the tunnel loop, viz.

\[ f_r = \frac{n c}{L_t}, \]  

where \( n \) is the harmonic number, \( c \) is the speed of sound, and \( L_t \) is the tunnel loop length. In this case, the resonant frequency is approximately 9 Hz. Therefore, operating the grid at this frequency will cause the pressures upstream and downstream of the grid to become in phase since by the time information propagates around the loop, the grid would have completed another open/close cycle and would be in the same configuration as when flow information was first transmitted during the previous cycle. As a result, at frequencies approaching this resonant frequency static pressure variations at the entrance of the test section will become increasingly in phase with the conditions leaving the grid. This causes the static pressure difference between the test section and grid exit to decrease, resulting in a reduced ability to drive the flow, which causes the drop in velocity amplitude.
Based on the presented results, it is apparent that there exists a limitation on the amplitude of oscillations that can be generated for a given freestream velocity and grid frequency. This limitation is addressed by operating the grid in Mode 1, where the flapping angle $\beta$ determines the resulting amplitude, as shown in Figure 4.8. It is important to note here that a flapping angle of $90^\circ$ results in an amplitude equal to that obtained in Mode 0, and so the two modes are effectively equivalent. Furthermore, for $0^\circ \leq \beta \leq 90^\circ$, the resulting grid area is given by

$$A_g = A_\infty - \frac{N_w d^2}{2} \sin \beta,$$

where $A_\infty$ is the test section area, $N_w$ is the number of grid wings, and $d$ is the diagonal of an individual wing. As a result, a higher flapping angle causes the effective test section area to decrease, which results in an amplification of the velocity, assuming the flow is incompressible. This is in agreement with the conclusion reached by Rennie et al. (2017) who modelled the effect of downstream louvers on test section flow velocity. Therefore, Mode 1 offers the capability to adjust the flow amplitude while maintaining a constant frequency and mean velocity.

Modes 2 and 3 allow for further customization of the resulting velocity time series. Depending on the frequency at which Mode 2 is operated, the velocity response changes from that of a typical step response at the lower frequencies to a more triangular response as the frequency is increased. This is due to the fact that Mode 2 essentially involves two step changes in wing position, one to open them and the other to close them. As a result, at higher frequencies the time between these two-step inputs decreases and there-
Figure 4.8: Variation of upstream flow amplitude with flapping angle for Mode 1 at a grid frequency of 1 Hz and a mean velocity of 4 m/s.

Therefore the flow does not have enough time to settle, which causes the observed triangular approximation. The phase-averaged velocity for these cases is shown in Figure 4.9.

Figure 4.9: Comparison of velocity response for Mode 2 at a grid frequency of (a) 0.1 Hz, (b) 0.5 Hz and (c) 1 Hz.

Another parameter that allows further modifications to the resulting time series is the percentage of the oscillation period, $T$, during which the wings remain open. It can be seen in Figure 4.10 that changing the open time enables the customization of the peak velocity location, where $0.5T$ corresponds to the wings remaining open for 50% of the period before being closed. During a $0.5T$ open time, the peak occurs halfway through the period at $\phi = 180^\circ$, which corresponds to the time when the wings are closed. Similarly, for $0.25T$ and $0.75T$, the peaks occur at about a quarter and three-quarters of the period respectively.

Mode 3 was used to approximate a purely sinusoidal flow velocity. As mentioned by Greenblatt (2016), the velocity response time scales involved in opening and closing the grid wings are different, therefore it is necessary to compensate for these differences in
order to produce a perfectly sinusoidal flow velocity. This can be done by individually varying the open duration, closed duration, opening speed and closing speed of the grid wings in order to generate the flow velocity seen in Figure 4.11.

Figure 4.10: Effect of open time on velocity time series for Mode 2, presented here for a grid frequency of 1 Hz.

Figure 4.11: Sinusoidal flow velocity approximation using Mode 3 with a frequency of 0.09 Hz.

Figure 4.12: Comparison of unsteady flow conditions used in literature to those produced by the active grid.

The capabilities of the active grid are now assessed by comparing the produced flow parameters to those typically used for experiments in unsteady aerodynamics, such as those conducted by Strangfeld et al. (2016), Grandlund et al. (2014) and Yang et al. (2017). Figure 4.12 shows the operating envelope of the active grid resulting from the conducted experiments. The reduced frequency was calculated based on a chord length
of 0.25 m which is typical of experiments conducted in the FCET recirculating wind tunnel. Furthermore, the lower limit of the envelope represents the flow obtained using Mode 1 at $\beta = 30^\circ$, while the upper limit was obtained using the complete rotations of Mode 0. Therefore, any desired combination of $k$ and $\sigma$ within the shown limits may be obtained by actuating the grid at a suitable $\beta$. It is shown that the present setup successfully produces an appreciable portion of the conditions used in literature. The conditions that lie outside the envelope shown in Figure 4.12 are a result of the smaller tunnel dimensions used in the cited studies. For example, Strangfeld et al. (2016) used an open-circuit wind tunnel whose test section area is 0.6 m$^2$ (compared to 0.9 m$^2$ in this study) and has a total length that is less than the total loop length of the current tunnel. Additionally, Grandlund et al. (2014) used a closed-circuit wind tunnel with a loop length of 30.2 m (compared with 40 m of the present tunnel), and a test section area of 0.36 m$^2$. That tunnel also offers the capability to further increase the amplitude for a given frequency through the use of a breather as previously explained (Rennie et al., 2017). Finally, Yang et al. (2017) used a multi-fan array in an open-circuit tunnel which offers further capabilities to produce customized oscillations. The reason that smaller tunnels produce higher amplitudes is due to the smaller volume of air that has to be accelerated and decelerated by the fan. This will be shown more formally in the following section.

4.3 Dynamic Model

In order to better understand the results presented in the previous section and predict the resulting flow conditions under inputs different from those tested experimentally, it is helpful to model the active grid and wind tunnel system. Therefore in this section, the dynamic model presented by Greenblatt (2016) for an open-ended blowdown wind tunnel will be extended to reflect the current experimental setup.

To begin with, the tunnel is unfolded in a manner similar to that done by Rennie et al. (2017). Since the flow in the test section is of interest, the cut is made at a point just after the active grid. A schematic of the unfolded tunnel is shown in Figure 4.13, where the different sections are named according to the following convention: test sections are given the name $t$, the active grid is $g$, plenums are $p$, diffusers are $d$, the contraction is $c$, the fan section is $f$, the vented junction is $j$, and heat exchanger sections are $h$.

Assuming the flow is incompressible, the unsteady Bernoulli equation is integrated along a streamline that runs along the entire wind tunnel circuit \textit{viz}.

$$\rho \int_0^g \frac{\partial u(t)}{\partial t} dx + \frac{1}{2} \rho(1 + k_i) \int_0^g du(t)^2 + \int_0^g dp(t) = 0, \quad (4.8)$$
where \( u(t) \) was defined in (4.3), \( k_l \) is the secondary head losses, \( p(t) \) represents the pressure and \( \rho \) is the density. \( u'(t) \) is neglected in the current analysis since the purpose of this model is to predict the amplitude and frequency of the resulting flow, which are both pertinent to \( \tilde{u}(t) \). Therefore, in what follows, the expression \( A' \) will be used as a shorthand notation for the derivative of the quantity \( A \) with respect to time.

Realizing that \( \bar{u} \) is constant in time, and dropping the time notation for simplicity, (4.8) reduces to

\[
\rho \int_0^g \frac{\partial \tilde{u}}{\partial t} \, dx + \frac{1}{2} \rho(1 + k_l) \int_0^g du^2 + \int_0^g dp = 0. \tag{4.9}
\]

The integral appearing in the first term of (4.9) is evaluated across the different tunnel sections \( v\text{ì}z. \)

\[
\int_0^g \frac{\partial \tilde{u}}{\partial t} \, dx = \tilde{u}'_{t2}L_{t2} + \tilde{u}'_jL_j + \int_j^{d2} \tilde{u}'_{d2} \, dx + \tilde{u}'_{p2}L_{p2} + \int_{p2}^h \tilde{u}'_h \, dx + \tilde{u}'_fL_f + \int_f^{d1} \tilde{u}'_{d1} \, dx + \tilde{u}'_{p1}L_{p1} + \int_{p1}^c \tilde{u}'_c \, dx + \tilde{u}'_{t1}L_{t1} + \tilde{u}'_gL_g, \tag{4.10}
\]

where \( L \) represents the length along the centerline of the different sections. The integral is retained in some of the terms of (4.10) since the flow velocity inside the contraction, diffusers and heat exchanger section is dependent on the streamwise location \( x \). The next step is to express all the velocity terms appearing in (4.10) in terms of the test section velocity. The grid area is also broken down into a mean and fluctuating component according to

\[
A_g = \bar{A}_g + \tilde{a}_g. \tag{4.11}
\]

Since the flow rate is conserved in all sections of the tunnel, the average and total flow rates may be respectively expressed as

\[
\bar{q} = A_{t2}\bar{u}_{t2} = A_j\bar{u}_j = A_{d2}(x)\bar{u}_{d2} = A_{p2}\bar{u}_{p2} = A_h(x)\bar{u}_h = A_f\bar{u}_f = A_{d1}(x)\bar{u}_{d1} = A_{p1}\bar{u}_{p1} = A_c(x)\bar{u}_c = A_{t1}\bar{u}_{t1} = \bar{A}_g\bar{a}_g + \frac{\tilde{a}_g\tilde{u}_g}{\tilde{a}_g\bar{a}_g} \tag{4.12}
\]
and

\[ q = \tilde{q} + \bar{q} = A_{t2} \tilde{u}_{t2} + A_{t2} \bar{u}_{t2} = A_j \tilde{u}_j + A_j \bar{u}_j = A_{p2} \tilde{u}_{p2} + A_{p2} \bar{u}_{p2} = A_h(x) \tilde{u}_h + A_h(x) \bar{u}_h \\
= A_f \tilde{u}_f + A_f \bar{u}_f = A_{d1}(x) \tilde{u}_{d1} + A_{d1}(x) \bar{u}_{d1} = A_{p1} \tilde{u}_{p1} + A_{p1} \bar{u}_{p1} \\
= A_c(x) \tilde{u}_c + A_c(x) \bar{u}_c = A_{t1} \tilde{u}_{t1} + A_{t1} \bar{u}_{t1} = \bar{A}_g \tilde{u}_g + \tilde{A}_g \bar{u}_g + \bar{a}_g \tilde{u}_g + \tilde{a}_g \bar{u}_g. \]

(4.13)

Neglecting second order terms, \( \tilde{q} \) may be obtained by subtracting (4.12) from (4.13) to get

\[ \tilde{q} = A_{t2} \tilde{u}_{t2} = A_j \tilde{u}_j = A_{d2}(x) \tilde{u}_{d2} = A_{p2} \tilde{u}_{p2} = A_h(x) \tilde{u}_h = A_f \tilde{u}_f \]

(4.14)

Using (4.14), the velocities in the different tunnel sections may be expressed in terms of the test section velocity \( \tilde{u}_{t2} \) to give

\[ \int_0^g \frac{\partial \tilde{u}}{\partial t} \, dx = L_e \tilde{u}_{t2}' - \frac{\bar{A}_g L_g \tilde{a}_g'}{\bar{A}_g}, \]

(4.15)

where \( L_e \) is defined as an equivalent tunnel loop length and is given by

\[ L_e = A_{t2} \left( \frac{L_j}{A_j} + \int_{j}^2 \frac{1}{A_{d2}(x)} \, dx + \frac{L_{p2}}{A_{p2}} + \int_{p2}^h \frac{1}{A_h(x)} \, dx + \frac{L_f}{A_f} + \int_{f}^{d1} \frac{1}{A_{d1}(x)} \, dx + \frac{L_{p1}}{A_{p1}} + \int_{p1}^c \frac{1}{A_c(x)} \, ds + \frac{L_{t1}}{A_{t1}} + \frac{L_g}{A_g} \right) + L_{t2}. \]

(4.16)

Substituting (4.15) back in (4.9) and evaluating the remaining integral terms results in

\[ \rho L_e \tilde{u}_{t2}' - \rho \bar{a}_g L_g \tilde{a}_g' + \frac{1}{2} \rho (1 + k_l) \left( \bar{u}_g^2 - \bar{u}_{t2}^2 \right) + p_g - p_{t2} = 0, \]

(4.17)

where quantities at \( s = 0 \) have been allocated the subscript \( t2 \) since they represent the conditions inside the test section downstream of the grid. The velocity and pressure terms can now be broken down into their mean and fluctuating components and evaluated \( \text{viz.} \)

\[ \rho L_e \tilde{u}_{t2}' - \rho \bar{a}_g L_g \tilde{a}_g' + \frac{1}{2} \rho (1 + k_l) \left( \bar{u}_g^2 + 2 \bar{u}_g \bar{u}_g + \bar{u}_{t2}^2 - 2 \bar{u}_{t2} \tilde{u}_{t2} - \tilde{u}_{t2}^2 \right) \]

(4.18)

\[ + \bar{p}_g + \bar{p}_g - \bar{p}_{t2} - \bar{p}_{t2} = 0. \]

Time averaging (4.18) and subtracting the time averaged result back from (4.18) results in an equation that represents the dynamics of \( \tilde{u}_{t2} \), and is given by

\[ \rho L_e \tilde{u}_{t2}' + \rho (1 + k_l) (\bar{u}_g \bar{u}_g - \bar{u}_{t2} \bar{u}_{t2}) + \bar{p}_g - \bar{p}_{t2} - \rho \bar{a}_g L_g \tilde{a}_g' = 0. \]

(4.19)
Furthermore, using (4.12) and (4.14) gives the following relations for the mean and oscillating grid velocity:

\[ \bar{u}_g = \frac{A_{t_2}}{A_g} \bar{u}_{t_2}, \tag{4.20} \]

and

\[ \tilde{u}_g = \frac{A_{t_2}}{A_g} \bar{u}_{t_2} - \frac{\bar{u}_g}{A_g} \tilde{a}_g. \tag{4.21} \]

These are then substituted in (4.19), and after some simplifications result in

\[ \rho L_e \ddot{u}_{t_2} + \rho (1 + k_l) \left( \frac{A_{t_2}^2 - A_g^2}{A_g^2} \right) \bar{u}_{t_2} \bar{u}_{t_2} = \rho (1 + k_l) \left( \frac{A_{t_2}^2}{A_g^3} \right) \tilde{u}_{t_2}^2 \tilde{a}_g + \tilde{p}_{t_2} - \tilde{p}_g + \rho \frac{\bar{u}_g L_g \tilde{a}_g}{A_g}, \tag{4.22} \]

Dividing (4.22) by \( \rho (1 + k_l) \bar{u}_{t_2}^2 \) gives

\[ \frac{L_e}{(1 + k_l) \bar{u}_{t_2}^2} \ddot{u}_{t_2} + \left( \frac{A_{t_2}^2 - A_g^2}{A_g^2} \right) \frac{\bar{u}_{t_2}}{\bar{u}_{t_2}} = \left( \frac{A_{t_2}^2}{A_g^3} \right) \tilde{a}_g + \frac{\tilde{p}_{t_2} - \tilde{p}_g}{\rho (1 + k_l) \bar{u}_{t_2}^2} + \frac{\bar{u}_g L_g}{(1 + k_l) \bar{u}_{t_2}^2 A_g} \tilde{a}_g. \tag{4.23} \]

The output of the obtained first order system is defined as \( \epsilon = \frac{\tilde{u}_{t_2}}{\bar{u}_{t_2}} \), and after some simplifications (4.23) is expressed as

\[ \left( \frac{A_g^2}{A_{t_2}^2 - A_g^2} \right) \frac{L_e}{(1 + k_l) \bar{u}_{t_2}} \epsilon' + \epsilon = \left( \frac{A_{t_2}^2}{A_{t_2}^2 - A_g^2} \right) \tilde{a}_g + \left( \frac{A_g^2}{A_{t_2}^2 - A_g^2} \right) \frac{\tilde{p}_{t_2} - \tilde{p}_g}{\rho (1 + k_l) \bar{u}_{t_2}^2} + \left( \frac{A_g^2}{A_{t_2}^2 - A_g^2} \right) \left( \frac{A_{t_2} L_g}{(1 + k_l) A_g^2 \bar{u}_{t_2}} \right) \tilde{a}_g'. \tag{4.24} \]

It is important to take note of several key features of the above system. First, and as also reported by Greenblatt (2016), the tunnel may be considered a variable low pass filter whose cut-off frequency depends on the system’s time constant, which is given by

\[ \tau = \left( \frac{A_g^2}{A_{t_2}^2 - A_g^2} \right) \frac{L_e}{(1 + k_l) \bar{u}_{t_2}}. \tag{4.25} \]

The dependency of \( \tau \) on the grid blockage is shown in Figure 4.14. Increasing the blockage, which is synonymous with increasing \( \beta \) and decreasing \( \bar{A}_g \), causes a drop in the time constant and a rise in the cut-off frequency. This translates into an increase in the amplitude of the test section velocity, and is in agreement with the experimental observation shown in Figure 4.8. Similarly, increasing the mean freestream velocity \( \bar{u}_{t_2} \) causes an increase in the velocity amplitude as shown experimentally in Figure 4.6. The effect of
the tunnel size $L_e$ was addressed by Greenblatt (2016), who stated that a larger tunnel (with a greater length and cross-sectional areas) dampens oscillations in the velocity due to the inertia of the air inside of it.

While the presented model was shown to qualitatively predict the results obtained, the focus now shifts to assessing whether this model can accurately predict the performance of the active grid and wind tunnel system quantitatively. Specifically, it is desired to predict the amplitude of the velocity oscillations under a given set of conditions, which includes the freestream speed and the grid mode of operation.

Adopting similar simplifications to those used by Greenblatt (2016), the secondary losses are set to zero, and the input of the system is assumed to be driven solely by the variation in grid area, thus the input to the system reduces to

$$g(t) = \left( \frac{A^2_{r2}}{A^2_{r2} - A^2_g} \right) \frac{\bar{a}_g}{A_g}.$$  \hfill (4.26)

In reality, the input function may differ from the proposed simplification in (4.26) due to the fact that other terms contribute to driving the system, such as the pressure rise across the fan, which is represented by the difference in pressure fluctuations, $\tilde{p}_{r2} - \tilde{p}_g$, across the tunnel circuit as shown in (4.24). However, the magnitude ratio is of concern in
this case, which is the ratio of the output to input, and is therefore representative of the relative relationship between the applied input and the corresponding output. In order to compare the theoretical response of the system to that obtained from experiments, Greenblatt (2016) proposed an “experimental” magnitude ratio defined by

$$M_e = \frac{\tilde{u}_p}{\tilde{u}_s},$$

where $\tilde{u}_p$ represents the peak amplitude obtained from the velocity measurements while the grid is running, and $\tilde{u}_s$ is the static amplitude which is obtained by calculating the difference in velocities when the grid is static at $\beta = 0^\circ$, and when the grid wings are set to a specific $\beta$ corresponding to one of the flapping angles from Mode 1. Figure 4.15 shows that the experimental results are in good agreement with the model. Some discrepancy was first observed for the fully closed case (i.e. when $\beta = 90^\circ$). This is due to the fact that in reality, when the grid is closed, the free cross sectional area is not equal to zero due to the spaces between the grid wings, and the spacing between the front and back meshes of the grid. Therefore, setting $\beta = 80^\circ$ as a means of accounting for those open areas, provides a better prediction of the resulting amplitude.

![Figure 4.15: Comparison of experimental magnitude ratio with that predicted from experiments for different operating modes and flow frequencies, plotted as $\omega = 2\pi f$. Maximum error on $M_e$ was 6%.

Furthermore, plotting the magnitude ratio from the different runs against the non-dimensional flow frequency $\omega\tau$ (Greenblatt, 2016) results in the collapse of data from all the runs as shown in Figure 4.16. In fact, the resulting curve may be used as a guide in order to predict the flow amplitude for a given set of experimental conditions, all of which
Figure 4.16: Collapse of data from all experimental runs when plotted against non-dimensional flow frequency $\omega \tau$. Maximum error on $M_\epsilon$ was 6%.

Figure 4.17: Static amplitude map for different test conditions. These amplitudes are represented in the non-dimensional grid frequency. Using the static amplitude map shown in Figure 4.17, it is possible to find $\tilde{u}_s$ and then determine the peak amplitude $\tilde{u}_p$ during grid operation by using (4.27).

### 4.4 Turbulence Considerations

After characterizing the oscillating flow produced by the active grid, the focus now shifts to the analysis of $u'(t)$. Figure 4.18 shows a comparison between the power spectral
density (PSD) of the total velocity and the turbulent component downstream of the grid. The effectiveness of the triple decomposition method is noted by the observation that the dominant flow frequency was attenuated in $u'(t)$ for the tested frequencies.

It is important here to note the presence of some residual harmonics in the PSD spectrum at flow frequencies of 1 Hz and 4 Hz. The effect of these peaks and their contribution to the overall energy is better illustrated by considering the premultiplied spectra $fE(f)$, shown in Figure 4.19, which is representative of the energy distribution across different frequencies. It is apparent that the energy from the dominant flow frequency has been attenuated, however some of the residual harmonics still contribute to the flow energy. These peaks accounted for 1% and 3% of the total energy for the 1 Hz and 4 Hz cases, respectively. Therefore, it is expected that these peaks will alter the turbulence statistics but not significantly due to their low contribution to the overall energy. Similar peaks arising from rod rotation were noted by Mydlarski & Warhaft (1996)
when operating their active grid in synchronous mode. In their case, the observed peaks contributed to 6% of the total energy and the turbulence statistics were shown not to be affected when comparing the results with the random grid operation mode. Furthermore, the effect of these harmonics on the dissipation scales may be studied by considering the dissipation spectra given by \( f^2 E(f) \). It can be seen in Figure 4.20 that the contribution of these peaks to the dissipation spectrum is minimal, and so the dissipation scales are not affected.

![Figure 4.19](image1.png)  
Figure 4.19: Comparison between the premultiplied spectra of the total velocity component and periodic components at flow frequencies of (a) 1 Hz and (b) 4 Hz.

![Figure 4.20](image2.png)  
Figure 4.20: Comparison between the dissipation spectra of the total velocity component and periodic components at flow frequencies of (a) 1 Hz and (b) 4 Hz.

Another observation from Figure 4.18 is the presence of a high frequency peak at 800 Hz which appears on the spectra for flow frequencies of 4 Hz and 8 Hz. The appearance of this peak at the higher grid frequencies suggests that the pressure disturbances resulting
from grid actuation have excited some acoustic mode of the tunnel. This may be verified by modelling the tunnel as a rectangular cavity whose dimensions correspond to the loop length, width and height of the cross-sectional area of the test section. Furthermore, the allowed vibration angular frequencies of a rectangular cavity are given by Kinsler et al. (1999) as

\[ \omega_{lmn} = c \left[ (l\pi/L_x)^2 + (m\pi/L_y)^2 + (n\pi/L_z)^2 \right]^{1/2}, \quad (4.28) \]

where \( l, m \) and \( n \) correspond to the mode numbers, \( c \) is the speed of sound while \( L \) denotes the dimensions of the tunnel in the \( x, y \) and \( z \) directions. Interestingly, the third mode of the frequencies in (4.28) is equal to 773 Hz, which is close to 800 Hz, and implies that the observed peak is the result of an acoustic excitation of the tunnel modes.

Upstream of the grid, the PSD for \( u'(t) \) showed significant residual peaks from the periodic component, especially at the higher grid frequencies. An example of these peaks is shown in Figure 4.21. The majority of these peaks arise due to the subtraction of the periodic component from the total velocity. This may be verified by comparing the resulting \( u'(t) \) PSD spectrum with the spectrum when the grid is fully open and not moving, which does not show any of the observed peaks in \( u'(t) \).

![Figure 4.21: Comparison of upstream power spectral density spectrum at a mean speed of 4 m/s and grid frequency of 4 Hz for \( u(t) \), \( u'(t) \) and the case when the grid is open and not moving.](image)

The residual peaks were attributed to deviations in the cycle to cycle repeatability of the flow. Such deviations are due to phase jitter of the velocity signal, as in Figure 4.22a, and some drift in the mean value of the velocity over the duration of the run, as shown in Figure 4.22b. For instance, the drift shown in Figure 4.22b was about 0.8%
over a sampling time of 10 minutes and yet resulted in significant contamination of the PSD spectrum as shown in Figure 4.21. This results in the increase of the \( u'(t) \) time scale due to the presence of remnant oscillations that cause the signal to stay correlated for a longer time, which significantly increases the uncertainties associated with the upstream turbulence measurements. Additionally, these peaks cause the overestimation of the upstream turbulence intensity due to an increase in the RMS value of \( u'(t) \), as a result of the residual periodicity. Consequently, the upstream turbulence intensity values showed an increase with freestream velocity, due to the increase in amplitude at the higher velocities, as shown in Figure 4.23. Furthermore, the overestimation of turbulence intensity values is shown for the 4 Hz case at \( Re_M = 2 \times 10^4 \) and \( Re_M = 4 \times 10^4 \).

![Figure 4.22: Comparison between first three flow cycles (blue), middle cycles (red) and last three cycles (green) of an experiment run for (a) mode 1 at \( \beta = 30^\circ \) and 1 Hz and (b) mode 0 at 4 Hz.](image)

On the other hand, Figure 4.24 shows that downstream of the grid for Mode 0, the turbulence intensity undergoes little variation as \( Re_M \) is increased. Furthermore, this trend was observed across all tested frequencies with absolute differences in turbulence intensity not exceeding 1% for different mean speeds at a given grid frequency. This difference is less than the 2.2% reported by Hearst & Lavoie (2015) when operating the same grid in a fully random mode. Those results are included on Figure 4.24 for reference. This is indicative of the fact that during Mode 0, the active grid approximates the behaviour of a passive grid, in that the turbulence intensity stays somewhat constant across different Reynolds numbers due to the spatial homogeneity with which the grid is operated during this mode. Furthermore, the decrease in the turbulence intensity value of Mode 0 when compared with the fully random mode is due to the fact that Mode 0 imposes less disturbances to the flow since the grids are driven at a constant frequency.
Figure 4.23: Variation of upstream turbulence intensity with grid Reynolds number.

and in a constant direction, as opposed to the randomly varying frequency and direction for the fully random mode. In fact, this observation was also reported by Mydlarski & Warhaft (1996) who actuated their active grid using both random and synchronous modes.

Figure 4.24: Variation of downstream turbulence intensity with grid Reynolds number.

The dependence of turbulence intensity on Ro is shown in Figure 4.25. For Ro > 10, or for frequencies less than 4 Hz, turbulence intensity values lie between 6.5% and 7%. This region of constant turbulence intensity was also reported by Hearst & Lavoie (2015), and provides flexibility in choosing a desired mean flow speed and oscillation frequency while maintaining a somewhat constant value of turbulence intensity. For Ro < 10, at frequencies of 8 Hz, there is a noticeable increase in the turbulence intensity. This is
thought to be due to the proximity of the forcing frequency (8 Hz) to the inertial scaling range, which leads to the excitement of the dissipation rate in a manner similar to that studied by Cekli et al. (2010), who found that there exists a grid modulation resonant frequency which leads to the enhancement of the dissipation rate. The aforementioned hypothesis can be investigated by actuating the grid at higher frequencies and studying the effect that the inertial range forcing has on the produced turbulence, however this is beyond the scope of the current work.

The turbulence generated by the active grid during Mode 0 may be better characterized by considering the relevant length scales produced. First, Taylor’s frozen flow hypothesis was used to find the streamwise integral length scale (Kurian & Fransson, 2009) viz.

\[ L = \bar{u} \int_0^\infty R_{u'u'}(t)dt, \]  

(4.29)

where \( R_{u'u'}(t) \) is the autocorrelation function based on \( u'(t) \). Next, the Taylor microscale is given by

\[ \lambda = \sqrt{\frac{\bar{u}^2}{(\partial u'/\partial x)^2}}, \]  

(4.30)

while the dissipation Kolmogorov scale (Kolmogorov, 1941) is

\[ \eta = \frac{\nu^{3/4}}{\bar{\epsilon}^{1/4}}, \]  

(4.31)

where \( \nu \) is the kinematic viscosity and \( \bar{\epsilon} \) is the mean turbulent kinetic energy dissipation rate. The dissipation rate is estimated using the method presented by Mi et al. (2005). Assuming that the turbulence is homogeneous and isotropic and using Taylor’s frozen flow hypothesis, \( \bar{\epsilon} \) is expressed as

\[ \bar{\epsilon} = 15\nu \left( \frac{\partial u'}{\partial x} \right)^2 = \frac{15\nu}{\bar{u}^2} \left( \frac{\partial u'}{\partial t} \right)^2. \]  

(4.32)

Figure 4.26 shows the variation of the aforementioned scales with the grid Reynolds number, and for different grid actuation frequencies. The integral length scale remains constant at the lower grid Reynolds numbers and then increases slightly as the Reynolds number is increased before reaching a constant value again. This is in contrast with the results reported by Hearst & Lavoie (2015), where the length scale showed a continuous increase with increasing grid Reynolds numbers. Additionally, the obtained \( L/M \) values were significantly less than those obtained when the grid was actuated randomly. The reason for this discrepancy in behaviour is due to the fact that when the grid is operated
in the current synchronous mode, its effective mesh length is smaller than during a random mode where rods can become significantly out of phase with one another. This results in the creation of gaps in the grid that are much larger than those created when the grid is operated in synchronous mode when the rods stay in phase with each other. Therefore, the current modes cause less variability in the length scales of the produced turbulence and an overall smaller size of those scales. Similar results were also reported by Mydlarski & Warhaft (1996) using their synchronous mode in comparison with the random mode.

Furthermore, the Taylor microscale shows a slight increase with increasing grid Reynolds number. Such a trend also represents a departure from typical active grid behaviour reported by Hearst & Lavoie (2015) and Larssen & Devenport (2011). In fact, the variation of the microscale shown in Figure 4.26b resembles that produced by the passive grids used in Zilli (2017). This observation further reaffirms the earlier claim that the active grid resembles a passive grid when actuated in a synchronous mode, at least in its inability to produce a large scaling range as also evidenced by the spectra in Figure 4.18. However this is only true in some respects, as operating the grid in Mode 0 still offers the flexibility of slightly adjusting the produced length scales by varying the grid Reynolds number, while maintaining a fairly constant turbulence intensity and flow frequency, as opposed to the nearly constant length scales generated by passive grids (Zilli, 2017). This feature is specifically desirable in unsteady aerodynamics experiments, as it allows the creation of turbulent eddies that are on the order of model chord lengths typically used in such studies. Finally, the variation of the dissipative scales with the grid Reynolds number showed similar behaviour to the results presented by Hearst & Lavoie (2015). Therefore, operating the grid using this mode presents a compromise between passive and active grid behaviour, where certain characteristics of each grid type are observed here. In fact, a parallel may be drawn between this behaviour and that observed by Weitemeyer et al. (2013), who observed a transition between classical and fractal grid behaviour using an active grid that was operated in a manner that allowed for different realizations of locally varying solidity.

While the previous results were presented for full rotations of the grid wings during Mode 0, the flapping angle of Mode 1 also has a pronounced effect on the turbulence intensity. As shown in Figure 4.27, increasing the flapping angle causes the turbulence intensity to increase significantly. This is due to the fact that the higher flapping angles cause greater disturbances to the flow due to their increased blockage and induce greater fluctuations, thereby increasing the turbulence intensity (Hearst & Lavoie, 2015).

The turbulence characteristics may be further customized by using a similar scheme
Figure 4.26: Variation of (a) streamwise integral length scale, (b) Taylor microscale and (c) dissipation scale with grid Reynolds number.
Figure 4.27: Variation of turbulence intensity with flapping angle at a mean speed of 4 m/s.

to that implemented by Hearst & Ganapathisubramani (2017), in that the grid will be spatially divided across different operating modes. In this case, the grid was actuated in a “hybrid” mode, which involves running some of the wings using Mode 0, while the remainder of the wings run using the “Fully Random (FR)” mode developed by Hearst & Lavoie (2015). This involves transmitting a random signal to the motors using a Gaussian distribution that is bound by the desired values for wing rotational rates, cruise times, and rotation directions.

Tests were run at a mean speed of 4 m/s with a Mode 0 frequency of 0.5 Hz and FR frequencies of $3 \pm 2$ Hz and $15 \pm 5$ Hz. Furthermore, the following test codes designate the physical distribution of wings between Mode 0 and FR mode:

- **1S** - 1 serial port uses Mode 0, while the other uses FR, which results in grid wings being evenly divided between the two modes
- **HR** - Horizontal rods are driven in FR
- **HHR** - Half of the horizontal rods are driven in FR
- **VR** - Vertical rods are driven in FR
- **HVR** - Half of the vertical rods are driven in FR

The 1S mode involves sending two closely simultaneous signals to each serial port of the active grid, one commanding a Mode 0 movement, while the other an FR movement. On the other hand, an individual signal is sent to each of the motors for all the other
modes in order to target specific rods. As a result of this, it is expected that the wing movement across the grid will not be uniform. Therefore, it was imperative to check the homogeneity of each of those modes. It can be seen in Figure 4.28 that the VR case shows the most homogeneous turbulence intensity profile taken in the spanwise direction, $y$. This is due to the fact that this mode contains the greatest number of randomly actuated rods, 30, with the remaining 20 being driven by Mode 0. As a result, the time delay involved in targeting individual motors was minimized. Interestingly, this mode also offers a more homogeneous profile when compared to Mode 0, as a result of the random rotation of some of the rods, which is more effective at homogenizing the flow than a correlated mode (Hearst & Lavoie, 2015). Furthermore, the homogeneity of this mode is reflected in the uniformity of the produced velocity amplitude, which was shown to be a function of blockage in (4.25). Figure 4.29 shows the amplitudes obtained using VR, in contrast to Mode 0 and the HHR mode, which has the greatest number of individually targeted rods and the least number of randomly actuated rods.

Therefore, the ability to produce custom turbulence is limited using the current setup. However, the proposed modes show promise in their ability to customize the produced turbulence. For instance, at $y/M = 0$, the turbulence intensity can be varied from 8.5 % to 9.7 %, compared to Mode 0, where turbulence intensity was equal to 6.3 %. However, this comes at the cost of a reduced velocity amplitude since the mean grid area is reduced when the grid is operated in the hybrid mode. Compared to the base case of $\sigma = 0.23$, the amplitude drops to anywhere between 0.11 and 0.19 using the previously mentioned modes. It is also worth noting that the hybrid modes offers the ability to vary the produced scales in accordance with the study carried out by Hearst & Lavoie (2015).
Chapter 5
Investigation of Unsteady Turbulence

5.1 Theoretical Considerations

The previous discussions on the turbulence characteristics of the flow generated by the active grid were focused on time-averaged quantities in an effort to understand the effect of different grid parameters on the turbulence produced. In this chapter, the focus will shift more fundamentally to phase-averaged quantities as a means of understanding the dynamics of turbulence under the action of certain imposed fluctuations. Going back to the triple decomposition of the flow \( \text{viz.} \)

\[ u = \bar{u} + \tilde{u} + u', \tag{5.1} \]

the question is to investigate how \( u' \) reacts to an imposed \( \tilde{u} \), and how the different parameters of the imposed fluctuations govern the response of the turbulent flow component.

Starting from the governing flow equations, namely mass and momentum conservation, Reynolds & Hussain (1972) found that the effect of an organized wave is transmitted to the turbulence field through oscillations of the turbulent Reynolds stresses. However, understanding the dynamics of such oscillations and how they relate to \( \bar{u} \) produces a significant closure problem in which many resulting terms are unknown. As a result, Reynolds & Hussain (1972) recommended investigating the kinetic energy budgeting of the different flow components as means of gaining insight into their behavior.

The change in the turbulent kinetic energy (TKE) will now be investigated. (5.1) is
substituted in the momentum equation, which is given in Einstein notation by

\[
\frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} = - \frac{1}{\rho} \frac{\partial p}{\partial x_i} + \nu \frac{\partial^2 u_i}{\partial x_j^2},
\]

(5.2)

to give

\[
\frac{\partial (\bar{u}_i + \tilde{u}_i + u'_i)}{\partial t} + (\bar{u}_j + \tilde{u}_j + u'_j) \frac{\partial (\bar{u}_i + \tilde{u}_i + u'_i)}{\partial x_j} = - \frac{1}{\rho} \frac{\partial p}{\partial x_i} + \nu \frac{\partial^2 (\bar{u}_i + \tilde{u}_i + u'_i)}{\partial x_j^2},
\]

(5.3)

Multiplying (5.3) by \( u'_i \), expanding the resulting terms and rearranging gives

\[
u ' \bar{u}'_i \frac{\partial \tilde{u}'_i}{\partial t} + \nu ' \tilde{u}'_i \frac{\partial \bar{u}'_i}{\partial t} + \nu ' \bar{u}'_i \frac{\partial u'_i}{\partial t} + \nu ' \tilde{u}'_i \frac{\partial \bar{u}'_i}{\partial x_j} + \nu ' u'_i \frac{\partial \tilde{u}'_i}{\partial x_j} + \nu ' \bar{u}'_i \frac{\partial \tilde{u}'_i}{\partial x_j} + \nu ' \tilde{u}'_i \frac{\partial \bar{u}'_i}{\partial x_j} + \nu ' \tilde{u}'_i \frac{\partial \tilde{u}'_i}{\partial x_j} = - u'_i \frac{1}{\rho} \frac{\partial p}{\partial x_i} + u'_i \nu \left( \frac{\partial^2 \bar{u}'_i}{\partial x_j^2} + \frac{\partial^2 \tilde{u}'_i}{\partial x_j^2} + \frac{\partial^2 u'_i}{\partial x_j^2} \right).
\]

(5.4)

It is important now to note a few assumptions that reflect the present problem and approximate the conditions of the experiments. First, it is assumed that the mean velocity does not vary in time and therefore \( \frac{\partial \bar{u}_i}{\partial t} \) is equal to zero. Furthermore, the mean and periodic velocity are assumed to be constant everywhere due to the spatial homogeneity of the flow, i.e. \( \frac{\partial \bar{u}_i}{\partial x_j} \) and \( \frac{\partial \tilde{u}_i}{\partial x_j} \) are equal to zero. Invoking these assumptions in (5.4) results in

\[
u ' \bar{u}'_i \frac{\partial \bar{u}'_i}{\partial t} + \nu ' \tilde{u}'_i \frac{\partial \bar{u}'_i}{\partial t} + \nu ' \bar{u}'_i \frac{\partial u'_i}{\partial t} + \nu ' \tilde{u}'_i \frac{\partial \bar{u}'_i}{\partial x_j} + \nu ' u'_i \frac{\partial \tilde{u}'_i}{\partial x_j} + \nu ' \bar{u}'_i \frac{\partial \tilde{u}'_i}{\partial x_j} + \nu ' \tilde{u}'_i \frac{\partial \bar{u}'_i}{\partial x_j} + \nu ' \tilde{u}'_i \frac{\partial \tilde{u}'_i}{\partial x_j} = - u'_i \frac{1}{\rho} \frac{\partial p}{\partial x_i} + u'_i \nu \frac{\partial^2 u'_i}{\partial x_j^2}.
\]

(5.5)

Additionally the following simplifications, which can be derived from simple differentiation rules, are implemented:

\[
u ' \bar{u}'_i \frac{\partial \bar{u}'_i}{\partial t} = \frac{\partial}{\partial t} \left( \frac{1}{2} u'_i^2 \right)
\]

(5.6a)

\[
u ' \bar{u}'_i \frac{\partial u'_i}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \frac{1}{2} u'_i^2 \right)
\]

(5.6b)

\[
u ' \frac{\partial^2 u'_i}{\partial x_j^2} = \frac{\partial}{\partial x_j} \left( u'_i \frac{\partial u'_i}{\partial x_j} \right) - \left( \frac{\partial u'_i}{\partial x_j} \right)^2.
\]

(5.6c)
Substituting (5.6) back in (5.5) gives

\[
 u'_i \frac{\partial \tilde{u}_i}{\partial t} + \frac{\partial (\frac{1}{2} u'^2_i)}{\partial t} + \tilde{u}_j \frac{\partial (\frac{1}{2} u'^2_i)}{\partial x_j} + \tilde{u}_j \frac{\partial (\frac{1}{2} u'^2_i)}{\partial x_j} + u'_j \frac{\partial (\frac{1}{2} u'^2_i)}{\partial x_j} =
\]

\[
 -u'_i \frac{1}{\rho} \frac{\partial p}{\partial x_i} + \nu \frac{\partial}{\partial x_j} \left( \frac{\partial (\frac{1}{2} u'^2_i)}{\partial x_j} \right) - \nu \left( \frac{\partial u'_i}{\partial x_j} \right)^2 .
\]

(5.7)

Defining \( K = \frac{1}{2} (u'^2_i) \) as the TKE, its total rate of change with respect to time may be expressed as

\[
 \frac{DK}{Dt} = \frac{\partial (\frac{1}{2} u'^2_i)}{\partial t} + \tilde{u}_j \frac{\partial (\frac{1}{2} u'^2_i)}{\partial x_j}.
\]

(5.8)

Finally, phase averaging and rearranging (5.7), results in the following relationship that describes the phase averaged change in the TKE viz.

\[
 \left\langle \frac{DK}{Dt} \right\rangle = \left\langle -u'_i \frac{1}{\rho} \frac{\partial p}{\partial x_i} - u'_j \frac{\partial (\frac{1}{2} u'^2_i)}{\partial x_j} \right\rangle - \tilde{u}_j \left\langle \frac{\partial (\frac{1}{2} u'^2_i)}{\partial x_j} \right\rangle + \left\langle \nu \frac{\partial}{\partial x_j} \left( \frac{\partial (\frac{1}{2} u'^2_i)}{\partial x_j} \right) \right\rangle
\]

\[
 + \left\langle -u'_i \frac{\partial \tilde{u}_i}{\partial t} \right\rangle - \left\langle \nu \left( \frac{\partial u'_i}{\partial x_j} \right)^2 \right\rangle.
\]

(5.9)

Equation (5.9) is very similar to the transport equation obtained by Reynolds & Hussain (1972), where the first three terms on the right hand side denote the transport of TKE due to pressure, the oscillating velocity and viscosity, respectively. The last term denotes the dissipation rate of TKE.

However, the main result obtained from (5.9) is the second to last term which represents the coupling between the periodic and stochastic components of the flow. Therefore, this term contributes to the production of TKE by means of the interactions between the turbulence and oscillating components of the velocity. Furthermore, it depends on the frequency and amplitude of the imposed periodic fluctuations, through the time derivative term. The effect of these parameters will be illustrated using experimental results in the following section.

### 5.2 Experimental Observations

In order to understand the dynamics of turbulence during actuation of the mean flow, it is helpful to first study the turbulence response to a step input in velocity. As a result, the active grid is actuated using Mode 2 at a mean velocity of 4 m/s and the very low frequency of 0.025 Hz in order to allow enough time for the flow to settle before
proceeding with the actuation. The chosen frequency results in a period of 40 s which means that after a closing or opening motion the grid stays in a constant position for 20 s. This time was deemed sufficient for the flow to settle since the calculated system time constant (based on (4.25)) was 1.1 s and therefore a duration of 20 s equates to \( \approx 20\tau \), which is much greater than the settling time of \( 4\tau \) for a first order system. The corresponding phase averaged velocity and turbulence intensity are shown in Figure 5.1a.

Several things must be pointed out before proceeding. First, it is obvious that the falling and rising response of \( \tilde{u}(t) \) exhibits different response times, even though they were driven at the same opening and closing wing speed. This is a clear indication of the different velocity response when the wings close and open, which was mentioned in section 4.2. The response when the grid closes, seen here at \( 225^\circ \leq \phi \leq 360^\circ \), is faster than when the grid opens at \( 45^\circ \leq \phi \leq 135^\circ \). In fact, this variation in the velocity response proves to be convenient in the present analysis as it allows for the comparison of the response of the turbulence intensities to two different step velocity inputs. It is apparent that the turbulence intensity gets modulated based on the velocity response. Furthermore, during the fast response case when the wings are closing, the turbulence intensity shows significantly more overshoot than the slower case when the wings are opening. A similar response is seen in Figure 5.1b when the grid is operated using Mode 0 at a frequency of 0.1 Hz. Two peaks occur at phases of \( 170^\circ \) and \( 280^\circ \) as a result of the flow accelerating and decelerating respectively. When the velocity is at peak value, the wings start closing and a fast rise in turbulence intensity is observed which then decreases, until the velocity reaches its minimum and the wings start opening again. At that point, the turbulence intensity increases, at a slower rate, then settles to a lower value at a phase of \( 225^\circ \), before the same cycle starts again. Therefore, the rate of change of the velocity plays a crucial role in the response, whereby a slower rate causes the turbulence to respond slowly and with lower overshoot values, while a higher rate induces the opposite behaviour. This is in agreement with the results found in the previous section, where the term \( \langle -u_i \partial \tilde{u} / \partial t \rangle \) alters the TKE budget.
Figure 5.1: Phase variation of the periodic velocity component and turbulence intensity for (a) a step input and (b) Mode 0 at 0.1 Hz for a mean velocity of 4 m/s.
It is now helpful to study the phase-averaged modulation of turbulence intensity under different frequencies of the forcing. First, the two peaks in turbulence intensity, which were discussed earlier, are both present at frequencies of 0.1 Hz and 0.5 Hz as shown in Figure 5.2a and 5.2b, respectively. However, the first peak associated with flow acceleration was less pronounced in the 0.5 Hz compared to the 0.1 Hz case (located at 2% away from steady state compared to 4% at 0.1 Hz). Additionally, this peak was not perceivable at 1 Hz, while also being less pronounced at 2 Hz. On the other hand, the second peak, associated with flow deceleration, was always noticeable across the different frequencies. This is due to the fact that as the frequency of the flow is increased the turbulence has less time to respond to the imposed oscillations and so the slower response gets attenuated, especially since the response to flow acceleration is slower than that for flow deceleration as explained earlier and shown in Figure 5.1a. It is also important to note that the dissipation rate follows the same phase-averaged behaviour as the turbulence intensity whereby the same two peaks are observed and gradually get attenuated as the forcing frequency is increased. Therefore, another effect of increasing the flow frequency is the decrease in the peak-to-peak amplitude of the turbulence intensity and dissipation rate until a “smoother” state is reached at the higher frequencies. Additionally, the phase of the dissipation with respect to velocity oscillations gets modulated by the flow frequency. At 0.1 Hz, the peak in dissipation is close to the velocity peak with a phase difference of 19° that increases to 115° as the frequency is increased to 1 Hz. Interestingly, at 2 Hz the dissipation peak approaches the peak velocity again and is located at 11° behind it. The behaviour of the peak-to-peak amplitude and phase of dissipation will be discussed more globally next.

The phase averaged deviations from the mean dissipation rate are given by

\[ \langle \delta \epsilon \rangle = \langle \epsilon \rangle - \bar{\epsilon}, \]  

(5.10)

where \( \langle \epsilon \rangle \) is the phase averaged dissipation rate and \( \bar{\epsilon} \) is the mean dissipation rate. Next, a non-dimensionalization of the peak-to-peak amplitude of \( \langle \delta \epsilon \rangle \) (denoted by \( A_{\delta \epsilon} \)) is proposed \( \text{viz.} \)

\[ \sigma_{\epsilon} = \frac{A_{\delta \epsilon}}{\bar{\epsilon}}. \]  

(5.11)

Furthermore, the phase variation of the dissipation rate is better understood when considering the phase difference between the change in grid area and the dissipation, rather than the difference with respect to \( \tilde{u}(t) \), since it shifts in phase for different flow frequencies. Therefore, the phase difference between the area and \( \tilde{u}(t) \) (denoted as \( \phi_{\tilde{a}} - \phi_{\tilde{u}} \)) is obtained first, using the previously derived first-order model of the system. Then, using
Figure 5.2: Phase variation of velocity, turbulence intensity and dissipation rate at 4 m/s and for grid frequencies of (a) 0.1 Hz (b) 0.5 Hz (c) 1 Hz and (d) 2 Hz. The maximum error on the phase averaged dissipation rate was 6.4%.
the results of Figure 5.2, the phase difference between $\tilde{u}(t)$ and dissipation (denoted as $\phi_{\tilde{u}} - \phi_\epsilon$) can be estimated. Using these two quantities, the phase difference between the area and dissipation ($\phi_{\tilde{a}} - \phi_\epsilon$) is calculated.

Figure 5.3 shows the variation of $\sigma_\epsilon$ and $\phi_{\tilde{a}} - \phi_\epsilon$ with respect to flow frequency. The dimensionless dissipation amplitude data from different freestream velocities collapse reasonably well, and the resulting curve is considered representative of the TKE dissipation response under action of an oscillating freestream. It is interesting to note that at 0.1 Hz, $\sigma_\epsilon$ takes on different values, which is indicative of the varying quasi-steady states of the flow at the tested freestream speeds, where the dynamic effects of the forcing are not fully expressed.

On the other hand, the phase plot in Figure 5.3 shows a consistent trend where the area leads the dissipation rate at 4 m/s up to a frequency of 1 Hz, and at 7 m/s and 10 m/s up to a frequency of 2 Hz. However, beyond those frequencies at their respective freestream speeds, a noticeable change in behaviour occurs whereby the dissipation response shifts closer to the imposed area changes. For instance, at 4 m/s the phase difference changes from $-295^\circ$ to $-70^\circ$ as the frequency is increased from 1 Hz to 2 Hz. Similar shifts occur at 7 m/s and 10 m/s by increasing the frequency from 2 Hz to 4 Hz. This change of behavior occurs as a result of the proximity of the forcing, and any associated harmonics, to the inertial scaling range of the flow. This forcing then elicits a response similar to that shown by Cekli et al. (2010) and in Figure 4.25, whereby an excitation of the turbulence field takes place. The PSD spectrum of $u'(t)$, shown in Figure 5.4, shows that as the freestream speed increases from 4 m/s to 10 m/s the onset of the scaling range also shifts to higher frequency values. Therefore, this explains why the observed change in behaviour occurs at 4 Hz for the higher speeds compared to 2 Hz at 4 m/s.

Therefore, the discussed behaviour is indicative of the obvious coupling between the periodic and turbulent parts of the flow as suggested in the preceding section. Furthermore, it also suggests that the TKE may be viewed as a dynamic system, whose input is supplied by the oscillating flow component (through the $\langle u'_i \frac{\partial \tilde{u}_i}{\partial t} \rangle$ term of (5.9)), while its output is the dissipation rate. Figure 5.3 also shows that the dimensionless dissipation amplitude is insensitive to the amplitude of the imposed velocity fluctuations at frequencies above 0.25 Hz, since for a given frequency, $\sigma_\epsilon$ is constant for the different freestream speeds.

The modulation of the phase-averaged dissipation rate also raises the question of whether the oscillating component might have any effect on its scaling behaviour. This
Figure 5.3: Amplitude and phase difference variation of the turbulent kinetic energy dissipation rate with respect to frequency and mean velocity.

Figure 5.4: Power spectral density spectra of $u'(t)$ for the cases where a change in dissipation response takes place.

can be investigated by calculating the phase variation of the scaling constant \textit{viz.}

$$\langle C_\epsilon \rangle = \frac{\langle \epsilon \rangle \langle L \rangle}{\langle u' \rangle^3}.$$  \hspace{1cm} (5.12)

In the preceding expression, the phase averaged integral length scale was calculated using (4.29), with the phase averaged velocity over one period being used as the convective velocity instead of the mean velocity (Kahalerras \textit{et al.}, 1998). This method was used by
Valente & Vassilicos (2011), who also employed the algorithm presented in Kahalerras et al. (1998) to convert temporal signals into spatial ones. However, such a correction did not significantly affect the results, as reported by Valente & Vassilicos (2012) who used the classical Taylor’s frozen flow hypothesis and an identical experimental setup. Furthermore, the integral time scale of $u'(t)$ was assumed to be constant since the majority of the sampled periods exhibited fairly small variations in their time scale as shown in Figure 5.5. Additionally, the obtained time scale for $u'(t)$ was about 0.01 s, which is much less than the time scale of the imposed oscillations, and so $u'(t)$ and $\tilde{u}(t)$ are effectively decoupled. It is important to note here that a more rigorous example for the calculation of the integral length scale was presented by Gomes-Fernandes et al. (2012) who used Particle Image Velocimetry (PIV) as a reference to calculate the needed spatial correlations, however given the current experimental setup and the demonstrated validity of the proposed calculation method for the integral length scale, the aforementioned technique was implemented.

![Figure 5.5: Variation of integral time scale for each flow period at 4 m/s for different flow frequencies.](image)

The resulting phase variation of $C_\epsilon$ for different flow frequencies is shown in Figure 5.6. It is apparent that $C_\epsilon$ shows appreciable fluctuations over time, in agreement with the quasi-periodic fluctuations observed by Goto & Vassilicos (2016a). Therefore, $C_\epsilon$ is not a constant value, which might suggest a departure from the well-known scaling by Taylor (1935). As discussed earlier, such departures have been readily reported in literature through experiments and numerical simulations (Vassilicos, 2015), whereby $C_\epsilon$
Figure 5.6: Phase variation of $C_\epsilon$ at 4 m/s for different flow frequencies.

was shown to follow the scaling given by

$$C_\epsilon = \frac{\text{Re}_M^{p/2}}{\text{Re}_\lambda^q},$$

(5.13)

in the initial decay regions of grid-generated turbulence. In (5.13), $p$ and $q$ are approximately equal to unity and $\text{Re}_\lambda$ is the local Reynolds number $\text{viz.}$

$$\text{Re}_\lambda = \sqrt{\frac{u'^2}{\nu}}.$$  

(5.14)

The result of plotting $C_\epsilon/\sqrt{\text{Re}_M}$ against $\text{Re}_\lambda$ is shown in Figure 5.7. It can be seen that, overall, the data from different cases follows the scaling given in (5.13) for a portion of the flow cycle, which is in agreement with the results presented by Goto & Vassilicos (2016a). The data shown in Figure 5.7 resulted in least square fit values of $q$ ranging from 1.05 to 1.2. Therefore, this is an indication that the oscillating component does indeed influence the scaling behaviour of $C_\epsilon$. However, Figure 5.7 also shows that $C_\epsilon$ exhibits a dependency on the frequency of the oscillating flow. Lower frequencies show a plateau in the values of $C_\epsilon$ for increasing $\text{Re}_\lambda$. This is especially prominent at a mean speed of 7 m/s and a frequency of 0.1 Hz. This plateau and the phases at which it occurs are shown more clearly in Figure 5.8.

The plateau region of constant $C_\epsilon$ is clearly visible at $C_\epsilon/\sqrt{\text{Re}_M} = 10^{-3}$. Furthermore, it also corresponds to regions of fairly constant flow acceleration at $90^\circ \leq \phi \leq 225^\circ$ and $315^\circ \leq \phi \leq 45^\circ$. Outside of those regions, $C_\epsilon$ appears to follow the scaling dis-
Chapter 5. Investigation of Unsteady Turbulence

Figure 5.7: Phase variation of $C_\epsilon/\sqrt{Re_M}$ with $Re_\lambda$ at 4 m/s and 7 m/s for different flow frequencies. Error bars were omitted for clarity, but are similar to those shown in Figure 5.6.

cussed earlier. Such observations suggest that the scaling behaviour varies throughout the duration of one period, between the classical Taylor dissipation law, and the new scaling identified in (5.13). This shift in dissipation scaling was also shown by Goto & Vassilicos (2016b) using DNS and spatially periodic forcing, whereby the forcing was turned off when the dissipation reached its maximum rate. Goto & Vassilicos (2016b) then identified a critical time after the forcing had been switched off, whereby $C_\epsilon$ shifted from a scaling obeying (5.13) to having an approximately constant value. Therefore, it is possible here that the lower flow frequencies allow a greater duration for the flow to settle and consequently reach the critical time identified by Goto & Vassilicos (2016b), thereby causing the observed shift in scaling, before the flow cycle continuing and causing a shift to the $Re_\lambda$ dependent scaling again.
Figure 5.8: (a) Variation of $C_\epsilon/\sqrt{\text{Re}_M}$ with $\text{Re}_\lambda$ at 7 m/s and 0.1 Hz with phases identified in accordance with (b), the phase variation of the velocity. Red symbols correspond to regions of constant $C_\epsilon$, while magenta symbols indicate scaling according to (5.13).
Chapter 6

Conclusion

The work presented in this thesis served two main purposes. The first was to investigate the capabilities of an active grid in producing a wide range of unsteady flows, which will facilitate research in unsteady aerodynamics and enable the extension of current work in this field. The second purpose was to provide insight into the dynamics of turbulence under the action of an oscillating freestream, which is valuable for understanding how turbulence behaves away from its equilibrium state. The practical importance of the first objective lies in the need to simulate unsteady flow conditions that are present in nature and in many engineering applications such as wind turbines and UAVs. On the other hand, understanding the fundamentals of turbulence under imposed perturbations allows for the design of more efficient devices for combustion, mixing, cooling and reduction of aerodynamic noise.

Experiments were done in the recirculating wind tunnel facility at the University of Toronto Institute for Aerospace Studies. The active grid was placed in the middle of the test section to allow measurements to be taken both upstream and downstream of the grid. Velocities were measured using constant temperature anemometry during various modes of operation of the grid. Tests were done at a multitude of freestream speeds, grid frequencies and operation modes.

Results showed a clear modulation of the freestream velocity both upstream and downstream of the grid. Furthermore, downstream measurements exhibited identical fluctuations to those seen upstream except with superimposed turbulent fluctuations. As a result, the method of triple decomposition was used in order to break the flow down into its 3 components: the mean velocity, periodic component and turbulent fluctuations. Higher freestream speeds produced greater amplitudes, while increasing the flow frequency caused the amplitudes to drop sharply before asymptoting to a constant value. Using Mode 1, the amplitude of the flow may be controlled through setting a desired wing
flapping angle. Greater flapping angles caused larger amplitudes. The time series of the generated flow can be customized using Mode 2. At lower frequencies, approximations to a step change in velocity were observed, and as the frequency was increased, the velocity resembled a triangular time variation. It was also shown that the peak velocity position may be shifted in phase by using an appropriate time during which the wings remain open. Additionally, Mode 3 allowed for the creation of a purely sinusoidal freestream velocity. Finally, it was shown that the active grid was capable of recreating a significant portion of test conditions typically used for experiments in unsteady aerodynamics.

The generated velocity amplitudes and their dependency on various grid and tunnel parameters was demonstrated through the extension of the dynamic model presented by Greenblatt (2016) to the case of a recirculating wind tunnel. It was shown that, to a first order approximation, the tunnel behaves as a low-pass filter whose cut-off frequency depends on the average grid area, mean velocity, free tunnel area and an equivalent length that takes the various tunnel areas into account. Therefore, through a judicious choice on these parameters, the level of velocity attenuation may be controlled. The model is in good agreement with the experimental results and so can be used as a guide to produce the desired unsteady flow characteristics. Given a desired velocity amplitude and a reduced frequency, the time constant that produces these conditions can be determined, which in turn allows for the calculation of the needed average grid area.

During the production of unsteady flows, the active grid was shown to resemble a passive grid. First, the produced turbulence intensity remained constant across different grid Reynolds numbers and Rossby numbers. However, the produced turbulence intensity may be customized by operating the grid in Mode 1 and changing the flapping angle. Higher flapping angles resulted in greater turbulence intensity values. Furthermore, the produced integral length scale showed a slight increase with increasing grid Reynolds number. This behaviour represents a compromise between passive grids, where the integral length scale stays constant, and active grids where it shows a significant increase with increasing Reynolds numbers. This was attributed to the fact that during unsteady flow production, the mesh length of the grid shows greater variability than that of passive grids, but less than the variability produced when operating active grids in fully random modes. Additionally, the produced Taylor microscale was found to increase at higher Reynolds numbers, which is in contrast with results reported for active grids. The turbulence intensity of the produced flow may be further customized by operating the grid in a “hybrid” mode during which some of the wings run randomly to introduce turbulence into the flow, while the others generate the needed flow periodicity. The tested modes showed potential in altering the turbulence intensity compared to the base case,
but caused the amplitude to decrease due to the lower blockage during such a mode.

The effect of the freestream oscillations on the generated turbulence was first addressed by investigating the TKE budget for the given experimental conditions. It was shown that under the present assumptions, there exists a term that depends on the frequency of the imposed oscillations which alters the kinetic energy budget. Therefore, this suggests that the turbulence field does show some dependency on the periodic flow component.

This dependency was illustrated through the obtained experimental results, where a step change in the blockage of the grid caused an overshoot in the phase averaged turbulence intensity values before settling again. The size of the overshoot depended on the acceleration of the flow, where greater accelerations caused more overshoot. Similar modulation of the turbulence intensity was observed during Mode 0 of grid operation. The dissipation rate of TKE also exhibited similar phase-averaged fluctuations, and dependencies on the frequency of oscillations. At higher frequencies, peaks in the dissipation and turbulence intensity were attenuated. The dissipation amplitude decreased as the frequency increased while also showing a phase shift with respect to the periodic velocity component. Normalizing the phase averaged dissipation fluctuations with the mean value of dissipation caused the collapse of runs done at different freestream velocities. However, at low frequencies the dimensionless dissipation amplitude varied for the different freestream speeds due to the quasi-steady nature of the flow at those frequencies. The phase of the dissipation rate was then studied with respect to the phase of the changes in grid area. It was shown that dissipation consistently lags the grid area with a notable change in behaviour as the frequency is increased. This was attributed to the proximity of the forcing frequency to the inertial scaling range which excites the turbulence and alters its response.

The phase averaged scaling behaviour of the flow was also investigated by calculating $C_\epsilon$ during a flow period. Appreciable fluctuations were seen in $C_\epsilon$ values, which suggests that it might not be a constant as postulated by Taylor (1935). In fact, the literature shows plenty of evidence that $C_\epsilon$ exhibits a dependency on the inverse of the local Reynolds number, in regions where the turbulence is still in the initial period of decay. This dependency was observed in this work for various test cases, especially at the higher frequencies. However at the lower frequencies, the flow period showed two different regions: one with constant $C_\epsilon$ and another that follows the local Reynolds number dependent scaling. The constant $C_\epsilon$ region corresponded to phases where the acceleration of the flow is nearly constant.
Based on the stated conclusions, the following recommendations for future work are made:

1. The serial communication protocol used to control the grid in this study was sufficient to execute the synchronous modes (Mode 0 to 3), however it showed some limitations when trying to exercise greater control over individual motors in hybrid mode. Such limitations are due to the need to address each motor individually when sending different commands on one serial port, which causes time delays between motors. This poses no significant issues for the randomly actuated rods, but will cause inhomogeneities in the flow amplitude as shown in section 3.4. Therefore, it is recommended that an alternate communication protocol be implemented, or for the rod movement mechanism to be redesigned in order to use a smaller number of motors.

2. While the current work centered on creating temporal velocity oscillations in the freestream velocity, it is also valuable to assess the active grid’s capability in generating spatially varying gusts in the streamwise or spanwise directions, and compare its performance to gust generators typically used in literature (Tang et al., 1996) or with multiple fan arrays (Yang et al., 2017). This may be realized by experimenting with different wing shapes, such as mounting airfoil sections in place of the wings, however that would entail significant modifications to the grid, which might limit its usability in producing the unsteady flows documented in this work.

3. At the time of submittal of this thesis, it is believed that the frequency of the periodic velocity component plays an important role in determining the response of the turbulence field through its dissipation and scaling behaviour, as shown by the analysis in chapter 5. This analysis must be further extended in order to understand the physics governing the observed shift in dissipation scaling during a flow period at the lower frequencies. This can be done by investigating the scale-by-scale budgeting during the flow cycle, and by understanding how the found forcing term influences this behaviour.

4. The calculation of $C_\epsilon$ in section 4.2 involved calculating the integral length scale by employing Taylor’s frozen flow hypothesis using the phase averaged velocity and the integral timescale. While this is sufficient as a first approximation, it is recommended that the presented results be verified using a more rigorous method of calculation, such as using a hot-wire array to obtain the needed spatial correlations, or through the use of PIV. While the interrogation window using PIV might be
limited, Gomes-Fernandes et al. (2012) overcomes this limitation by fitting a function to the observed correlation, within their field of view, and then integrating that function. Additionally, it is recommended that streamwise velocity measurements be made at various locations downstream of the grid in order to quantify the amount of turbulent kinetic energy production due to the grid movement itself.
Bibliography


