Flow over a partially liquid filled cavity

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FLOW OVER A PARTIALLY LIQUID FILLED CAVITY

by

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An electronic version of this thesis is available at http://repository.tudelft.nl/.
With this thesis I will finish my education at the TU Delft. After all those years as a student, it does feel strange now that point is finally near. I know for sure that I will no longer be a student, but besides that it is all blank. Nonetheless, I am very excited to see what will cross my path.

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ABSTRACT

In the oil and gas industry flexible risers connect the subsea pipelines to the offshore installations. These flexible risers are multi-layered structures with a corrugated carcass as innermost layer. For high dry gas flow rates through such risers, tonal noise and mechanical vibrations appear. Research found that at each corrugation in the flexible riser a mixing layer is formed between the free-stream gas flow and the quiescent gas in the corrugation. In the mixing layer vortical structures develop, that periodically interact with the corrugation trailing edge. This creates pressure fluctuations that are heard as tonal noise. On occasion the impingement frequency in the corrugations may become ‘locked-in’, generating synchronized impingement in all the corrugations of the riser. This induces even higher periodic pressure fluctuations. Due to the high working pressure in the flexible risers, the pressure fluctuations can excite the riser (end connections) that ultimately ends in failure.

As this is unwanted, research with the purpose of attenuating noise from flexible risers is performed. One promising factor to reduce noise is increasing the liquid content of the gas stream. From laboratory experiments it was observed that the liquid in the flow filled the corrugations, but how the liquid exactly altered the flow and noise behaviour could not be retrieved. The present work continues with the research on wet gas flows in flexible risers, though more fundamentally. Here the flow of air in and around a single cavity that is partially filled with water, is investigated instead of a corrugated pipe section. This is done by means of planar Particle Image Velocimetry (PIV) measurements for three different Reynolds numbers of the air flow ($Re_{L+} = 2.7 \times 10^4 - 5.3 \times 10^4$) at five different water levels (0%-90% of the cavity depth), for both laminar and turbulent inflow.

The investigation showed that, overall, the water film distribution in the cavity, for any filling degree, was not affected by the lowest Reynolds number air flow. For higher Reynolds numbers the air flow displaced more water from the cavity trailing edge upstream, giving a slightly bevelled water surface for all filling degrees. The flow of air in the partially liquid filled cavity always contained the expected components of a cavity flow: a recirculation cell in the cavity area, a mixing layer spanning the cavity and jet-like flow on the edges.

For the lowest Reynolds number the main recirculation cell decreased in size with adding water and disappeared at 90% filling. For the higher Reynolds numbers a skewed main recirculation cell appeared at all filling degrees, which almost always filled the complete cavity area. The internal cavity flow, in effect, influenced the mixing layer. For higher filling degrees, the mixing layer growth rate and spreading, in general, decreased. It is thought that the lower velocity in the recirculation cell and the upward shifted position hinder mixing layer expansion.

For the laminar, low Reynolds number flow the mixing layer contained well-defined vortical structures, that were expected to be periodic. The mixing layer over a partially liquid filled cavity at higher Reynolds numbers contained ill-defined vortical structures, originating from broken up well-defined vortical structures. These vortical structures were expected to be more random in nature. For the turbulent flows ill-defined vortical structures appeared immediately after separation from the leading edge, without break up. The transition in nature of the vortical structures happened when increasing the Reynolds number of the air flow and when adding water to the cavity from a certain minimum Reynolds number. It is thought that the transition resulted from destabilizing influences, originating from the inflow or the recirculation flow. The above holds till a 90% water level in the cavity, thereafter no vortical structures appeared in the mixing layer.

Under certain circumstances, liquid is expected to accumulate in small amounts in most of the corrugated sections of the flexible riser. Destabilizing influences and the following appearance of ill-defined vortical structures on a large scale are hence foreseen. The noise level produced by the individual corrugations of the flexible riser will be similar, but will contain a lower tonal noise component. More important is that the random nature of the vortical structures is expected to prevent locked-in flexible risers and thereby excessive levels of noise and mechanical vibrations.
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INTRODUCTION

1.1. BACKGROUND
In the oil and gas industry risers form the connection between the subsea pipelines and the offshore installations, a depiction of this is seen in figure 1.1. Starting from the seabed these risers can extend for several hundreds of meters and have typical diameters ranging up to 0.35 m. Application wise the risers are used for various purposes such as oil and gas production and fluid injection [10]. As a consequence of this, they transport a multiphasic mixture containing gas, oil, water, sand and sometimes even injected chemicals. During operation the risers are subjected to harsh service conditions being [34, 60]:

- High temperatures: fluid temperatures that can reach up to 170 °C.
- High pressures: both internal fluid pressure (with a maximum of 400 bar) and external hydrostatic pressure (risers reach depths of 2-3 kilometers below the water surface).
- Dynamic environment: the surface currents can induce large deformations and therefore large mechanical stresses.
- Aggressive environment: the internal flow can be contaminated with toxic substances and in addition both the internal and external flow may lead to corrosion.

Over the years several riser designs were implemented. Traditionally, risers were manufactured from steel and were named ‘rigid’ risers, due to the characteristic structural properties [4]. In the past few years, however, flexible risers emerged. These are multi-layered structures with the aim of high structural axial stiffness and relatively low structural bending stiffness, such that they can resist deformation and the corresponding stresses [30]. To achieve this, a mix of steel and reinforced thermoplastics are utilized for the layers [43]. An artist impression of the distribution of the layers in relation to each other, together with a short description of the function of each layer is given in figure 1.2. Depending on the complexity of the flow and the exposed conditions, the amount of layers applied to a flexible riser can be varied.

Owing to the better suited structural properties and the layer adaptability of the flexible risers, conform to the aforementioned hostile environment, they are more favourable for operation in offshore fields than rigid risers. This made operation of more offshore fields along with reduced costs possible.

1.2. MOTIVATION
Unfortunately, the flexible risers do not only have advantages over the rigid risers. Given the nature of the flow, the service conditions and the large length of the risers, they can in general be seen
Introduction

Figure 1.1: Flexible risers linking the subsea pipelines and the offshore facility [61]

Figure 1.2: Internal structure of a flexible riser, adapted from [12]

As the most critical components of the offshore facilities. Risk assessment revealed that the failure frequency per year of flexible risers is in the order of $O(10^{-3})$. This is high in comparison to rigid risers and other piping components, since they have failure frequencies that are one or two orders of magnitude smaller [35]. Absolutely taken the possibility of riser failure is still small, but since the consequences of leakage can be catastrophic, rigorous measures are taken.

The most common failure mode of flexible risers is damage to the outer sheath by the formation of cracks and holes in this layer. Other failure modes that frequently arise are collapsing of layers, bursting of layers, tensile failure, compressive failure, torsional failure, fatigue, erosion and corrosion [44]. The failure modes have various reasons; sometimes they seem to occur randomly, but they also appear by human errors (e.g. exceeding operating envelope boundaries). While the latter failures easily can be explained, a posteriori failure evaluations of the former category have shown that these failures arise by mechanisms on which knowledge and understanding at the moment of designing has been inadequate [31]. Technological developments of flexible risers will augment the possibilities for operation, but on the other hand will also be paired with an increase of new failure modes of the first type.

An example of a new failure mode of the first type is vibration induced fatigue, which became significant when throughput of dry gas flow through flexible risers increased [39]. At the moment at least seven facilities have experienced this type of failure and have reported excessive levels of noise and mechanical vibrations, ultimately followed by failure of the risers end connections due to fatigue. The noise and mechanical vibrations were not always present, they only appeared above a specific flow rate and showed jumps in frequency depending on the flow rate. Limiting the production was therefore proposed as a temporary solution for other offshore installations. Though since the demand for natural gas will only increase, this was not deemed as a satisfactory long-term solution.

Following this, further investigations on the reason of failure has been executed. Thus far large progress significant to flexible risers is booked in the form of a large joint industry project called ‘Flexible Risers’ in which oil and gas companies participated [36]. From this project good comprehension of the governing mechanism was achieved by combining field experiments, laboratory experiments and numerical simulations [7]. Besides this, diverse studies from other industries that use corrugated pipes as well and have had similar problems, also contributed much to the understanding of the governing mechanism. A very recent one is the work of Nakiboglu [45].

From the conducted research in the literature it has been shown that the noise and mechanical vibrations are caused by a flow induced pulsation mechanism emerging from the internal geometry (carcass) of the flexible riser [57]. The carcass is a pipe with an Agraff profile, i.e. an alternating sequence of corrugations and plateau sections (see figure 1.3). Usually this profile is distributed axissymmetrically over the pipe, but on occasion a helical component can also be present. When flow passes through the corrugated pipe, mixing layers are formed over each of the corrugations in which
vortical structures may develop, as is shown in figure 1.4. The vortical structures can (partially) impinge on the trailing edge of the corrugations, which results into pressure fluctuations. When this happens periodically a tonal noise is generated. As each corrugation in the flexible riser will have a periodic interaction with vortical structures at a slightly different frequency, noise containing several frequencies is generated in the flexible riser. The general name for this phenomenon is Flow Induced Pulsations (FIP), but in relation to flexible risers in the offshore industry it also goes under the name of ‘singing risers’. On occasion the vortex impingement frequency can couple with one of the acoustic modes of the flexible riser. In this case a ‘lock-in’ mechanism appears that synchronizes the vortex impingement in all the corrugations of the pipe, hence leading to tonal noise of a distinctive frequency and even higher pressure fluctuations. Since the gas flow is at a pressurized state in the order of 100 bar, the pressure fluctuations induced by impinging vortical structures, either amplified by the lock-in mechanism or not, are considerable. Consequently the dynamic pressure pulsations can excite the riser and the riser end connections, causing mechanical vibrations. The vibrations act as an additional noise source, but, more important, they also result in high dynamic stresses that ultimately lead to fatigue failure of the riser end connections. The whole fatigue process for a locked-in flexible riser can be quite fast, failure reports have been made just nine hours after the first vibrations were observed [59].

In the search of attenuation of the noise and the associated mechanical vibrations a number of conditions were tested. One of the parameters that was experimented with was the liquid content of the gas flow. In laboratory situations it was perceived by Belfroid et al. [6] that noise was clearly existing for dry gas flows, but was weakened or even absent for wet gas flows. What the underlying reason for this is, is not yet known. In the same study they opted for three mechanisms that could potentially explain this attenuation of noise. As noise is the net effect between production and damping, attenuation is twofold and can either be attained by reducing the noise source or increasing the noise damping. The first two mechanisms that were proposed are related to reducing the noise production, while the third mechanism is based on increasing the noise damping and was carefully assessed further in reference [16]. These identified mechanisms are:

- Filled-up corrugations with liquid may produce less noise, since the noise emitted by the corrugations diminishes with the depth.
- The thicker upstream boundary layer of the gas flow, resulting from liquid deposition on the wall, could possibly reduce the source strength of the corrugations.
- The presence of a liquid film or droplets on the wall may lead to additional damping.

Nonetheless, owing to the nature of the measurements performed by Belfroid et al., high speed camera recordings as displayed in figure 1.5 and microphone measurements, at the moment no decisive answer could be given about the dominant mechanism responsible for the noise attenuation.
1.3. SCOPE OF THE RESEARCH
Considering the gap in research on multiphase flow in flexible risers, the present thesis aims at bringing further understanding to what extent the presence of the liquid phase in flexible risers influences the gas flow and the corresponding noise. Specifically the first mechanism, the filling-up of the corrugations with liquid, will be investigated into detail. This is done by examining the gas flow field in and around the corrugations, whereby special attention is given to the mixing layers spanning the partially liquid filled corrugations and the underlying recirculation zones. As the mixing layer flow, that is the noise source, is foreseen to change upon liquid addition through different influences from the recirculation zone. Conclusions on the noise production potential of the flow can then be drawn, based on the resulting mixing layer flow. Such work, with partially liquid filled corrugations, has not been performed before, according to the knowledge of the author. For the present study, though, a single cavity instead of a corrugated pipe (section) is chosen, since the pulsations that cause the noise emanate from the vortices in the cavity mixing layer and not from the corrugated pipe. The difference is that a corrugated pipe has more corrugations and a lock-in mechanism that may set in, which both lead to higher pressure fluctuations and thereby higher noise levels. The main drawback, however, is the difficulty of investigating the internal flow in a corrugated pipe with liquid addition, since droplet deposition on the wall may hinder the view. A single cavity does not have this limitation and is therefore a well suited first step to get a better insight into the behaviour and the noise production potential of the flow over a partially liquid filled corrugation.

The research has been performed at the Low Speed Laboratory, which is part of the Aerospace engineering faculty of Delft University of Technology. For the present study experimental investigations have been performed in an open jet wind tunnel by means of Particle Image Velocimetry (PIV). The research objective of the present study can be formulated as:

**Investigate the effect of filling up the cavity volume with liquid in terms of flow behaviour and the noise production potential**

This can be achieved by accomplishing the following tasks:

- Carry out a literature study on cavity flows.
- Perform experiments using PIV on a cavity structure under various filling degrees and Reynolds numbers.
- Compare the obtained cavity flow structures for partially liquid filled cavities to the cavity flow structures for empty cavities, as found in the literature.
- Translate the results from the partially liquid filled cavities to expectations for multiphase flow in flexible risers.

1.4. OUTLINE OF THE THESIS
The current Master thesis presents the results from research done on a partially liquid filled cavity immersed in a gas stream. The thesis is structured in the way given below.

The second chapter of the thesis will cover the fundamental physics behind fluid mechanics that is relevant for the present research. After this, the third chapter continues with a description of the flow in and around cavities. From here important prior knowledge is obtained, that is convenient in relation to the upcoming results. Chapter 4 will treat the experimental facility, including the wind tunnel, the cavity model and the components of the measurement setup. Moreover, some light will be shed on the procedures performed before and after the measurements. The fifth chapter presents the selected testing conditions and subsequently discusses the results. Finally the sixth chapter ends the report with the conclusions and recommendations for further study.
THEORETICAL BACKGROUND

Fluid mechanics is the discipline of physics concerned with fluids (liquids and gasses) at either stagnant conditions (fluid statics) or flowing conditions (fluid dynamics). Within the realm of fluid mechanics there are several subdivisions. First and foremost, there is the difference between single-phase flows and multiphase flows. Single-phase flows can be seen as flows of pure fluids or homogeneous mixtures, whereas multiphase flows are flows with different phases or with immiscible fluids. In the current study the focus is on single-phase gas flows, but some interaction between two fluids (gas-liquid) may also be addressed.

Since fluid mechanics is thus at the base of the events described in the subsequent chapters, the fundamental physics behind fluid mechanics are introduced in the present chapter. First the important conservation laws are covered, followed by the discussion of non-dimensional numbers and the treatment of shear flows. The main purpose of this chapter is to give readers enough background, so that the material and concepts discussed in the later chapters are understandable.

2.1. CONSERVATION LAWS

For all fluid flows the behaviour is governed by a general set of conservation laws\(^1\) (given below) plus an additional equation of state [42]:

\[
\frac{D\rho}{Dt} + \rho \vec{\nabla} \cdot \vec{u} = S_{\text{mass}} \tag{2.1}
\]

\[
\rho \frac{D\vec{u}}{Dt} = \vec{\nabla} \cdot \vec{\sigma} + \sum f_{\text{body}} \tag{2.2a}
\]

\[
\rho \frac{Du_i}{Dt} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + \lambda \delta_{ij} \frac{\partial u_j}{\partial x_j} \right] + \sum f_{\text{body},i} \tag{2.2b}
\]

\[
\rho \frac{De}{Dt} + p \vec{\nabla} \cdot \vec{u} = \mu \Phi - \vec{\nabla} \cdot \vec{q} + S_{\text{heat}} \tag{2.3}
\]

The first equation is the conservation of mass and governs the fact that under normal circumstances (discarding relativistic effects) the amount of mass in a system should be preserved (incoming mass equals outgoing mass; \(S_{\text{mass}} = 0\)). The second equation is the conservation of momentum, which in essence is the second law of Newton applied to an infinitesimal fluid element, but then per unit volume. The left-hand side states the local and convective acceleration of the fluid element, whereas the right-hand side contains the divergence of the stress tensor and the body forces, per unit volume, working on the fluid element. Using a constitutive relation that assumes the fluid is Newtonian, the stress tensor (\(\vec{\sigma}\)) may be rewritten such that the formula becomes equal to equation 2.2b.

\(^{1}\)Here the material derivative has been used, which consist of a local and convective derivative \(\frac{D\star}{Dt} = \frac{\partial \star}{\partial t} + \vec{u} \cdot \frac{\partial \star}{\partial \vec{x}}\).
This is the Navier-Stokes equation, which is the most important equation in the realm of fluid mechanics. The contribution of the divergence of the stress tensor is here divided into a part caused by hydrostatic pressure gradients and by shear stress gradients. The final equation is the conservation of energy and governs the internal energy of the flow. Since energy should be conserved, this equation states that the variation of internal energy plus the energy produced by work should be equal to the energy lost by viscous dissipation and heat transfer and added by a heat source.

In the situation relevant for the present study, where a gas flow is used that travels at low speeds relative to its speed of sound (low Mach number), viscous dissipation as well as temperature and density changes can be neglected. Because the density of the fluid is then constant throughout the domain, the flow is considered to be incompressible and the set of governing equations reduces to:

\[ \nabla \cdot \vec{u} = 0 \quad (2.4) \]
\[ \rho \frac{D\vec{u}}{Dt} = -\nabla p + \mu \nabla^2 \vec{u} + \sum f_{\text{body}} \quad (2.5) \]

From the implication that the density is a constant property, the conservation of mass is now simplified to the condition that the flow is divergence-free. Applying this to the Navier-Stokes equation, results in the dropping out of the second coefficient of viscosity. Finally for the incompressible situation the conservation of energy and an equation of state become redundant.

2.2. NON-DIMENSIONAL NUMBERS
Non-dimensional numbers are very important in fluid mechanics. They are ratios of physical quantities and can help to identify the dominant effects in the fluid flow. The non-dimensional numbers can be derived from the non-dimensionalization of equations and/or the Buckingham Pi theorem. Below a few non-dimensional numbers are highlighted that will be used throughout the report.

- Reynolds number
  The Reynolds number is the main non-dimensional number in fluid mechanics. It can be seen as the ratio between inertial and viscous forces in the flow and is represented by the following statement:

\[ \text{Re}_L = \frac{\rho UL}{\mu} \quad (2.6) \]

In this report the Reynolds number is defined for the gas stream only and the velocity scale is taken as the free-stream gas velocity \( U_\infty \). For the length scale various measures are taken in this report and the used length scale is always added in subscript next to the Reynolds number. Most often the length scale is taken as the distance \( L \) between the cavity leading edge and the cavity trailing edge. For small values of the Reynolds number the viscous forces are dominant in the flow, while for larger Reynolds numbers the inertial forces are leading. What follows from this, is that the Reynolds number also governs the transition of a flow from laminar to turbulent. Laminar flow is the situation where the flow moves as if it was ordered in smooth layers without disruption. Turbulent flow is more chaotic in nature, with fluid parcels locally demonstrating irregular behaviour through eddies. However, globally the flow can still be represented quite well by time-averaged properties. Under typical conditions (e.g. no pressure gradient) the laminar to turbulent transition for flow over a flat plate, that is aligned with the flow, appears at a Reynolds number \( \text{Re}_x \) (based on the downstream distance \( x \) from the flat plate leading edge) in the order of \( \mathcal{O}(10^5) \).

- Strouhal number
  The Strouhal number is a measure of the oscillatory flow mechanism in a stream. For low Strouhal numbers the steady state flow movement governs the frequency, whereas for high
2.3. SHEAR FLOWS

Strouhal numbers the viscosity dominates the flow oscillations [56]. Mathematically the Strouhal number is represented as:

\[ St_L = \frac{fL}{U} \tag{2.7} \]

In the current study the characteristic frequency is chosen as the frequency at which vortices impinge on the downstream cavity wall, the length and velocity scale are chosen as the free-stream gas velocity and the cavity span.

- Mach number
  The Mach number is the ratio between a local or global velocity scale of a medium and the speed of sound characteristic to that medium and is given as:

\[ M = \frac{U}{c_\infty} \tag{2.8} \]

Since the Mach number represents the flow speed relative to the speed of sound, a link can be made to the incompressible nature of flows. For low Mach numbers (\( M < 0.3 \)), as is the case for both fluid streams in the present study, the flow is assumed to be incompressible, whilst for higher Mach numbers the flow is compressible (\( M > 0.3 \)).

2.3. SHEAR FLOWS

In an empty non-bounded domain a fluid flow can freely travel at a free-stream velocity \( U_\infty \) with a uniform velocity profile, this is visualized in figure 2.1. However, once this stream comes into contact with a solid or fluid (boundary) at a different speed, a shear layer is formed. The shear layer originates from the velocity difference (shear) between both layers, which initially creates a velocity discontinuity. Due to viscosity this discontinuity cannot be maintained at the interface, the viscosity, namely, imposes a no-slip condition that should be satisfied. This will result in equal velocities and stresses across the interface. In order to achieve this, the fluid flow near the interface is influenced. It will result in a steep transition of the flow velocity over a layer of finite height that can be seen as the height of the shear layer. The layer starts somewhere in the free-stream area and extends to the interface (fluid-solid interaction) or the other free-stream area (fluid-fluid interaction).

Owing to the no-slip condition steep velocity gradients are thus present in the shear layer, which can be related to shear stresses by the viscosity. A consequence of the shear in the shear layer is the appearance of vorticity. Vorticity is a measure of the rotation of the fluid parcels in the flow and is defined as:

\[ \vec{\omega} = \vec{\nabla} \times \vec{u} \]

for two-dimensional flows:

\[ \omega_z = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \]

Since the flow velocity differs along the shear layer height, to some extent a rotation is realized on the fluid parcels that hence generates vorticity.

BOUNDARY LAYER

A boundary layer is the region where an interaction occurs between a fluid flow and a solid wall. At the wall a no-slip condition has to be satisfied, resulting in a zero interface velocity for the non-moving wall considered here. In the boundary layer the velocity reduces from the free-stream velocity attained at some height, to a zero velocity at the wall, as is shown in figure 2.1. The profile shape and thickness \( \delta \) over which this boundary layer develops depends on the nature of the flow and the Reynolds number. For flat plates aligned with the flow and with no pressure gradient, the boundary layer follows the Blasius profile in the laminar regime from the point where \( \text{Re}_x >> 1 \). Here the
non-dimensional height wise velocity development in the boundary layer is numerically stated as a function of the non-dimensional boundary layer height \( \eta = y \sqrt{U_{\infty} \nu x} \); the exact values of \( \eta \) and the corresponding velocity ratios can be found in textbooks like White [64]. The trend of this profile is approximately quadratic and the shape is independent of the downstream coordinate. In the turbulent regime the mean velocity profile (since turbulent flow is characterized by fluctuations) can be modelled by the Power law profile through the relation given in equation 2.10. Here a shape factor \( n \) determines the steepness of the velocity profile.

\[
\frac{U_{\infty}}{U(y)} = \left( \frac{y}{\delta} \right)^{\frac{1}{n}}
\]  

(2.10)

In the incompressible flow regime the magnitude of the shape factor is positively correlated to the momentum thickness based Reynolds number (definition of the momentum thickness given further on). Hence equations and graphs relating the two over a large range can for example be found in Johnson et al [23]. For a fully developed pipe flow this shape factor is equal to \( n = 7.0 \), though variations of the shape factor have been seen from \( n = 5.0 \) to \( n = 10.0 \). Since the momentum thickness increases in the downstream direction, the turbulent profile shape will thus change in the downstream direction as well. As the turbulent velocity profile becomes steeper, the shape factor will correspondingly increase in magnitude.

Both the laminar and turbulent boundary layer profiles are also plotted in figure 2.2. From this it is observed that the turbulent boundary layer is steeper near the wall than the laminar boundary layer. This means that a turbulent boundary layer contains more energy than a laminar boundary layer and can therefore withstand larger resistance before separation occurs.

To quantify the thickness of the boundary layer two measures will be used [25]:

- **95% boundary layer thickness**
  
  The 95% boundary layer thickness is a measure of the height at which the boundary layer will have a velocity that is equal to 95% of the free-stream velocity. This can be formulated as:

  \[
  \delta_{95} = y \text{ at } u = 0.95 U_{\infty}
  \]  

(2.11)

- **Momentum thickness**
  
  The momentum thickness is defined such that \( \rho U_{\infty} \theta \) is the momentum loss due to the presence of the boundary layer. The relation that represents \( \theta \) is given by:

  \[
  \theta = \int_{0}^{\infty} \frac{u}{U_{\infty}} \left( 1 - \frac{u}{U_{\infty}} \right) \, dy
  \]  

(2.12)

Finally these boundary layers will evolve in space (dotted line in figure 2.1). How large this development is, depends on the Reynolds numbers of the flow \( \text{Re}_x \). For the laminar and the turbulent
velocity profiles treated above, the following relations for boundary layer thickness growth have been derived [64]:

\[
\frac{\delta_{95}}{x} = \frac{4.0}{\sqrt{\text{Re}_x}}, \quad \theta = \frac{\delta_{95}}{10} \tag{2.13}
\]

\[
\frac{\delta_{95}}{x} = \left(\frac{0.012(n + 1)(n + 2)}{n}\right)\sqrt{\frac{\text{Re}_x}{n}}, \quad \theta = \frac{n\delta_{95}}{(n + 1)(n + 2)} \quad \text{valid for } \text{Re}_x > 1000 \tag{2.14}
\]

**Mixing layer**

A mixing layer is another type of shear layer, but in contrast to a boundary layer a free-stream flow is now not in contact with a solid wall, but with a fluid that has a different velocity and/or fluid properties. Due to viscosity again a no-slip condition is enforced at the interface between both layers, but since two fluids are involved in a mixing layer the interface velocity is non-zero.

The mixing layers are in the literature represented by an inlet flow with a higher velocity $U_1$ at the top and a lower velocity $U_2$ at the bottom [5], this is also shown in figure 2.3. From these velocities a velocity ratio $R$ is defined that compares the relative magnitude of the shear $\Delta U$ to the average flow velocity $\overline{U}$. In mathematical form this is:

\[
R = \frac{\Delta U}{\overline{U}} \quad \text{with } \Delta U = U_1 - U_2 \quad \text{and } \overline{U} = \frac{U_1 + U_2}{2} \tag{2.15}
\]

Initially the discontinuous velocity jump across the interface will prevail, but viscosity will smooth this out and form a smooth overlap between both layers over a characteristic mixing layer thickness. Just as for the boundary layers the thickness of the mixing layer is expected to increase with the spatial downstream distance. In the present study the vorticity thickness is used as measure of this length scale and is defined as:

\[
\delta_\omega(x) = \frac{U_1 - U_2}{(\frac{\partial u}{\partial y})_{\text{max}}} \tag{2.16}
\]

The vorticity thickness is preferred over the momentum thickness of the mixing layer, as for cavity flows large variations in the momentum thickness are found through the underlying recirculation zone [54]. Though it should be remarked that the length scale found from this definition is only a measure of the mixing layer spreading, which shows the same trends as the ‘real’ mixing layer spreading and is by no means equal to the real mixing layer spreading.

The velocity distribution that the laminar inflow to the mixing layer will take on, is best described by a tangent hyperbolic function:

\[
u(y) = \overline{U}\left[1 + R \tanh\left(\frac{y}{2\delta_\omega}\right)\right] \tag{2.17}
\]

The velocity profile (as depicted in figure 2.3) has an inflexion point and is therefore (inviscidly) unstable, which means that velocity disturbances will be unstably amplified by a Kelvin-Helmholtz instability mechanism. This mechanism results from the shear force at the interface and will cause amplification of the disturbances at that location, as they are propagated downstream. Initially for laminar flows the amplification will cause an exponential growth rate, which will turn linear (turbulent) later on. In the region where the amplification is exponential the disturbances are still small and manifest itself through waves. If the disturbances in the mixing layer get large enough, non-linear effects will dominate and lead to the rolling-up of disturbances into a periodic array of span wise vortices. This is depicted in figure 2.4.
Figure 2.4: Rolling-up of the mixing layer by the Kelvin-Helmholtz instability mechanism, adapted from [22]
Cavity flows are defined as flows in and around hollow spaces and appear in various forms. One type is the response of a cavity, without a top edge or with a partial top edge, to an exterior flow field. Since the cavity flow is imposed by a mixing layer, that is formed by the interaction between an exterior free-stream flow and an internal cavity flow, this configuration is named a mixing layer-driven cavity. Another well-known form of cavity flow is the response of a cavity, with top edge, imposed to a top edge motion. In this case a mixing layer is absent and instead the flow configuration is enforced by the top edge drift, this arrangement is called a lid-driven cavity. The former configuration is of much importance in the area of flow induced noise, while the latter configuration is widely used as benchmark configuration in the area of computational fluid dynamics.

The current thesis is limited to the mixing layer-driven cavity without a top edge, but with a partial liquid filling. Although this is the focus, no studies were found in the literature on partially liquid filled cavities. Therefore, mainly the behaviour of single-phase cavity flows is treated in this chapter, such that an indication of the expected gas flow behaviour in a partially liquid filled cavity, as will be encountered in chapter 5, is obtained. This is complemented by some additional statements on partially liquid filled cavities, based on the observations of Belfroid et al. [6] for multiphase flow in corrugated pipes.

The chapter will start with an introduction of different cavity geometries. Following this, the rest of the chapter will be devoted to the flow characteristics of the mixing layer-driven cavities.

3.1. CAVITY GEOMETRY

In the literature rectangular cavity structures are modelled as a backward facing step followed at a certain distance by a forward facing step. The geometry of the cavity is characterized by three dimensions: cavity length \( L \), cavity depth \( D \) and cavity width \( W \) and depending on these dimensions distinction can be made between several cavities [14]. A description of these cavity types can be found below, whereas illustrations of all these cavities are given in figures 3.1 and 3.2.

- Deep cavities
  Cavities for which the length and depth obey the following relation: \( L/D < 1 \).

- Shallow cavities
  Cavities for which the length and depth obey the following relation: \( L/D > 1 \). Further subdivision of these cavities is possible into [41]:
    - Open cavities
      For these cavities a mixing layer is formed at the top corner of the cavity leading edge that impinges with the top corner of the cavity trailing edge. This occurs for length to depth ratios between 1 < \( L/D < 9 \).
– Transitional cavities
For these cavities a mixing layer is formed at the top corner of the cavity leading edge that impinges with the bottom corner of the cavity trailing edge. This occurs for length to depth ratios between $9 < \frac{L}{D} < 12$.

– Closed cavities
For these cavities a mixing layer is formed at the top corner of the cavity leading edge that impinges with the bottom edge of the cavity. Behind this stagnation point another mixing layer is then formed at the bottom edge that impinges with the cavity trailing edge. This occurs for length to depth ratios higher than $\frac{L}{D} > 12$.

- Wide cavities
Cavities for which the length and width obey the following relation: $\frac{L}{W} < 1$. For wide cavities the three-dimensional effects can be considered small and are often neglected.

- Narrow cavities
Cavities for which the length and width obey the following relation: $\frac{L}{W} > 1$. For narrow cavities the flow is three-dimensional.

In our experiments an empty cavity structure will be used that is shallow open, which is conform to the corrugations present in flexible risers. Furthermore this cavity is also wide, such that roughly a two-dimensional mixing layer behaviour is obtained.

3.2. CAVITY FLOW
Initially, when the free-stream flow has not yet reached the cavity, the flow in the cavity is quiescent. The upstream flow, either in laminar or turbulent state, will propagate with velocity $U_\infty$ towards the cavity. Once it reaches the cavity opening a shear $\Delta U$ will present itself over the cavity opening, equal to the free-stream velocity $U_\infty$. The shear over the cavity may lead to the appearance of three different ‘modes’ in the cavity depending on the cavity mode parameter $\frac{L}{\theta}$, which is the ratio between the cavity span and the inflow momentum thickness of the boundary layer. The modes and the intervals at which they present themselves for a laminar incompressible flow are [13]:

- $\frac{L}{\theta} < 81$: Non-oscillating mode
- $81 < \frac{L}{\theta} < 120$: Shear layer mode
- $120 < \frac{L}{\theta} < 155$: Wake mode

The observed behaviour corresponding to each mode is also graphically represented in figure 3.3. One should, however, be cautious with simply applying these boundaries to every case, since they were found for a fully two-dimensional, laminar case. In planar cavities, as are used in the present study, three-dimensional effects are present in the flow. This can cause instances where the cavity mode parameter is substantially higher than $\frac{L}{\theta} = 120$, while the flow is still in the shear layer mode. The presence of these three-dimensionals thus prevents the appearance of the wake mode in planar cavity flows and instead the shear layer mode shows up. For a turbulent incoming flow these three modes should also appear, though no expertise is available in the literature on the regions in which these modes appear with respect to the cavity mode parameter.

From the three aforementioned modes the non-oscillating mode and the shear layer mode are most relevant to our study, since they are characteristic to the flow in flexible risers without and with periodic pressure pulsations on the corrugation trailing edge. Therefore these modes will receive most attention in this chapter. The wake mode will, for the sake of completeness, be shortly explained as well.
3.2. CAVITY FLOW

3.2.1. NON-OSCILLATING MODE AND SHEAR LAYER MODE

FLOW STRUCTURE

The flow structure of the non-oscillating mode and the shear layer mode are very similar and feature a mixing layer spanning the cavity length, together with a recirculation area in the cavity region. We will start by treating the mixing layer dynamics of both modes, since this is the main difference between both modes.

In the mixing layer spanning the cavity, disturbances are subjected to a Kelvin-Helmholtz instability mechanism. This means that the disturbances are amplified, while they are transported downstream by the flow. By increasing the cavity span or the Reynolds number of the incoming flow a stronger Kelvin-Helmholtz instability mechanism is induced, that consequently gives a more energetic mixing layer with more violent oscillations [3].
Just as for a free mixing layer formed by a laminar inflow, as treated in section 2.3, the amplification of a cavity mixing layer has an exponential growth rate over the initial cavity section, where the disturbances are still small. Once the disturbances become larger, non-linear effects come into play and the realized growth rate will be almost linear [3]. For turbulent inflow the presented growth rate is immediately linear. For cavity mixing layers the linear growth rate for laminar incoming flows normally varies between $\frac{d\delta}{dx} = 0.10 - 0.20$, whereas for the turbulent incoming flows this upper limit is set to $\frac{d\delta}{dx} = 0.37$. This (slightly) deviates from the growth rate of a free mixing layer, as this mixing layer is bounded by a cavity structure from below. The positive growth rate and large spreading prevail for most of the cavity length, only close to the cavity trailing edge the mixing layer growth rate and spreading decay.

Due to the shear between the free-stream flow and the internal cavity flow, clockwise oriented vorticity is produced in the mixing layer. In the initial section, with small disturbances, this vorticity will manifest itself as a large and continuous streak. Further downstream the streak separates in vortical structures owing to the non-linear contributions. The process of the mixing layer breaking up into vortical structures over the cavity area is graphically represented in figure 3.4. For the shear layer mode the vortical structures organize themselves into well-defined periodical structures, whereas for the non-oscillating mode they do not [24]. For the non-oscillating mode the structures are smaller and look more like a region of broadband turbulence. In this report we will name these vortical structures ill-defined, following Oshkai et al [47]. The vortical structures are propagated downstream with a velocity between $0.4U_\infty - 0.7U_\infty$ [48]. As the distance over which the vortical structures are convected increases, the structures grow in size but reduce in strength [8].

At a certain moment the mixing layer and the intrinsic vortical structures reach the trailing edge and will (partially) impinge on it, as seen in figure 3.6. The mixing layer flow and a part of the vortical structures are then diverted into the cavity area. This will happen for both modes and as a result a wall jet flow appears at the cavity trailing edge [27]. This wall jet flow will drive a recirculation flow in the region bounded by the cavity edges and the mixing layer. This recirculation zone consists of at least one large clockwise rotating recirculation cell, and sometimes a secondary recirculation cell located at the cavity leading edge and a tertiary recirculation cell located at the cavity trailing edge, as depicted in figure 3.5. The size and position of the main recirculation cell depends on the Reynolds number and the nature of the incoming flow. As the Reynolds number is higher and the flow is turbulent, stronger mixing layer oscillations appear that transfer more energy to the internal cavity area [18]. This lead to a stronger, larger and a more upstream placed main recirculation cell.

In the internal cavity area the recirculating flow will be retarded at the cavity edges and hence boundary layers form at these places. Owing to the flow direction the edges contain mainly counter clockwise vorticity, as is depicted in figure 3.7.

In terms of mean pressure behaviour, as shown in figure 3.8, it is perceived that at the height of the mixing layer the flow is subjected to an almost constant pressure over the cavity span. Variations in pressure, however, occur as the mixing layer approaches the downstream edge. At the centre of the main recirculation cell a local low pressure area appears and very close to the downstream edge the pressure gradient becomes adverse, ending in a stagnation point at the downstream wall [20]. Additionally, the vortical structures, which have an intrinsic low pressure, introduce an unsteady pressure component through impingement on the cavity trailing edge (see again figure 3.6). From here it is found that when the vortical structures are located just upstream of the trailing edge, an elevated pressure is felt. As the vortical structure comes closer (such as during impingement), its low pressure core makes the pressure at the wall drop again [27]. For the shear layer mode these collisions occur periodically. Hence periodic pressure fluctuations are generated that can be heard as tonal noise. For the non-oscillating mode the collisions are random in nature and therefore create random pressure fluctuations. This can be heard as white noise.

What is described above will happen for single-phase cavity flows. For the mist flow experiments
3.2. CAVITY FLOW

Figure 3.4: Cavity mixing layer and the development of vortical structures, clockwise oriented vorticity is negative [8]

Figure 3.5: Flow structure of a shallow cavity [11]

Figure 3.6: Pressure fluctuations due to vortical structures impacting with the downstream wall, time progresses in a clockwise sequence [28]

Figure 3.7: Mean vorticity distribution in a shallow cavity, clockwise oriented vorticity is negative [8]

Figure 3.8: Mean pressure in a shallow cavity [20]

performed by Belfroid et al. [6] the droplets will deposit in the cavity and form a liquid film that correspondingly should change the internal cavity flow. For shallow cavities at low gas velocities they observed that the film remained uniformly distributed at the bottom of the cavity. Furthermore they found that upon increasing the gas velocity, the liquid film was eventually completely displaced
from the trailing edge. This is supposed to be the result of the high dynamic pressure of the gas flow that was diverted into the cavity. As this stream impinges on the liquid surface, liquid is displaced upstream. Thereby creating a negatively inclined gas-liquid interface with a maximum angle of 75°. In essence the cavity will then have a slanted ‘bottom edge’ and from a study of Kuo et al. [26], where cavities with solid slanted bottom edges were investigated, a slanted bottom edge lead to a more skewed main recirculation cell. This recirculation cell was less powerful than a circular main recirculation cell that would appear for a flat bottom edge. How the mixing layer is affected by the presence of a liquid phase for various gas velocities is not known from the study of Belfroid et al. Though Kuo et al. found for cavities with slanted bottom edges that also the mixing layer displayed less strong oscillations.

**SELF-SUSTAINED OSCILLATIONS**

While the non-oscillating mode has no intrinsic oscillations, the shear layer mode can be characterized by two self-sustained oscillations. For the present study the treatment of self-sustained oscillations is limited to the periodic vortex-wall interaction. The vortex-wall interaction is the result of velocity disturbances in the mixing layer. Under the influence of the Kelvin-Helmholtz instability mechanism they develop into oscillations that roll-up into vortical structures and periodically impinge on the cavity trailing edge. The disturbances in the mixing layer will emanate from feedback that can either be hydrodynamic or acoustic in nature [14]. The hydrodynamic feedback always occurs and governs the feedback for incompressible flows. In contrast, the acoustic feedback solely appears for compressible flows and is also dominant over the hydrodynamic feedback in this situation. Since the flows considered in the present study are all incompressible, acoustic feedback is not relevant. The acoustic feedback will therefore not be treated here and the interested reader is referred to Gloerfelt [14].

The hydrodynamic feedback can present itself in two manners. The first way in which hydrodynamic feedback occurs, is through the mixing layer. The mixing layer can be seen as a row of line vortices, each with a circulation $\Gamma$. By Biot-Savart induction each vortex produces an induced velocity, which can be seen as a disturbance, that affects the surrounding fluid and thus also the mixing layer. The induced velocity that one such vortex in the mixing layer generates, is equal to:

$$u_{vortex} = \frac{\Gamma}{2\pi r}$$

(3.1)

where $r$ is the distance from the vortex centre. Secondly, the internal cavity flow can also function as feedback. The upward flow of the main recirculation cell towards the mixing layer is embedded with vorticity originating from the vortical structures that are entrained at the downstream edge and the boundary layers on the cavity edges. Just as the induced velocity, the vorticity in the stream can again be seen as a velocity disturbance and this disturbance may be absorbed by the mixing layer. Which of the two hydrodynamic mechanisms is dominant depends on the cavity dimensions. It has been reported that for relatively short cavity spans the Biot-Savart induction is dominant, whereas for larger cavity spans the recirculation effect is leading [27]. Unfortunately, no statements were made regarding the dimensions of short and large cavity spans. However, from the $1/r$ dependence of the Biot-Savart induction and the span of the cavity structure in the present study, it is thought that the recirculation effect will be dominant in our case. A posteriori evaluations of the measurements confirmed that this was true.

A frequency for the vortex-wall interaction can be derived by considering the time needed for the vortical structures in the mixing layer, that move at speed $U_c$, to traverse the cavity of length $L$. The time needed for the upstream propagation of disturbances by hydrodynamic feedback can be neglected, since this happens almost instantaneous. Rewriting the frequency in terms of the Strouhal number then gives [28]:

$$St_L = (N - \alpha) \frac{U_c}{U_\infty}$$

(3.2)
Here $N$ corresponds to the amount of large vortical structures formed in the mixing layer, which normally varies between $N = 1$ and $N = 3$. The term $\alpha$ is an empirical parameter that is usually approximated as $\alpha = 1/4$, since this gives the best agreement with experiments. The ratio between the convective velocity and the free-stream velocity is customarily taken as a constant value that ranges between $0.4 < \frac{U_c}{U_\infty} < 0.7$. The resulting Strouhal number for the vortex-wall interaction process, as observed in the literature, normally varies between 0.5 and 3.2 [28].

### 3.2.2. Wake Mode

In the wake mode, opposed to the previous two modes, the mixing layer will not span the complete cavity. Instead it will repeatedly collide with the bottom edge, which induces an unstable behaviour. The flow instability provokes large interaction between the free-stream flow and the internal cavity flow and results into the characteristic feature of the wake mode: ejection of large vortical structures from the cavity area. These ejected vortical structures have a size that is comparable to the depth of the cavity.

It is noteworthy to mention that the only experimental study reporting this phenomenon (Gharib et al. [13]) was performed using an axisymmetric cavity in a water tunnel. Mainly numerical studies were performed on this mode and they only experienced the wake mode in incompressible two-dimensional simulations. From this it is concluded that the wake mode is caused by a hydrodynamic effect that is two-dimensional in nature. By introducing three-dimensional disturbances, it was even seen that the flow transitioned back from the wake mode to the shear layer mode.
In this chapter further information is given on the performed experiments. First the experimental facility will be covered, followed by the cavity model. The remaining part of the chapter is dedicated to the experimental techniques: Hotwire Anemometry and Particle Image Velocimetry. Here we will describe the measurement setups and shed some light on the procedures performed before and after the measurements.

4.1. Wind Tunnel

In the present investigation the M-tunnel at the Low Speed Laboratory of Aerospace Engineering is used. The M-tunnel is a wind tunnel that can be used in an open jet or closed loop configuration, here the first variation is chosen. For the open jet scenario the air enters the wind tunnel via a filter and is subsequently lead to the diffuser. After the diffuser the air passes the settling chamber and the nozzle, from where it finally exits the wind tunnel through an outlet section of constant cross-sectional area. The cross-sectional area of this outlet section is 40 cm by 40 cm. In the open jet mode the maximum velocity that the air can reach at the wind tunnel exit is around 35 m/s. The turbulence level that the flow attains at the exit is low (< 1%), due to the gauzes in the settling chamber and the large contraction ratio of the cross-sectional area in the nozzle.

The free-stream velocity at the wind tunnel exit is measured indirectly by means of a Pitot-static tube, which was located close to the end of the outlet section. The static pressure and the total pressure readings of the Pitot-static tube are fed to a Mensor DPG 2400 pressure difference meter that could measure a pressure difference with an uncertainty of 0.1 Pa. The pressure uncertainty induces a maximum error in the velocity measurement of 0.08% (based on the velocity magnitude) and can therefore be safely neglected. The pressure difference meter was already calibrated and before use validated against some static water column heights.
### 4.2. Cavity Model

For the cavity model a flat plate with a cavity machined into it is used, as seen in figure 4.2. The flat plate is made of Plexiglas and has a length of $L_{\text{flat plate}} = 500$ mm, a width of $W_{\text{flat plate}} = 400$ mm and a depth of $D_{\text{flat plate}} = 25$ mm. The leading edge of the flat plate is made elliptical such that a neat boundary layer will form on the model, that does not separate at the juncture between the leading edge and the flat plate section. This elliptical edge has an aspect ratio of 4, which from previous studies \cite{3, 27} is deemed sufficient to prevent flow separation. Along both sides the flat plate is also equipped with two 200 mm high Plexiglas plates, to reduce the three-dimensional effects in the flow over the cavity model.

The cavity itself is located 150 mm from the leading edge of the plate. This is done to obtain a small incoming laminar boundary layer at the cavity leading edge, that will lead to the appearance of the shear layer mode over the cavity span. The cavity geometry has a curved leading edge with a radius of curvature of $r = 10$ mm and an orthogonal trailing edge. This configuration will give a better comparison with the flow in flexible risers and stimulates the noise production of the cavity \cite{45}. Owing to the curved leading edge the total length of the cavity mouth becomes $L + r = 40$ mm; the cavity width and the cavity depth are, respectively, $W = 360$ mm and $D = 20$ mm. The cavity thus does not cover the full width of the flat plate, which ensures that the liquid cannot escape from the cavity volume through the sides.

As an additional measure to prevent flow separation on the top side, a deflection plate is fitted towards the end of the flat plate. This plate, placed at the top side, will create extra resistance at this side. As a result the flow will stay attached at the top side, but will separate from the bottom side. The deflection plate has a length of $L_{\text{def}} = 80$ mm and a width of $W_{\text{def}} = 400$ mm and is placed 50 mm above the flat plate at an angle of attack of $\alpha_{\text{def}} = 2^\circ$ with respect to the horizontal axis. This specific angle was chosen, since from experiments it was found to prevent boundary layer separation on the top side of the flat plate, without affecting the boundary layer shape (too much) by its presence.

During the measurements the cavity model was placed 50 mm behind the wind tunnel exit, at a height halfway the outlet section.

### 4.3. Hotwire Anemometry

Before doing PIV measurements, the boundary layer that arrives at the cavity leading edge needs to be quantified. For this an intrusive method named Hotwire Anemometry (HWA) was used, which enables doing point velocity measurements in space over time. HWA works through a hotwire probe whose working element is a metallic wire that is inserted into the flow. When the HWA system is switched on, the temperature of the metallic wire increases due to Joule heating. If the wire is thereafter subjected to flow, the wire temperature will again drop because of forced convection. A convenient property of metals is, however, that an induced temperature change will lead to a proportional change in electrical resistance. By making use of this fact and linking the hotwire probe in an electrical configuration called a Wheatstone bridge, the change in wire resistance may cause a response in the voltage and the current in the electrical circuit. The response of the voltage or the current can subsequently be related to the velocity magnitude at the wire position. For additional information on the HWA measurement technique the reader is referred to the practical guide provided by Dantec Dynamics \cite{33}.

In the present section the HWA instrumentation is covered first, these components together form the HWA setup as shown in figure 4.3. After this, attention is paid to the selected settings, the calibration procedure and the measurement procedure.

**Instrumentation**

The HWA instrumentation comprises a hotwire probe, a traversing system, a Wheatstone bridge with amplifier and a data acquisition system. Here each of these components is shortly elucidated.
• Hotwire probe
The hotwire probe is the part of the system that is immersed in the flow. The hotwire probe consist of a lead on which two prongs and a metallic wire are welded. The hotwire probe used in this investigation is from Dantec Dynamics.

• Traversing system
The hotwire probe is mounted to a mechanical traversing system, which makes it possible to accurately displace the hotwire probe in space during operation. This system is powered by two electro motors, such that the hotwire probe can move in two directions normal to each other with a precision of one tenth of a millimeter. The traversing system is controlled by a graphical user interface in LabVIEW.

• Wheatstone bridge with amplifier
The Wheatstone bridge is an electrical configuration containing four arms, with each arm having a certain resistance. For one of these arms the resistance is made up of the hotwire probe and the BNC cable connecting the hotwire probe to the Wheatstone bridge. In case of no flow the resistance of this arm is constant, but when the hotwire probe is exposed to a flow the resistance of this particular arm changes. For the used operation mode of the HWA system, the amplifier correspondingly increases the current in the bridge. As a result the wire resistance and the voltage in the circuit will increase. In the present investigation the 56C17 CTA bridge from Dantec Dynamics is employed.

• Data acquisition system
The final component is the data acquisition system. This system uses a National Instruments BNC-2111 block connector that is connected to the computer. The computer, which contains a data acquisition board, receives the analog voltage signal from the Wheatstone bridge and converts it into a digital signal. The digital signal is then stored on the hard disk as a text file.

Settings
For the HWA system the following settings were applied during the present study:

• Operation mode
The HWA system can be run in various operation modes. For the current situation it was chosen to run the HWA system in the Constant Temperature (CTA) mode, instead of the Constant Voltage (CVA) and the Constant Current (CCA) mode. In the CTA mode the temperature of the hotwire is tried to keep constant during operation by the aforementioned actions of the Wheatstone bridge and the amplifier. As a result there is no effect of thermal inertia, which leads to a better performance of the sensor [55].

• Overheat ratio
Secondly the overheat ratio had to be set on the Wheatstone bridge. The overheat ratio \( a \) can be seen as a non-dimensional temperature rise of the hotwire and is defined in the following manner:

\[
a = \frac{R_w - R_0}{R_0}
\]

Here \( R_w \) is the hot hotwire resistance (HWA system switched on) and \( R_0 \) the cold hotwire resistance (HWA system switched off). A higher value for the overheat ratio leads to a HWA system that is more sensitive to fluctuations. Though it will increase the noise of the measured signal as well. Therefore an overheat ratio of \( a = 0.5 \) is chosen in the present study, which is an optimum trade-off between both effects.
• Sampling frequency
Finally the sampling frequency of the hotwire probe has to be set. The sampling frequency of
the HWA system cannot be chosen too high else the samples will be correlated, but if the fre-
quency is chosen too low the measurements will take longer than necessary. For the boundary
layer measurements the samples have to be uncorrelated and therefore a moderate frequency
of 1000 Hz was used, which is justified by the small time scale of the boundary layer flow.

CALIBRATION PROCEDURE
After implementing the previous settings, the HWA system can be calibrated. During the calibra-
tion procedure the voltage over the Wheatstone bridge is related to the velocity at the hotwire tip.
Therefore the wind tunnel velocity was varied between 1 m/s and 21 m/s, with steps of 1 m/s.

Ideally the hotwire probe should measure the free-stream velocity during the calibration proce-
dure without the model being present in the wind tunnel. Though due to the difficulty of placing
the model in the wind tunnel and the fact that the velocity is only affected by the presence of the
model very close to the model surface, the calibration was performed with the hotwire probe 100
mm above the flat plate surface. Since this is well outside the boundary layer of the flat plate, the
velocity here is expected to match the free-stream velocity.

The obtained voltage signal over the Wheatstone bridge is then related to the velocity at the
hotwire tip by a fourth order polynomial function [55] of the following form:

\[ U = c_0 + c_1 E + c_2 E^2 + c_3 E^3 + c_4 E^4 \]  

(4.2)

The fourth order polynomial function is inspired from the inverse of King's law, which approxi-
mately assumes a fourth order relation between the free-stream velocity and the measured volt-
age. A plot of the calibration points and the proposed calibration function can be seen in figure
4.4, which shows that the fourth order polynomial accurately relates the calibration points in the
voltage-velocity space. The error bars are not plotted in the figure, since they are very small and
therefore hardly visible. It can, however, be stated that during the calibration procedure the aver-
age uncertainty in the velocity, with a 95% confidence interval, was equal to 0.1% of the respective
velocity magnitude.

Figure 4.3: HWA setup; 1. traversing mechanism and
2. hotwire probe

Figure 4.4: HWA calibration function

MEASUREMENT PROCEDURE
After completion of the calibration the actual measurements can start. Firstly the position of the
hotwire probe is set to a height of 1 cm with respect to the model surface. This position is chosen
as the reference to determine how far the traverse could be moved downwards. Then starting from
a position of 5 mm, the traverse was lowered in steps of 0.2 mm. Close to the point where the
theoretical boundary layer should start, the step size was halved to 0.1 mm. The measurements were then continued until a distance of 0.5 mm from the solid wall to ascertain that the hotwire probe would not touch the wall. At every point a number of samples of the flow was taken. The number of samples that were taken, determine the accuracy with which the mean velocity can be derived. The higher the amount of samples, the smaller the uncertainty of the mean is, but more samples also lead to longer measurements. As a trade-off between accuracy and time, the number of samples that are taken is chosen equal to $N = 1000$. Combined with the sampling frequency this results into a measurement time of 1 second per step.

All the text files that were obtained from the measurements were transported to MATLAB where the final data evaluation took place. From a posteriori analysis of the data an uncertainty is estimated based on the fluctuations in the free-stream area. For a 95% confidence interval the uncertainty varies between $\Delta u = 0.004 \text{ m/s}$ for 10 m/s to $\Delta u = 0.010 \text{ m/s}$ for 20 m/s. In comparison to the measured velocity magnitudes, this error is negligible.

4.4. PARTICLE IMAGE VELOCIMETRY

After the HWA measurements were performed the investigation continued with the main purpose of the research: the Particle Image Velocimetry (PIV) measurements in and around the cavity. In contrast to HWA, PIV is a non-intrusive method that allows one to obtain the complete velocity field. For PIV measurements the flow is seeded with tracer particles. These tracer particles have a Stokes number smaller than one, so that the movement of the particles is imposed by the fluid flow. During the measurement the flow located at the area of interest (henceforth called Field of View or FOV) is multiple times highlighted by two short pulses of a laser sheet. As a consequence, the tracer particles in the flow are illuminated. The illuminance of the tracer particles at these two instances is captured on the image frames of a digital camera. In between the image frames the particles have displaced over a certain distance, which can be obtained by cross-correlating the image frames with each other. The cross-correlation operation is, however, not done on the image frames as a whole, but on smaller rectangular pixel groups called interrogation windows. From this the local displacement in the interrogation window is found, that with knowledge of the magnification factor and the time between the consecutive photographs leads to the velocity vector in that region. By dividing both image frames completely into interrogation windows and repeating the cross-correlation operation for every interrogation window, the velocity field in the FOV is acquired. The above gives a short introduction to the PIV measurement technique and for more information the reader should resort to the book of Adrian and Westerweel [1].

First the PIV instrumentation will be treated, that forms the PIV setup as shown in figures 4.5 and 4.6. After this the selected settings, the calibration procedure, the measurement procedure, the data reduction techniques and the post-processing steps are covered.

INSTRUMENTATION

The PIV instrumentation consist of a fog generator, a laser plus imaging optics, a digital camera and imaging software. In the following a small explanation is provided of these four components.

- Fog machine
  For the tracer particles a fog was used. The fog was created by a ‘SAFEX Twin Fog double power’ fog generator and contains particles with a mean diameter of 1 $\mu$m. The tracer particles were made from a liquid mixture called the SAFEX normal power mix fluid, that produces particles consisting for 30% of water and 70% of glycol.

- Laser
  To obtain a laser sheet a ‘Spectra Physics Quanta Ray PIV 400’ laser is used together with imaging optics. This laser emits a wavelength of $\lambda = 532$ nm with a maximum energy of 400 mJ per
pulse for 6 ns long. As the system contains two independent lasers the minimum separation time between two light pulses is 0 s. The Gaussian-shaped laser beam is by a series of components directed to the measurement location and transformed into a two-dimensional diverging laser sheet. This path is partially shown by the green line in figure 4.5.

- Digital camera
For capturing the illuminance of the tracer particles the LaVision Imager Intense is used, which is a camera equipped with a CCD sensor. The sensor has a resolution of 1280 pixels by 1024 pixels with a pixel pitch of 6.45 $\mu$m. The minimum time interval between recording the first and second image frame with this camera is 500 ns, and by also taking into account the duration of the individual image frames and the time needed for reading-out the sensor, a theoretical acquisition rate of 10 Hz is found. As in the ideal situation this is the limiting component in the image acquisition process, the maximum theoretical acquisition rate of the PIV system is 10 Hz. In practice this, however, reduced to an acquisition rate of 3.4 Hz, since the data transfer from the camera to the computer and the data storage pose to be bottlenecks.

- Imaging software
The imaging software is used for the image acquisition on which PIV relies. The software controls the light source and the camera, such that the laser pulses and the image frames are synchronized. For this investigation the DaVis 7.2 software of LaVision is used. In addition to control the image acquisition, the imaging software can also be used for post-processing of the resulting images.

**SETTINGS**
The advantageous properties of the PIV measurement technique are, unfortunately, also paired with a more complicated nature than HWA. Therefore more settings have to be set before the measurements can start.

- Magnification
The magnification is the ratio between the image length and the real length. The image length is equal to the length of the CCD sensor, whereas the real length depends on the chosen FOV. The FOV embraced the cavity and some distance up- and downstream of the cavity. From initial calculations a magnification of $M = 0.16$ was computed. Prior to the measurements, however, the contributions to the FOV of the sections up- and downstream of the cavity were believed to be too large. Therefore a larger magnification of $M = 0.26$ was used during the
measurements. For convenience in the latter stages of the selected settings, it was therefore also chosen to use a lens with a focal length of \( f = 55 \text{ mm} \) for this magnification.

- **F-stop**
  Another important parameter that has to be determined is the f-stop \( f_\theta \) of the lens. The f-stop is the ratio between the focal length of the lens and the lens aperture \( D_{\text{aperture}} \) and is mathematically defined as:

\[
f_\theta = \frac{f}{D_{\text{aperture}}}
\]  

Choosing a value for the f-stop is not unambiguous, since it depends on several other aspects. In short the motivations that influence the choice for a specific f-stop value are [55]:

- **Focal depth**
  The focal depth \( \delta z \) is the distance over which the illuminated particles can be imaged in focus. For PIV measurements the focal depth has to be larger than the laser sheet thickness. From the definition of the focal depth given in equation 4.4, it is seen that a larger f-stop induces a larger focal depth.

\[
\delta z = 4.88 \lambda f^2_\theta \left( \frac{M + 1}{M} \right)
\]  

- **Diffraction**
  Diffraction is detrimental to PIV measurements since it causes the illuminance of the tracer particles to diffract over a small area on the image frame. For large amount of diffraction the images can therefore become very blurry. For PIV measurements, though, a little bit of diffraction can also be advantageous, because it can ensure that an imaged tracer particle is larger than a single pixel. This will avoid peak-locking, which is the instance where the displacements of the imaged tracer particles cannot be determined with sub-pixel accuracy. The diameter of the imaged tracer particles is governed by the following relation:

\[
d_{\tau} = \sqrt{d_{\text{geo}}^2 + d_{\text{diff}}^2} \text{ where }
\]

\[
d_{\text{geo}} = M_{\text{particle}}
\]

\[
d_{\text{diff}} = 2.44 \lambda (1 + M) f_\theta
\]

This equation shows that a larger f-stop induces more diffraction.

- **Brightness**
  In brighter images the illuminated light of the tracer particles will have a higher intensity. For PIV measurements a high brightness is thus expected to be favourable. It is reasoned that a larger aperture diameter produces a brighter image, since more light can enter the lens. From the definition of the f-stop given in equation 4.3, it can then be implied that a larger f-stop results into a less bright image.

From the above, three criteria can be formulated with respect to the f-stop value: the focal depth has to be larger than the laser sheet thickness, the diffraction should lead to an imaged particle diameter of at least 1 pixel and the brightness should be as high as possible. From these constraints an initial f-stop value of \( f_\theta = 9 \) was found to satisfy all criteria. This value was used as reference, but during the measurements it was experienced that this f-stop value gave a poor contrast between the tracer particles and the background. Therefore the f-stop value was increased to \( f_\theta = 11 \), which gave a better contrast between the tracer particles and the background.
### EXPERIMENTAL FACILITY

- **Interrogation window size**
  The image frames captured during the measurements, are later on cross-correlated to acquire the velocity field in and around the cavity. For the cross-correlation the image is divided up into interrogation windows, which is quite an important parameter. Large interrogation windows have a low spatial resolution, as detailed information characterized by small length scales (e.g. small vortices and gradients with right to space) cannot be retrieved. These are, namely, averaged out over the whole interrogation window. For smaller interrogation windows the spatial resolution is higher, but at the cost of more noise in the data. For the present study a trade-off was made between both effects and the final data representation is based on square interrogation windows of 32 by 32 pixels throughout the whole image. By combining this size with the magnification it is found that each interrogation window corresponds to a physical size of 0.77 mm by 0.77 mm.

- **Time step**
  The minimum time that the first image frame is separated from the second image frame is limited by the camera to 500 ns. From a cross-correlation point of view, this is way too small. Since in this case no significant displacement of the tracer particles in an interrogation window can be found. It is normal practice to take the time step \( \Delta t \) such that the tracer particles have moved over a distance equal to one quarter of the smallest interrogation window size [55]. This is called the one quarter law and in mathematical form this looks like:
  \[
  \frac{MU_\infty \Delta t}{L_{\text{interrogation window}}} \leq \frac{1}{4}
  \]  
  Though since a multipass approach with decreased interrogation window size (which is explained further on) is used in the present study, the time step obtained with this rule can rather be seen as a reference value than a strict measure.

Based on the three free-stream velocities of 10 m/s, 15 m/s and 20 m/s during the PIV measurements, an initial magnification of \( M = 0.16 \) and a final interrogation windows size of 32 by 32 pixels, the following time steps were derived from equation (4.6): \( \Delta t_{10 \, m/s} = 34 \mu s \), \( \Delta t_{15 \, m/s} = 22 \mu s \) and \( \Delta t_{20 \, m/s} = 17 \mu s \). When implementing these time steps in practice, satisfactory results were found and these time steps were therefore also used throughout the measurement campaign.

### CALIBRATION PROCEDURE

After the settings have been selected, the calibration of the camera can begin. When the camera captures the image frames at two consecutive instances in time, the local displacements between the first and second image frame can be found through cross-correlating the interrogation windows in the images. Though the displacements that are computed with this operation are in pixels and not in the physical unit of meters, which has our interest. In addition, since with the used setup the camera will record the images under an angle, the horizontal distance between two pixels will correspond to a different physical length than the vertical distance between two pixels. Therefore a calibration is performed by the DaVis software before the PIV measurements start. In this way the physical location of the pixels in space...
is obtained and a mapping from the image space to the real space can be made. For this purpose a calibration grid is employed, that is placed at FOV. The calibration grid, as displayed in figure 4.7, consists of ‘+’ markers, that have a horizontal and vertical length of 1 mm and a centre-to-centre interspacing of 2 mm. With this calibration grid the calibration procedure performed by the DaVis software was successfully underwent.

**MEASUREMENT PROCEDURE**

Once the calibration was finished the PIV measurements could start. During the measurement campaign the fog generator fogged up the complete room before every experiment. In this way a uniform seeding density could be obtained during the measurements.

During each of the measurement series at least 2000 images were shot, such that good mean flow properties could be obtained. In the end this led to 20 minute long measurement series. Samples of the image frames that were shot during the measurement series for an empty cavity and a cavity with a 50% liquid filling degree at 10 m/s are depicted in figures 4.8 and 4.9.

![Figure 4.8: Raw image of an empty cavity at 10 m/s](image1)

![Figure 4.9: Raw image of a 50% filled cavity at 10 m/s](image2)

**DATA REDUCTION TECHNIQUES**

For the cross-correlation of the interrogation windows a multipass approach with decreased interrogation window size was used, together with a 50% overlap.

In a nutshell, the multipass approach with decreased interrogation window size starts the cross-correlation at a larger interrogation window size than the desired one. For this larger interrogation window size (which was 64 by 64 pixels in our case) the pixel shifts are recovered, which can be converted into the corresponding velocity field. Thereafter another pass (i.e. the computation of the velocity field from the image frames) can be performed, at either the same or a smaller interrogation window size. Here we first chose to do a pass at a smaller interrogation window size of 32 by 32 pixels, followed by a second pass at the same interrogation window size. Though the trick is now that in the multipass approach the velocity field is not again calculated from scratch. The pixel shifts that make up the previous velocity field are namely used as pre-shifts on the interrogation windows for the current computation of the velocity field. In the end this gives more accurate and reliable velocity fields [2].

The overlap concerns the way the image is split into interrogation windows. For 50% overlap the number of interrogation windows is doubled since they overlap for 50% with each neighbour. This is beneficial as tracer particles on the edge of an interrogation window, that do not contribute much to the cross-correlation without overlap, are now shifted towards the centre of the next interrogation window. In this way they will contribute more to the cross-correlation and result into a higher spatial resolution.
Performing the cross-correlation in this manner, leads to the extraction of 80 velocity vectors in the horizontal direction and 64 velocity vectors in the vertical direction.

After the cross-correlation was performed, the pixel shifts for the interrogation windows were investigated for peak-locking. As the flow in our FOV consist of two parts, namely a free-stream flow outside the cavity and an internal cavity flow, the pixel shifts in these two regions are analysed separately. We will here only show the (fractional) displacement histograms of the horizontal pixel shifts in figure 4.10 for the free-stream velocity of $10\ m/s$ at no liquid filling. This is done since all the remaining configurations and all the vertical pixel shifts displayed the same trend. In the free-stream area, the pixel shifts peak around a single value, as shown in the left histogram. Since this peak value is not necessarily equal to an integer value, this is not caused by peak-locking. Instead it is the result of the uniform displacement of the particles in the free-stream area, as shown in the middle histogram. The pixel shifts seem to be centred around the mean value with a Gaussian profile. In the internal cavity area the fractional pixel shifts show an approximately uniform distribution. This is also what is expected for a PIV measurement. Overall, this confirms that we did not (significantly) suffered from peak-locking in the present study.

![Displacement histogram for horizontal pixel shifts, binsize = 0.04; Left: fractional pixel shift in free-stream area, middle: absolute pixel shift free-stream area, and right: fractional pixel shift in cavity area](image)

**POST-processing DaVis**

After the velocity fields were procured, some post-processing steps in DaVis were performed with the aim of removing spurious velocity vectors and making the velocity fields more suitable for data analysis. Regarding the tracking of spurious velocity vectors two methods are used: a Signal to Noise Ratio (SNR) filter and a median filter. The SNR filter compares the ratio between the highest peak found by cross-correlating two image frames and the second highest peak. If the ratio of these two peaks was smaller than 1.1, the resulting velocity vector was labelled as spurious. These vectors were removed, recalculated by linear interpolation of the neighbouring velocity vectors and finally reinserted into the velocity field. Next, a median test was applied, which calculates the median from a set of velocities containing one velocity vector that is denoted as the centre and all its eight neighbours. If the difference in magnitude between the velocity vector in the centre and the median is larger than a certain threshold (chosen here as twice the root mean square of the velocity set), the velocity vector is denoted as spurious. Here again the new velocity vector was recalculated from linear interpolation of the neighbouring velocity vectors. Though the velocity vector was only reinserted once the threshold for the median filter was satisfied.

From a posteriori evaluations it was found that the amount of spurious vectors in an instantaneous velocity field varied between 0.5% and 5%, which from the literature [63] was shown to be adequate. The spurious vectors seemed to be the result of reflections, shadows and sometimes insufficient seeding. To make the velocity data more suitable for post-processing in MATLAB, the filtering was followed by applying a mask over the region where a wall or liquid was present.

All the aforementioned steps lead to a velocity field as shown in figure 4.11. This can be seen as a matrix of size 80 by 64 containing velocity data.
4.4. PARTICLE IMAGE VELOCIMETRY

POST-PROCESSING MATLAB
In MATLAB the visualization of relevant fluid mechanics phenomena, from the acquired velocity fields, took place. In the next chapter these figures will be discussed in further detail, and here the computation carried out to obtain these figures is explained. The main focus of this report will be on the mixing layer and the way it is influenced by the recirculation zone, that is formed in a partially liquid filled cavity area. Therefore the following quantities were chosen:

**Figure 4.11: Instantaneous velocity field obtained by PIV, colours based on the x-component of the velocity (red is the highest magnitude and blue is the lowest magnitude)**

Mean velocity magnitude and streamlines  
Firstly, the mean velocity field and the streamlines are presented, because they will give an indication of the velocities in and around the cavity. The time-averaged velocity field is found by averaging the x-component of the velocity \( u \) and the y-component of the velocity \( v \) over a minimum of \( N = 2000 \) images for each spatial coordinate according to equation 4.7:

\[
\overline{u}_{i,j} = \frac{1}{N} \sum_{n=1}^{N} u_{i,j}^n \quad \text{and} \quad \overline{v}_{i,j} = \frac{1}{N} \sum_{n=1}^{N} v_{i,j}^n \quad (4.7)
\]

Here the subscript \( i \) and \( j \) denote the horizontal position and the vertical position in the velocity field as shown in figure 4.11, whereas the superscript \( n \) stands for the instantaneous image number. The mean magnitude of the flow velocity is then found by Pythagorean addition of both the x-component and y-component of the velocity.

The calculation of the streamlines is inspired from the definition of the stream function \( \Psi \) as:

\[
\bar{u}_{i,j} = -\frac{\partial \Psi_{i,j}}{\partial x} \approx -\frac{\Psi_{i+1,j} - \Psi_{i,j}}{dx} \rightarrow \Psi_{i,j} = \Psi_{i,j} - \bar{u}_{i,j}dx \quad (4.8)
\]

\[
\bar{v}_{i,j} = \frac{\partial \Psi_{i,j}}{\partial y} \approx \frac{\Psi_{i,j+1} - \Psi_{i,j}}{dy} \rightarrow \Psi_{i,j} = \Psi_{i,j} + \bar{v}_{i,j}dy
\]

Here \( dx \) and \( dy \) denote the horizontal and vertical step size, which are constant and equal to each other. To start this computation the magnitude of the stream function \( \Psi \) at one spatial coordinate has to be known. Since there is no flow at the wall, it is chosen to equalize the magnitude of the stream function \( \Psi \) in the left bottom corner, at a solid wall, to an arbitrary value of zero. The value of the stream function at the remaining points is then found by cycling through the velocity field using the formulae in equation 4.8. The drawback of this differential-based method is that the exact value of the stream function cannot be retrieved. This is, however, not a problem since we are interested in the streamlines, rather than the stream function magnitude.

**Mean and instantaneous vorticity distribution**  
Secondly, since the mixing layer is subjected to analysis in this report, the vorticity will be investigated. The vorticity shows the rotation of the fluid parcels in the mixing layer and the internal cavity area and whether vortical structures are formed in the mixing layer. To calculate the vorticity several methods are available. Looking at the definition of vorticity given in equation (2.9) and the shape of the velocity data, the computation of vorticity via a central second order difference (see equation (4.9)) would be the simplest method. Though the simplicity of this method comes at a price, since this way of calculating the vorticity tends to amplify the noise present in the measured velocity. Because experimental data always contains noise, this effect can be substantial. Therefore the approach is switched to a filtered second order difference, as is given in equation (4.10). Here a mild low-pass filter is used on the measured velocity, that
attenuates the noise, before the vorticity is computed [1]. Other methods to compute the vorticity from the measured velocity with even lower noise and/or even higher accuracy are also available in the literature. These beneficial properties are, unfortunately, paired with increased smoothing of the vorticity field. Furthermore, since the methods are dependent on more neighbours that lie further away, it is more difficult to apply these methods in the near-wall region. As a consequence of those two reasons, these methods were not chosen.

\[
\omega_{i,j}^n = \frac{v_{i+1,j}^n - v_{i-1,j}^n}{2dx} - \frac{u_{i,j+1}^n - u_{i,j-1}^n}{2dy}
\]

central second order difference \hspace{1cm} (4.9)

\[
\omega_{i,j}^n = \frac{1}{8dx dy} \begin{bmatrix}
\frac{d}{dx} [u_{i,j+1}^n + 2u_{i,j-1}^n + u_{i+1,j}^n + u_{i-1,j+1}^n] \\
\frac{d}{dy} [v_{i+1,j+1}^n + 2v_{i+1,j-1}^n + v_{i,j+1}^n + v_{i,j-1}^n] \\
\frac{d}{dx} [u_{i+1,j}^n + 2u_{i,j+1}^n + u_{i+1,j+1}^n] \\
\frac{d}{dy} [v_{i,j+1}^n + 2v_{i,j+1}^n + v_{i+1,j+1}^n]
\end{bmatrix}
\]

filtered second order difference \hspace{1cm} (4.10)

With the formula in equation (4.10) the instantaneous vorticity distributions were computed. The mean vorticity distribution \(\overline{\omega}\) was also obtained by equation (4.10). Only here the instantaneous velocity components \(u^n\) and \(v^n\) were replaced by the time-averaged velocity components \(\overline{u}\) and \(\overline{v}\), as computed from equation (4.7).

**Vorticity thickness** Thirdly, the vorticity thickness will be treated to infer the mixing layer spreading in the mean flow. Since the definition of the vorticity thickness (as given in equation (2.16)) contains the gradient \(\frac{\partial u}{\partial y}\), again a filtered second order approach is used in the computation. The gradient is then approximated as:

\[
\frac{\partial u}{\partial y}_{i,j} = \frac{2[2\overline{u}_{i,j+1} + (\overline{u}_{i+1,j+1} + \overline{u}_{i-1,j+1})] - (2\overline{u}_{i,j-1} + (\overline{u}_{i+1,j-1} + \overline{u}_{i-1,j-1}))}{8dy}
\]

From this the vorticity thickness is computed as:

\[
(\sigma_\omega)_i = \frac{U_\infty}{\max_j \left(\frac{\partial u}{\partial y}\right)_{i,j}}
\]

(4.12)

**Turbulence intensity** Finally, the turbulence intensity will, in contrast to the mean flow properties, give some quantitative information on the fluctuating flow. This can be related to the strength of the mixing layer and the intrinsic vortical structures. The turbulence intensity is defined as the ratio between the turbulent fluctuations in the flow and the flow magnitude in the free-stream area:

\[
I_{i,j} = \frac{\sqrt{\left(u'_{i,j}\right)^2 + \left(v'_{i,j}\right)^2}}{U_\infty}
\]

(4.13)

where: \(u'_{i,j} = \sqrt{\frac{1}{N-1} \sum_{n=1}^{N} \left(u_{i,j}^n\right)^2}\) and \(v'_{i,j} = \sqrt{\frac{1}{N-1} \sum_{n=1}^{N} \left(v_{i,j}^n\right)^2}\)

The error quantification for the PIV measurements is derived from this, as the mean of the turbulent fluctuations in the free-stream area, which should be zero, is taken as measure of the standard deviation. A region is chosen instead of a single point, because then the error induced by the vector field computation will be cancelled out. For the 10 m/s case this lead to a minimum uncertainty in the x-component and the y-component of the velocity, with a 95% confidence interval, of \(\Delta u = 0.008\)
m/s and $\Delta v = 0.009 \text{ m/s}$. Whereas for the 20 m/s case, the maximum uncertainty in the x-component and the y-component of the velocity, also with a 95% confidence interval, was $\Delta u = 0.020 \text{ m/s}$ and $\Delta v = 0.023 \text{ m/s}$. The uncertainty for all cases was much smaller than the free-stream velocity and the turbulent fluctuations and could therefore safely be neglected.
5

RESULTS AND DISCUSSION

In this chapter the results obtained from the measurement campaign are presented. First the testing conditions (e.g. velocities and filling degrees) are defined that will be evaluated. Once this is done, the boundary layer at the cavity leading edge is characterized. These measurements are performed since the incoming boundary layer profile is of utmost importance for the flow over the cavity.

The chapter then proceeds to the PIV measurements, where specific attention will be given to the mixing layer and the underlying recirculation zone. The flow is analysed by covering the flow field, the vorticity in and around the cavity, the mixing layer spreading and the turbulence intensity. This will give an indication on: how the mixing layer and recirculation zone change upon liquid addition, whether the shed vortices in the mixing layer periodically interact with the trailing edge and also whether the interaction may be attenuated by adding liquid.

The investigation starts with an empty cavity with laminar inflow for reference purposes. After this the partially liquid filled cavities with laminar and turbulent inflows are treated. The chapter finally ends with the discussion, where the important features are repeated and some expectations are made regarding to flexible risers.

5.1. TESTING CONDITIONS

During the wind tunnel experiments three different free-stream air velocities were used. For the present work these velocities are 10 m/s, 15 m/s and 20 m/s, corresponding to Reynolds numbers based on the cavity span of: \( Re_{L+} = 2.7 \times 10^4, 4.0 \times 10^4 \) and \( 5.3 \times 10^4 \). The lower bound of the velocity range is chosen at 10 m/s, since from the density difference between gases and liquids no significant interaction is expected in the cavity area for lower velocities. The two higher velocities are chosen to see the effect of a varying Reynolds number (or more specific: varying inertial forces) and to observe how this correspondingly changes the mixing layer behaviour and the gas-liquid interaction in the cavity area. Since it is reasoned that the inertial forces, which are dependent on the velocity, will be dominant for the perceived flow behaviour, large velocity increments of 5 m/s are used.

Additionally, the choice of the aforementioned velocities is backed up by the availability of reference work for empty cavities from Koschatzky [24] and Parkhi [49]. They both used the same cavity structure, which besides the curved leading edge, is similar to the cavity structure in the current study. This implied a shorter cavity span, giving a cavity characterized by a length to depth ratio of \( l/D = 1.5 \). The velocities they evaluated were 10 m/s (Parkhi), 12 m/s and 15 m/s (Koschatzky), which due to the smaller cavity span resulted into Reynolds numbers in the range of \( Re_L = 2.0 \times 10^4 - 3.0 \times 10^4 \).

Near the model surface the flow will form a boundary layer, as was explained in section 2.3. For the three Reynolds numbers PIV measurements will be performed with a laminar and a turbulent incoming boundary layer, since the cavity flow can display a different mode depending on the boundary layer.
For the liquid phase water is selected, due to the low price and the abundant availability. The first PIV measurements on the cavity involve an empty cavity \((L+r)/D = 2.0\), which is evaluated for reference purposes. Following this, the cavity is filled up with water at four different levels, namely: 25\% (5 mm filled; \((L+r)/D = 2.7\)), 50\% (10 mm filled; \((L+r)/D = 4.0\)), 75\% (15 mm filled; \((L+r)/D = 8.0\)) and 90\% (18 mm filled; \((L+r)/D = 20.0\)). These heights were measured with a tape-line, so a maximum uncertainty of 2.5\% (0.5 mm) can be expected with regards to the water height. The lower limit of filling up the cavities was set to 25\%, since evaporation of water is an aspect that arises for flows where air-water interaction is present. As a result of evaporation, the water level was observed to decrease between 0.1 mm and 0.2 mm during the 20 minute measurements. For the lowest water level this gave an error of 4\% in the water height, which when combined with the tape-line error was at the edge of acceptability (< 5\%). As a consequence of this, cavities with water levels lower than 25\% were not evaluated. The maximum water level was taken at 90\%, because at higher filling degrees water droplets were blown out of the cavity for all three Reynolds numbers. Furthermore, the 50\% filling degree was chosen specifically, since Özsoy et al. [48] used a cavity structure with a length to depth ratio of \(L/D = 4.0\) in the Reynolds number range from \(Re_L = 1.6 \times 10^4 - 5.2 \times 10^4\).

In total this resulted into thirty possible configurations. In practice this was lower, since water was blown out of the cavity for: a water level of 90\% at Reynolds numbers higher than \(Re_L + r = 2.7 \times 10^4\) for both a laminar and turbulent boundary layer; and a water level of 75\% at a Reynolds number of \(Re_L + r = 5.3 \times 10^4\) for a turbulent boundary layer. The number of evaluated configuration was therefore reduced to 25.

5.2. INCOMING FLOW
From boundary layer theory, as described in subsection 2.3, the boundary layer profile shapes and thicknesses at the cavity leading edge can be predicted. If the boundary layer at the cavity leading edge agrees with the theory, aberrant changes in the flow behaviour of the partially liquid filled cavities with respect to the empty cavities in the literature, are more likely to be attributed to the presence of the water phase than a faulty incoming flow.

In the present experimental campaign three different Reynolds numbers: \(Re_L + r = 2.7 \times 10^4\), \(4.0 \times 10^4\) and \(5.3 \times 10^4\) will be used, with both a laminar and turbulent nature. To obtain a proper turbulent boundary layer before the cavity leading edge, the laminar boundary layer is forced to transition. The forcing is needed since close to the flat plate leading edge, the momentum thickness based Reynolds number of the boundary layer is lower than the critical value of a turbulent boundary layer [51]. This implies that natural transition has not yet occurred. The forcing, which adds extra momentum thickness to the boundary layer, is applied through a zigzag strip along the model width at a downstream position of 70 mm from the flat plate leading edge. For a Reynolds number of \(Re_L + r = 2.7 \times 10^4\) a zigzag strip of 0.75 mm high seemed to add the necessary momentum thickness, while for the remaining two Reynolds numbers a zigzag strip of 0.6 mm was sufficient.

BOUNDARY LAYER PROFILE
The laminar and turbulent boundary layer profiles that are determined at the cavity leading edge are shown in figure 5.1, together with the Blasius profile (laminar reference) and Power law profile (turbulent reference). From this it is seen that the shape of the laminar boundary layer profiles are in good agreement with the Blasius profile, only close to the wall the profiles deviate. This is because the thermal diffusivity in the near-wall region is increased due to presence of the model, as the Plexiglas surface conducts heat better than air. As a result the heat transfer and the corresponding voltage signal increase, inducing an artificially higher measured velocity in this region. In general this effect should weaken for higher Reynolds numbers, but here the opposite occurs. For the highest Reynolds number of \(Re_L + r = 5.3 \times 10^4\), it is therefore thought that some slight turbulent effects may already be present in the incoming flow. For the turbulent case the profiles satisfy a
Power law profile, but not with the widely used shape factor of \( n = 7.0 \), but with \( n = 4.5 \) (justification will follow later on). Due to the steeper profile of the turbulent boundary layer, the velocities close to the wall are higher than for the laminar boundary layer. Therefore the heat transfer in the near-wall region by forced convection is also higher in this case. The heat lost to the Plexiglas surface is hence a smaller fraction of the total heat loss, which explains why no significant velocity overestimation is seen in the near-wall region for turbulent boundary layers.

![Figure 5.1: Measured boundary layer profiles; left: laminar boundary layer, right: turbulent boundary layer](image)

**Boundary Layer Thickness**

From the boundary profiles the 95% boundary layer thicknesses (\( \delta_{95} \)) can be extracted and are given in table 5.1. Here the second and third column denote the thickness for both the laminar and turbulent situation. The first value in the second and third column is the experimentally measured thickness, whereas the value in between brackets is the theoretical thickness.

<table>
<thead>
<tr>
<th>Reynolds number ( Re_{L+r} ) [-]</th>
<th>Laminar ( \delta_{95} \times 1000 ) [m]</th>
<th>Turbulent ( \delta_{95} \times 1000 ) [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>( 2.7 \times 10^4 )</td>
<td>2.2 (1.9)</td>
<td>3.8 (3.6)</td>
</tr>
<tr>
<td>( 4.0 \times 10^4 )</td>
<td>1.9 (1.5)</td>
<td>3.5 (3.2)</td>
</tr>
<tr>
<td>( 5.3 \times 10^4 )</td>
<td>1.7 (1.3)</td>
<td>3.2 (3.0)</td>
</tr>
</tbody>
</table>

Comparing the values shows that there are some discrepancies between the experimental and theoretical thicknesses. This can be due to the reference height of the hotwire probe, which was determined with a tapeline and thereby induced a maximum systematic error in the experimentally measured thicknesses of 0.5 mm. Moreover, the computation of the theoretical boundary layer thicknesses is also prone to uncertainties, since it is calculated using measured values of: the downstream measurement position, which contains a maximum error of 0.5 mm; the free-stream velocity which contains a maximum measurement error of 0.08% and an user error by setting the wind tunnel at the right velocity; and the fluid viscosity, which shows a maximum error of 0.75% for a temperature increase of 1\( \degree \)C. These can influence the theoretical boundary layer thicknesses up to a difference of 0.1 mm. For the theoretical turbulent boundary layer thickness there is also the uncertainty accompanied with the shape factor magnitude, that may induce an additional error of 0.2 mm. From these contributions it is concluded that the experimental values are within the range of acceptability.

Since the velocities in the boundary layer close to the wall (< 0.5 mm) were not measured, some velocity data was missing. Therefore the momentum thickness was not determined from its definition given in equation (2.12). Instead it was computed using the formulae in equation (2.14),
which relates the 95% boundary layer thickness to the momentum thickness. For the laminar and turbulent boundary layer (under the circumstance of \(n = 4.5\)) the momentum thicknesses become:

<table>
<thead>
<tr>
<th>Reynolds number (\text{Re}_{L+r}) [-]</th>
<th>Laminar (\theta \times 1000) [m]</th>
<th>Turbulent (\theta \times 1000) [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>(2.7 \times 10^4)</td>
<td>0.22 (0.19)</td>
<td>0.48 (0.45)</td>
</tr>
<tr>
<td>(4.0 \times 10^4)</td>
<td>0.19 (0.15)</td>
<td>0.44 (0.40)</td>
</tr>
<tr>
<td>(5.3 \times 10^4)</td>
<td>0.17 (0.13)</td>
<td>0.40 (0.38)</td>
</tr>
</tbody>
</table>

From these turbulent momentum thicknesses the momentum thickness based Reynolds numbers are calculated, which appear to fluctuate between \(\text{Re}_\theta = 3.0 \times 10^2\) and \(\text{Re}_\theta = 5.3 \times 10^2\). In the study of Johnson et al. [23] the lowest evaluated Reynolds number is, however, \(\text{Re}_\theta = 1.0 \times 10^3\). The low magnitude in our experiment is the result of the small distance between the flat plate leading edge and the cavity leading edge, and the small free-stream velocity. The resulting effect is that we have a boundary layer with a different shape than found by Johnson et al. However, since our points lie only slightly outside the range of Johnson et al., this effect is not assumed to be very large and an extrapolation is proposed. Based on extrapolation, shape factors between \(n = 4.0\) and \(n = 4.5\) are expected for these values. As a shape factor of \(n = 4.5\) gave the best fit in our case, the choice for this shape factor was justified.

**CVATY MODE**

By relating the magnitude of the momentum thicknesses to the cavity length, the cavity mode in which the flow will operate is found through the cavity mode parameter \((L+r)/\theta\). From the experimentally found momentum thicknesses in table 5.2 the next cavity mode parameters are determined:

<table>
<thead>
<tr>
<th>Reynolds number (\text{Re}_{L+r}) [-]</th>
<th>Laminar ((L+r)/\theta) [-]</th>
<th>Turbulent ((L+r)/\theta) [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>(2.7 \times 10^4)</td>
<td>181</td>
<td>83</td>
</tr>
<tr>
<td>(4.0 \times 10^4)</td>
<td>211</td>
<td>91</td>
</tr>
<tr>
<td>(5.3 \times 10^4)</td>
<td>235</td>
<td>100</td>
</tr>
</tbody>
</table>

From here it is seen that all the laminar cavity parameters in table 5.3 have a magnitude higher than \((L+r)/\theta = 81\), meaning that none of them should be in the non-oscillating mode. They, unfortunately, do also have values that are outside the shear layer mode interval \((81 \leq (L+r)/\theta \leq 120)\). Though it is not expected to encounter the wake mode for these cases, but just the shear layer mode. The argument in favour of this statement is that a fully two-dimensional flow configuration is necessary for the wake mode to occur. Since the cavity has a finite width and thus side edges, three-dimensional disturbances are created in the flow, that prevent the possible occurrence of the wake mode. For the referenced works of Koschatzky and Parkhi the cavity mode parameter \(L/\theta\) varied between \(92 \leq L/\theta \leq 112\).

The intervals of the cavity mode parameter, corresponding to a certain mixing layer behaviour, were originally found for laminar incoming boundary layers. Hence the intervals give no indication of the mixing layer behaviour for the turbulent incoming boundary layers. They are only mentioned here for completeness.
5.3. EMPTY CAVITY FLOW FOR LAMINAR BOUNDARY LAYERS

In this section the empty cavity is treated for all three Reynolds numbers. This is primarily done, so that the differences with respect to the partially liquid filled cavities can be made clear later on. Furthermore it also gives us a chance to characterize the flow features of the cavity flow, as described in chapter 3, and to see how good the resemblance is to the aforementioned references in the literature, despite the curved leading edge.

FLOW FIELD

Plots of the time-averaged flow field in and around the cavity are presented in figure 5.2 through the velocity field and the streamlines. From all the contour plots the presence of a mixing layer, that originates from the cavity leading edge and spans the cavity, is observed. If a closer look is given to the velocity in the mixing layer, it is seen that it varies between 0.4 - 0.7 times the free-stream velocity, as found in the literature. Also it is non-symmetric around the cavity lip line (i.e. \( y/D = 0 \)) and preferentially spreads into the lower velocity stream [50]. Near the cavity trailing edge the mixing layer flow is diverted into the cavity area. This gives rise to a velocity pattern in the cavity consisting of: a downward jet-like flow at the trailing edge, an upward jet-like flow near/at the leading edge and a large clockwise rotating recirculation cell occupying the complete cavity height, which is in agreement with what was told in chapter 3. At the bottom a wall jet-like flow should also be present, but due to laser light reflections of the bottom wall this region is excluded in the raw PIV images. At the location where the flat plate passes into the curved leading edge and the top corner of the trailing edge, such reflections were also present and lead to spikes in the velocity distribution. This effect is, unfortunately, also visible in the upcoming figures, because as shown in the previous chapter all the computations regarding fluid mechanics phenomena are made using

Figure 5.2: Non-dimensional mean velocity magnitude and streamlines; top left: \( \text{Re}_{L+r} = 2.7 \times 10^4 \), top right: \( \text{Re}_{L+r} = 4.0 \times 10^4 \) and bottom: \( \text{Re}_{L+r} = 5.3 \times 10^4 \)
the velocity field. The drawback is that these two points also influence the streamline computation, since non-physical wiggles will always form right above those points in figure 5.2 (and in all the remaining figures in the report).

Upon comparing the flow and streamlines for the lowest two Reynolds numbers to the work of Parkhi in figures B.6 and B.7 and Koschatzky in figures B.1 and B.2 (see appendix B) a similar structure for the overall flow is seen. Only at the leading edge some differences are perceived, probably caused by the different leading edge shape of the cavity. Firstly, this included the different mixing layer behaviour at the start and secondly for both Reynolds numbers no smaller secondary recirculation cell is discerned at the left bottom corner of the cavity. On a quantitative base the velocity distribution in the mixing layer and in the cavity area (with exception of the area close to the leading edge) is also in accordance with the two references, showing approximately equal velocity magnitudes.

If the flow fields for different Reynolds numbers are compared to each other, it is seen that for higher Reynolds numbers the mixing layer does not separate immediately after the curved part of the leading edge starts. Instead it is more affected by the curvature of the leading edge. As a consequence of this, the mixing layer is deflected more downward and dips deeper into the cavity for higher Reynolds numbers. The fact that the flow wants to follow the downward bend at the leading edge can be explained by the Coandă effect [46]. However, such a downward bend is also accompanied with an adverse pressure gradient. For low Reynolds numbers the flow does not contain enough (kinetic) energy to counter this resistance and therefore separates almost immediately. But as the Reynolds number is increased, the (kinetic) energy of the boundary layer increases as well and it will be able to follow the curved edge somewhat longer.

Furthermore, when looking into the cavity area, the centre of the recirculation cell is seen to move upstream for higher Reynolds numbers. It is located at \( \frac{x}{(L+r)} = 0.72 \) for the lowest Reynolds number and at \( \frac{x}{(L+r)} = 0.47 \) for the highest Reynolds number. This observation is in accordance with the findings in the literature [47, 48].

**Vorticity**

Besides the mean magnitude and the streamlines, the vorticity is also covered. This is done, since overall a more clear representation of the mixing layer and the vortex shedding is attained by vorticity distributions than could be exhibited by the velocity field alone. When looking at the mean vorticity distributions for the different Reynolds numbers, which are depicted in figure 5.3, it is seen that the mixing layer emanates from the upstream boundary layer and forms the core of the mixing layer that is convected downwards. Since the velocity in the mixing layer is higher at the top than at the bottom, a clockwise rotation is induced on the fluid parcels. Because of our convention this leads to a negative vorticity in the mixing layer. In the same way the positive vorticity contribution on the cavity edges (bottom edge not visible in figure 5.3) can be explained. These regions are also the places where the maximum negative vorticity (mixing layer) and the maximum positive vorticity (cavity trailing edge) are located. This is in accordance with the observed results from Bian et al. in figure 3.4.

As the distance of the mixing layer to the trailing edge decreases, the negative magnitude of the mixing layer core vorticity becomes smaller. This has two reasons, which have to do with the size of the main recirculation cell. Firstly, because the velocities above and below the mixing layer located at the part of the mixing layer near the recirculation cell differ less from each other. This induces lower velocity gradients. In contrast, near the leading edge (further away from the recirculation cell) the fluid in the cavity area is almost stagnant, whereas the free-stream velocity is unchanged. Therefore larger velocity gradients and a higher vorticity occur. The second reason is related to the positive vortical structures that are diverted into the mixing layer by the recirculation cell (see red structures in the instantaneous vorticity distributions in figure 5.4). This also creates a lower negative magnitude of the mixing layer core vorticity in the downstream direction.
The small counter clockwise rotating vortices in the cavity area in figure 5.4 originate from the positive vorticity on the cavity trailing edge, cavity bottom edge and sometimes also from the cavity leading edge and are entrained by the mixing layer, which was also remarked by Haigermoiser et al [20]. These positive vortical structures are quite important, because they can be seen as disturbances. Hence mixing layer oscillations and shed vortical structures quickly arise after absorption of these disturbances due to amplification, as was explained in chapter 3. From examination of the instantaneous (non-time resolved) vorticity distributions in figure 5.4 the Kelvin Helmholtz instability, that is amplifying the disturbance, can be seen at work through the formation of vortical structures. This demonstration does make us believe that the mixing layer flow for the lowest Reynolds number of $Re_{L+r} = 2.7 \times 10^4$, and also for the middle Reynolds number of $Re_{L+r} = 4.0 \times 10^4$, is actually in the shear layer mode, since well-defined vortical structures are formed. Though no certainty in the periodicity can be gained, as we did not performed time-resolved PIV. For the middle Reynolds number the vortical structures, as displayed in figure 5.4, are larger than for the lowest Reynolds number and also two instead of one well-defined vortical structure are simultaneously visible. This point was also observed when comparing instantaneous vorticity distributions from Parkhi (figure B.8) to Koschatzky (figure B.3). For the middle Reynolds number the vortical structures also contain a similar centre vorticity as found by Koschatzky.

In contrast, for a Reynolds number of $Re_{L+r} = 5.3 \times 10^4$ initially large well-defined vortical structures are formed and the mixing layer flow is in the shear layer mode. During the downstream convection the vortical structures become unstable and break up in smaller parts. As the resulting structures resemble the structures seen in the work of Koschatzky for a turbulent inflow, it is proposed that they are ill-defined vortical structures and that the mixing layer flow is in the non-oscillating mode. The break up is supposedly caused by the slight turbulent influences that were
found in the laminar boundary layer for this Reynolds number. Therefore the mixing layer flow contains some features of the mixing layer formed by a fully turbulent boundary layer, which as will be shown later on, is in the non-oscillating mode. In this case these turbulent influences have a destabilizing effect on the vortical structures and result in break up. Though one has to note that turbulent influences are not necessarily destabilizing to vortical structures, as there is also a region where mixing layers formed by turbulent flows contain well-defined vortical structures. Since the flow is in the non-oscillating mode, the vortical structures should be more random in nature and thereby lack a periodic interaction with the trailing edge. This could, unfortunately, again not be found through our measurements as we did not do time-resolved PIV. What we could see was that the size of these structures was irregular, but always smaller than found for the lower two Reynolds numbers and therefore less strong. Furthermore, they were widely distributed over the cavity area, making the diversion of the structures into the cavity without impinging on the trailing edge also belong to the possibilities.

**VORTICITY THICKNESS**

The development of the vorticity thickness for a mixing layer formed by a laminar incoming boundary layer is initially characterized by an exponential growth rate. After a while the flow will turn turbulent and the growth rate will become linear. The shorter this regime of exponential growth, the faster the mixing layer will transition to a turbulent state, where mixing layer oscillations and the formation of vortical structures are promoted.

From the evolution of the non-dimensional vorticity thickness (scaled with the respective non-dimensional momentum thickness in table 5.2) along the cavity lip line in figure 5.5, the exponential growth rate regime and the linear growth rate regime are clearly perceivable. The exponential growth rate regime decreases in length upon increasing the Reynolds number, while the linear
growth rate regime increases in length. This implies earlier contact of the mixing layer with vortical structures at higher Reynolds numbers.

For the exponential growth rate regime mostly qualitative comparisons are done. The initial part of the three mixing layers spanning an empty cavity is exponential, as is proved in the insert in figure 5.5, and demonstrates the same behaviour as observed in the literature. In contrast, for the linear growth rate regime a quantitative analysis is performed. In the present study the linear growth along the streamwise direction is \[ \frac{d\delta_\omega}{dx} = 0.19 \text{ (Re}_{L+r} = 2.7 \times 10^4), \] \[ \frac{d\delta_\omega}{dx} = 0.28 \text{ (Re}_{L+r} = 4.0 \times 10^4) \] and \[ \frac{d\delta_\omega}{dx} = 0.31 \text{ (Re}_{L+r} = 5.3 \times 10^4). \] From a comparison of linear growth rates in the literature, it is found that the linear growth rate of the mixing layer formed by laminar incoming flows normally varies between \[ \frac{d\delta_\omega}{dx} = 0.10 \text{ - } 0.20 \text{ [3, 8, 20].} \] For the lowest Reynolds number the magnitude of the growth rate lies inside this main interval, though for the other two Reynolds numbers the values are slightly outside this interval. A possible reason may be the Coandă effect that deflects the higher Reynolds number flows more downwards, which enables larger spreading.

Figure 5.5 also shows that the mixing layer will increase in peak vorticity thickness for larger Reynolds numbers, which agrees with the trends in the literature [3]. It is thought that this is debit to the higher velocities in the cavity area for higher Reynolds numbers. According to the work of Koschatzky [24], higher velocities of the recirculation cell will lead to a lower core pressure of the recirculation cell. This attracts the mixing layer flow more downwards and thereby promotes a larger growth rate. This in combination with the earlier start of the linear growth rate for higher Reynolds numbers will eventually result into a higher peak vorticity thickness.

**Turbulence Intensity**

Next up is the turbulence intensity as shown in figure 5.6. From this it is observed that the turbulence intensity peaks in the mixing layer at the height of the cavity lip line, near the cavity trailing edge and on the flat plate section behind the cavity. The higher intensity in the mixing layer results from the oscillations in the mixing layer and the formation of vortical structures. The high
values near the trailing edge are caused by the vortex-wall interaction and the diversion of vortical structures into the cavity. The high magnitude after the trailing edge is the result of vortical structures that do not or partially impinge on the trailing edge and hence escape from the cavity. All of these are paired with large fluctuations in the x-component and the y-component of the velocity [47]. When increasing the Reynolds number, the turbulence intensity structure reaches further upstream. This increase is attributed to the faster transition of the flow to the linear growth regime at higher Reynolds numbers, which was also found through the vorticity thickness. As a consequence, velocity fluctuations appear earlier.

For both the lower and middle Reynolds number the turbulence intensity in the mixing layer increases as the trailing edge is approached. This eventually generates the highest turbulence intensity at the top corner of the trailing edge, where the well-defined vortical structures impinge on. The higher turbulence intensity in the mixing layer for the middle Reynolds number compared to the lower Reynolds number is the combined effect of an earlier and higher growth rate. Under this circumstance of well-defined vortical structures, this will lead to larger sized structures. This consequently increases the turbulence intensity in the mixing layer. Furthermore, the turbulence intensity in the cavity is also higher for the middle Reynolds number situation, owing to the larger, more energetic structures that are diverted into the cavity. Comparing the middle Reynolds number situation to the similar conditioned turbulence intensity plots obtained by Koschatzky in figure B.4, shows that the turbulence intensity structures have the same distribution and shape in both cases.

For the highest Reynolds number the turbulence intensity distribution in the mixing layer changes. Instead of increasing turbulence intensity when approaching the trailing edge, the turbulence intensity now peaks around $x/(L+r) = 0.4$ and then only decreases. As a result of this, the highest turbulence intensity does not occur near the trailing edge but near the leading edge. This seems strange
5.3. Empty cavity flow for laminar boundary layers

as the linear growth rate started earlier and was also higher than in the previous situations. The change is proposed to be related to the appearance of ill-defined vortical structures. Firstly, these structures break up in smaller structures, which will induce smaller velocity fluctuations. This leads to a lower turbulence intensity as one would expect from the previous two situations. Secondly, the vortical structures can also be diverted in the cavity area without impinging on (the top side of) the trailing edge. This will also decrease the turbulence intensity in the mixing layer and at the top part of the trailing edge. It does, however, increase the turbulence intensity in the cavity and the lower located part of the trailing edge.

A common problem with the turbulence intensity is that the individual points require a lot of images to converge to a final value. To show that the turbulence intensity is indeed converged, the velocity is evaluated for the lowest Reynolds number at two points. The first point (henceforth named point A) is located in the free-stream flow before the cavity, whereas the second point (henceforth named point B) is located in the cavity area; the exact location can be seen in figure 5.6. For both points three plots are given in figure 5.7, containing: the velocity over the images, the variation of the mean velocity over the images and the variation of the standard deviation of the velocity (turbulence fluctuation) over the images for the x-component and y-component of the velocity. From figure 5.7 it was found that already after 500 images the mean velocity and the standard deviation of the velocity are converging towards a final value. From the mean velocity at point A and point B it may also be observed that in the latter stages the mean x-component of the velocity slightly decreases, becoming less positive for point A and less negative for point B. This event is probably caused by the wind tunnel, as via the Pitot-static tube the flow output was seen to slightly reduce during the measurements. The resulting deviation in the mean x-component of the velocity was smaller than 0.1% of the velocity magnitude for point A, but for point B this effect was relatively larger due to the smaller mean magnitude. Since the development of the standard deviation was not severely affected by this decrease, the slight decrease was neglected. In the end, the mean and the standard deviation for both points and velocity components show variations with respect to the value hundred images earlier, smaller than 1% of its magnitude. This was deemed sufficient for the present study.
INTERROGATION WINDOW SIZE

In the previous chapter the choice for 32 by 32 pixels interrogation windows was announced. Since the mixing layer, however, contains high gradients, the choice of a certain interrogation window size may influence the results. Therefore the time-averaged x-component of the velocity, the time-averaged y-component of the velocity, the time-averaged vorticity and the turbulence intensity for the lowest Reynolds number are compared for three different interrogation window sizes along a vertical line, together with the vorticity thickness over the cavity span. This is done with the intent to find out what effect the interrogation windows size has on the measured flow properties. The vertical line and the variation of the mentioned flow properties are depicted in figure 5.8. The three interrogation window sizes are chosen as 16 by 16 pixels, 32 by 32 pixels (the standard for this report) and 64 by 64 pixels. In terms of the Kolmogorov length scale of the flow \( \eta = 0.02 \text{ mm} \); based on homogeneous isotropic turbulence with \( U_\infty = 10 \text{ m/s} \) and \( L + r = 40 \text{ mm} \) the window sizes are equal to 20\( \eta \) by 20\( \eta \) (16 by 16 pixels), 40\( \eta \) by 40\( \eta \) (32 by 32 pixels) and 80\( \eta \) by 80\( \eta \) (64 by 64 pixels). Smaller and larger interrogation windows were not used, since smaller interrogation windows lead to an considerable increase of the noise, whereas larger interrogation windows gave too coarse distributions.

Assessing the results shows that the case with the largest interrogation window size has, in vertical sense, three vectors in the steepest gradient area of the mixing layer, whereas the 32 by 32 pixels interrogation window case and 16 by 16 pixels interrogation window case have six and ten vectors in this area. From observation it is concluded that the x-component and y-component of the velocity show roughly the same behaviour. The largest deviation is found for the 64 by 64 pixels interrogation window, as it shows some over- and undershoot at the edges of the mixing layer. The reason for this was found when inspecting the velocity contour plots corresponding to this larger interrogation window size. It shows that the overshoot at the top is caused by the high speed streak on top of the mixing layer, which is also seen in figure 5.8. For the 64 by 64 pixels case this streak extends further downstream, owing to the larger interrogation window size and hence induces the top overshoot. The bottom undershoot can be explained in a similar manner. Here the low speed zone close to the trailing edge extends further upstream, due to the larger interrogation window size and thereby induces the bottom undershoot.

Figure 5.8: Variation of properties with the interrogation window size; top row from left to right: vertical line along which the properties are evaluated, the time-averaged x-component of velocity and the time-averaged y-component of velocity, bottom row from left to right: the time-averaged vorticity, the turbulence intensity and the vorticity thickness,
When the velocity data is used to calculate the time-averaged gradients and fluctuations for the vorticity and turbulence intensity, the small variations in space and also over the images lead to larger differences. For the vorticity this only induces a notable change in the peak magnitude of the mixing layer. Additionally, in the free-stream area and the cavity area the vorticity for the 16 by 16 pixels case displays some slight oscillations around the vorticity of the other two interrogation window sizes. This is due to the increase of noise coupled to reducing interrogation window size. For the turbulent intensity the peak magnitude is increased for smaller interrogation window sizes, but also the turbulent intensity values outside the mixing layer are slightly affected. For smaller interrogation windows the values outside the mixing layer have a shift towards the right-hand side. This stems from the increased noise at smaller interrogation window, that increases the base value of the fluctuations in the flow. Although the base value of the turbulent intensity is increased in the area outside the mixing layer, it still follows the overall behaviour demonstrated by the other two interrogation window sizes.

From the above it is concluded that the peak magnitude for the 32 by 32 pixels interrogation window case are under- and overestimated for, respectively, the larger and smaller interrogation window sizes. It seems that upon decreasing the interrogation windows size the peak magnitude increases and will eventually reach a final value. Via Richardson extrapolation \[62\] in combination with the data of all three interrogation window sizes an error estimate can be obtained to find out how much the peak magnitude in the 32 by 32 pixels case differs from ‘real’ peak magnitude. From the peak vorticity magnitudes in figure 5.8, it is calculated that the peak vorticity in the 32 by 32 pixels case shows an error of 12% (based on the peak vorticity for the 32 by 32 pixels interrogation window) with regards to the ‘real’ peak magnitude. The turbulent intensity shows a 32% error in peak magnitude. From this it is seen that it is possible to compare the peak vorticity magnitudes with different articles in literature. For the peak turbulent intensity magnitudes one should be more cautious and preferably only compare those qualitatively, due to the large error.

For the growth of the vorticity thickness it is seen that the base magnitude of the vorticity thickness is affected by the choice of the interrogation windows size. As the interrogation windows are chosen larger, the maximum value of the velocity gradient \(\frac{\partial u}{\partial y}\) becomes lower due to averaging out over a larger area. Whereas the maximum magnitude of the velocity gradient becomes larger when smaller interrogation windows are used, due to a smaller area over which they are averaged out. The consequence is that the base value of the vorticity thickness, which depends upon the inverse of the velocity gradient, is larger for larger interrogation windows. But since our interest is in the growth of the vorticity thickness and not in the base value, this not a problem. Concerning the growth, it is seen that as the interrogation window size decreases, the exponential growth rate regime prevails longer at the expense of the linear growth rate regime. The steepness of the growth rate does show a small increase after decreasing the interrogation window size from 32 by 32 pixels (cf. \(\frac{d\delta}{dx} = 0.19\) and \(\frac{d\delta}{dx} = 0.22\)). Though as this change is about 10%, the previous comparison with the literature is still justified. Another point of attention is the sudden bump in the vorticity thickness for the 64 by 64 pixels interrogation window case between \(0.1 < \frac{y}{(L+r)} < 0.4\). This is a combined effect of: the curved leading edge that momentarily deflects the flow downward, until separation, and the large size of the interrogation windows. As a consequence of this, the mixing layer will become wider in this section and the velocity gradient will decrease, inducing a higher base value of the vorticity thickness. After separation the mixing layer will again reduce in width, leading to a higher velocity gradient and a smaller base value of the vorticity thickness. For the smaller interrogation windows this effect is less pronounced, due to the smaller size of the interrogation windows.

5.4. PARTIALLY LIQUID FILLED CAVALITY FLOW FOR LAMINAR BOUNDARY LAYERS
The main purpose of the present study is the work done on partially liquid filled cavities, with which the present section will start for the laminar incoming boundary layers. First the behaviour of the
water film as a reaction to the air flow is covered. Thereafter the focus is switched to the air flow.

5.4.1. WATER FILM

As shown in the previous section for empty cavities, the air flow at the trailing edge of the cavity will be diverted into the cavity and forms a recirculation zone of varying complexity. After diversion, the air flow will at a certain depth come into contact with the water film. Here the air flow interacts with the water flow and may possibly displace it. This creates a deviated water surface with respect to the static situation, as was seen in the study of Belfroid et al [6]. The reason that this occurs can be explained from a simple physical approach. At the surface we on one side have the air flow that impinges on the water surface and on the other side we have the water film. In order to displace the water film locally, a hydrostatic pressure gradient of \( \frac{dp}{dx} = 9.8 \text{ Pa per mm} \) has to be overcome. This shows that theoretically a pressure of \( p = 196 \text{ Pa} \) will blow all the water out of the cavity. Based on the results from the empty cavity, it is found that the impinging flow on the water surface has a velocity between 30% - 50% of the free-stream velocity. The dynamic pressure that the diverted flow contains for the lowest Reynolds number is small, being about \( p_{\text{dyn}} = 15.0 \text{ Pa} \) at best. As the Reynolds number linearly increases, the dynamic pressure increases quadratically, leading to maximum pressures of \( p_{\text{dyn}} = 33.8 \text{ Pa} \) (\( \text{Re}_L + r = 4.0 \times 10^4 \)) and \( p_{\text{dyn}} = 60.0 \text{ Pa} \) (\( \text{Re}_L + r = 5.3 \times 10^4 \)). For higher Reynolds numbers therefore more water is expected to be displaced upstream.

After impingement on the water surface, the air flow is diverted upstream. From a momentum-based approach it can be derived that the air flow will then drag the water stream in the same direction, meaning that the water film is thus also characterized by a recirculation zone. Though since the velocity at the top of the water film is from right to left, it will rotate counter clockwise. Moreover, the shear that exists at the air-water interface will through the Kelvin-Helmholtz instability mechanism, lead to the formation of some small waves on the water surface.

For the present study the cavity was filled at four water levels (25%-90% of the cavity depth), while it was approached by a flow characterized by three different Reynolds numbers (\( \text{Re}_L + r = 2.7 \times 10^4, 4.0 \times 10^4 \) and \( 5.3 \times 10^4 \)). For the lowest Reynolds number the water surface barely reacted to the air flow at any filling degree. As shown in the left picture of figure 5.9 it stayed horizontal for all water levels, but some waves were generated at the air-water interface. This is in agreement with what Belfroid et al. obtained at low velocities. For the other two Reynolds numbers a different behaviour was discerned. Up to a liquid filling of 75%, the water surface looks as depicted in the middle picture in figure 5.9. If we move downstream from the cavity leading edge in this case, the water surface slightly increases till a point close to the cavity trailing edge. Here a water jump took place, across which the water level diminished. This ‘dent’ in the water surface located at the trailing edge, was caused by the strong diverted air flow into the cavity that locally displaced the water film. The dent was larger for a 25% filled cavity than for a 50% filled cavity, as the, to be discussed, diverted air flow into the cavity was higher in magnitude for the lower filling degrees. From visual observation it was also found that the generated waves at the air-water interface became more violent for higher Reynolds numbers. Besides this, they also contained a transverse component. For cavities at the higher two Reynolds numbers, that had a water level equal to 75%, the dent disappeared as is seen in figure 5.9. The water surface now only showed a single inclination, giving a water level at the leading edge that was higher than at the trailing edge. In conclusion, for all filling degrees the water was displaced from the trailing edge, which was similar to what Belfroid et al. found in the experiments at higher velocity. Complete displacement of water from the trailing edge was not achievable in our study, due to the large difference in density between both fluids.

As said, measurements were only performed for the scenarios where water stayed in the cavity. From the measurements it was found that for the filling degrees of 25%, 50%, 75% and 90% water droplets started to be blown out of the cavity, respectively, from a Reynolds number of \( \text{Re}_L + r = 6.5 \times 10^4, 6.8 \times 10^4, 5.8 \times 10^4 \) and \( 3.0 \times 10^4 \). Under the assumption that the impingement velocity is
5.4. PARTIALLY LIQUID FILLED CAVITY FLOW FOR LAMINAR BOUNDARY LAYERS

Figure 5.9: Water surface in the cavity; from left to right: the lowest Reynolds number \( \Re_{L+r} = 2.7 \times 10^4 \) at 25% filling, the highest Reynolds number \( \Re_{L+r} = 5.3 \times 10^4 \) at 25% filling and the highest Reynolds number at 75% filling

50% of the free-stream velocity and displaces only the last downstream quarter of the water surface, an elevated height of the first three quarters of the water surface is obtained. For each case this height is lower than the distance between the cavity lip line and the specific water surface in this case. This is in line with the observations, because otherwise most of the water in the cavity area would be blown out at once. It is therefore thought that some water at the elevated part of the water surface is sheared by the recirculating air flow and carried upward to the leading edge. At this point the droplets will, due to the higher inertia, be passed on to the free-stream flow that subsequently carries the water droplets over the cavity. This principle seems to hold for all water levels that were less than 90%. The difference is that for cavities filled for more than 90% with water, the water left the cavity at the trailing edge. This indicates that the flow behaviour in that case is different from the other cases and that the recirculation zone, formed by a downward flow at the trailing edge, is probably not present here. It is expected that in this case the mixing layer over the cavity will directly atomize some water from the film and thereby transports water out of the cavity. However, more comments on the exact air velocity field in this situation are saved for the next subsection.

Since no high-speed camera measurements were taken and tracer particles were not added to the water film, the surface waves and the water film movement could not be investigated on a more quantitative base. Nonetheless, it is thought that the partially liquid filled cavity will behave as a shallower solid cavity to the air flow, since the density of water is much higher than the density of air. The ‘bottom’ of this cavity can either be smooth, at low velocities, or more inclined at higher velocities. This will of course affect the recirculating flow that develops in the cavity area, as will be shown in the upcoming subsection.

5.4.2. AIR FLOW

For the three Reynolds numbers a cavity filled at 25%, 50%, 75% and in one instance even at 90% were subjected to measurements. In the present subsection only the figures for the Reynolds numbers of \( \Re_{L+r} = 2.7 \times 10^4 \) and \( \Re_{L+r} = 5.3 \times 10^4 \) are depicted. The cavity flow corresponding to the middle Reynolds number of \( \Re_{L+r} = 4.0 \times 10^4 \) is treated, but since varying the water level influences the air flow for the middle Reynolds number in a similar way as for the highest Reynolds number, the figures for the middle Reynolds number are omitted from the present subsection. However, all the figures obtained by post-processing the data from the PIV measurements, including the ones that will not be shown over here, are given in appendix C.

FLOW FIELD

Upon increasing the water level from 0% to 90%, the flow field in and around the cavity changes, as is depicted for the lowest and highest Reynolds number through the velocity magnitude and the streamlines in figure 5.10. In the raw images that leads to these figures, the area that the water film took in was masked out and turned white. As the water surface is affected by the air flow, this may sometimes lead to a non-flat shape of the cavity bottom edge. Besides the white area, also a black dotted line is drawn in the figures, which denotes the static liquid distribution if no air flow would be present. For the lowest Reynolds number flow this line coincides with the bottom edge of the cavity and is therefore not visible. For the higher Reynolds number flow, the actual water level may sometimes lie beneath this line. This is due to a combination of too little water in the
Figure 5.10: Non-dimensional mean velocity magnitude and streamlines; first row: $Re_{L+r} = 2.7 \times 10^4$ at 25\% filled (left) and 50\% filled (right), second row: $Re_{L+r} = 2.7 \times 10^4$ at 75\% filled (left) and 90\% filled (right), third row: $Re_{L+r} = 5.3 \times 10^4$ at 25\% filled (left) and 50\% filled (right) and fourth row: $Re_{L+r} = 5.3 \times 10^4$ at 75\% filled
5.4. Partially liquid filled cavity flow for laminar boundary layers

cavity, evaporation of water from the cavity and the transient, wavy air-water surface. The latter point induces large problems when determining the mean location of the interface. The interface location was therefore consciously under predicted, such that no data of the air flow at the interface was lost.

From analysing the velocity fields, the same global structure was found as for an empty cavity: a mixing layer spanning the cavity, a clockwise rotating recirculation cell and wall jets on the cavity edges (bottom edge is now visible). But some differences are seen with regards to the behaviour and size of these components. For the mixing layer initially no clear difference between the various filling degrees at constant Reynolds number is perceived. This is because at the start, the mixing layer is purely formed based on the inflow, which for all the constant Reynolds number sets is the same. As the mixing layer travels further downstream, the flow is affected by the recirculation area, which was also concluded by Haigermoser et al [19]. It is namely seen that, as before, the mixing layer preferentially grows into the slower bottom stream. The extent with which the mixing layer grows in the bottom stream, differs with the water level (and the Reynolds number). We will, however, refrain from any quantitative statements with respect to the variation in mixing layer spreading per water level, based on visual observations of figure 5.10. This is because the mixing layer and the recirculation zone are intertwined, which hence may lead to misleading conclusions. We will further treat the variation in mixing layer spreading with the water level in the upcoming vorticity thickness subsection. In terms of convection velocity the mixing layer still propagates at a speed between 40% - 70% of the free-stream velocity.

In the internal cavity area changes are also observed. For the lowest Reynolds number the main recirculation cell has the same circular structure as for an empty cavity, since the water film on the cavity bottom is very slightly affected by the air flow. For the higher two Reynolds numbers the water film on the bottom is influenced by the air flow, which results in loss of the circular structure of the recirculation cell. Instead the recirculation cell has a more skewed shape caused by the shape of the water surface. This observation is in line with the aforementioned study of Kuo et al. [26]. For the lowest Reynolds number the recirculation cell diameter scales with the depth of the cavity and eventually vanishes at a filling degree of 90%. This explains why the water in this case was blown out at the back of the cavity. For the middle and higher Reynolds number the main recirculation cell is characterized by its upstream extent rather than its diameter. For the middle Reynolds number the upstream extent is hardly sensitive to the depth, whereas for the highest Reynolds number it is not sensitive at all to the depth. An explanation can be found when looking closer at the internal cavity flow. In the cavity the flow will eventually start moving straight upward, if the resistance induced by the pressure gradient in the cavity is higher than the energy the flow contains. For the lowest Reynolds number the bottom is flat and due to the interplay between the internal cavity velocity and the pressure gradient, the flow will be diverted straight upward faster for smaller depths (i.e. smaller recirculation cells). The velocity in the recirculation cell decreases with the cavity depth, the only exception is the 50% filled cavity. A striking feature in this case is that the internal distribution, containing two large recirculation cells, leads to a higher pressure gradient than before, as the flow is diverted straight upward earlier, despite the higher velocity. For the higher Reynolds numbers, the inertial forces of the flow in the cavity have in absolute sense increased and now only show a decrease in velocity with lowering cavity depth. As a result of the sloped bottom in this case, the flow may already show a upward velocity component at the bottom, but the point where it will flow straight upward almost always extends till close to the leading edge, as the wall induces a pressure gradient here that becomes larger than the inertial force of the flow. This will subsequently cause straight upward flow diversion. The only large deviation is the middle Reynolds number at 75% filling. Due to the lower velocity here, the flow has a lower inertial force that cannot compete with the adverse pressure gradient till the leading edge. This results in a smaller recirculation cell.

Parallels can also be drawn with using a cavity structure of a larger length to depth ratio and
Figure 5.11: Non-dimensional mean vorticity; first row: $Re_{L+r} = 2.7 \times 10^4$ at 25% filled (left) and 50% filled (right), second row: $Re_{L+r} = 2.7 \times 10^4$ at 75% filled (left) and 90% filled (right), third row: $Re_{L+r} = 5.3 \times 10^4$ at 25% filled (left) and 50% filled (right) and fourth row: $Re_{L+r} = 5.3 \times 10^4$ at 75% filled.
5.4. PARTIALLY LIQUID FILLED CAVITY FLOW FOR LAMINAR BOUNDARY LAYERS

filling a cavity up with water to obtain a larger length to depth ratio. By pouring water in the cavity, the centre of the main recirculation cell shows a backward motion as the length to depth ratio increases. By performing tests on cavity structures with varying length to depth ratios, Ashcroft et al. [3] and Oshkai et al. [47] found the same to be true. Although there are some similarities between cavity structures at the same length to depth ratio with and without a water film, the exact internal flow structure remains different. A good example is the earlier mentioned work of Özsoy et al. in figure B.5. The measurement of Özsoy et al. at the lowest Reynolds number can be compared to the currently investigated 50% filled cavity at a Reynolds number of $Re_L = 2.7 \times 10^4$. For our case, the cavity area contains two recirculation cells that are about equal in size and are centred at a similar height. Whereas for Özsoy et al. the cavity area also contains two recirculation cells, but the main recirculation cell is clearly larger and has a higher located centre than the secondary recirculation cell. Besides this, the main recirculation cell in figure B.5 is also stretched in the stream wise direction at the top. So the recirculation cells in both cases are composed and arranged differently. This is thought to be the consequence of the different nature of the bottom edge. For our case this was a liquid bottom and for Özsoy et al. a solid bottom. It is proposed that the air-liquid interaction created a mixing layer on the bottom edge, that was different than the boundary layer on the solid bottom edge. The experiments of Özsoy et al. at a Reynolds number of $Re_L = 3.6 \times 10^4$ and at a higher Reynolds number of $Re_L = 5.2 \times 10^4$ are also similar to the higher Reynolds numbers used in the present study. Upon comparison of the two studies significant differences for the internal cavity flow are found, but these are caused by the different shape of the cavity bottom. Since this influences the internal cavity flow, no legitimate comparison could be made between these cases.

VORTICITY

The changes in the mean vorticity distribution under varying cavity filling levels, are depicted in figure 5.11 for the lowest and highest Reynolds number. Also here the global structure of highly negative vorticity in the mixing layer and highly positive vorticity on the cavity edges, as was seen for an empty cavity, remained intact. If we again start at the mixing layer, it is seen that the mixing layer core vorticity descends from the upstream boundary layer. Therefore the vorticity in the beginning of the mixing layer is similar for the empty and partially liquid filled cavities. But as concluded and discussed in the previous section, the mixing layer is influenced by the underlying recirculation area. This is visible through the attenuation of the mixing layer core vorticity over the stream wise direction (figure 5.12), which is caused by the velocity in the recirculation area and the absorption of positive vortical structures.

When looking closer at figure 5.12 for the lower Reynolds number situation, the attenuation of vorticity initially increases as the filling degree is increased, but later on weakens. This is thought to be related to the variation in the recirculation zone size. For the lower two filling degrees (25% and 50%), namely, a secondary vortex is formed upstream. This is paired with a large region of positive vorticity (see figure 5.11), that can be absorbed by the mixing layer. The increasing size of this region till a 50% filling degree, hence explains the increased attenuation. After a 50% filling degree this secondary recirculation cell and the corresponding positive vorticity region disappear. Therefore the attenuation is only governed by the small main recirculation cell and the positive vorticity it transports from the walls to the mixing layer. As this is less than before and also happens further downstream, the attenuation weakens. For the higher two Reynolds numbers these variations in attenuation of vorticity are much smaller. This is because there are no large changes in the internal flow distribution: the upstream extent of the main recirculation cell shows slight variation with the cavity depth and almost always extends till close to the cavity leading edge. Therefore no significant differences in attenuation appear.

The upstream extent of the positive vortical structures also governs the point where the unsteady mixing layer flow starts. The earlier this occurs, the larger the velocity fluctuations (and the well-defined vortical structures) in the mixing layer are, as was also observed by Kuo et al [26]. This
Figure 5.12: Non-dimensional mean mixing layer core vorticity; left: \( \text{Re}_{L+r} = 2.7 \times 10^4 \) and right: \( \text{Re}_{L+r} = 5.3 \times 10^4 \)

Figure 5.13: Non-dimensional instantaneous vorticity at 50% filling; top left: \( \text{Re}_{L+r} = 2.7 \times 10^4 \), top right: \( \text{Re}_{L+r} = 4.0 \times 10^4 \) and bottom: \( \text{Re}_{L+r} = 5.3 \times 10^4 \)

reasoning also explains that the mixing layer for increased water levels at low Reynolds numbers becomes more stable, as the disturbances are absorbed at a later stage.

Furthermore, one can look at the instantaneous vorticity distributions of the cavity flows to investigate the vortical structures that are shed by the mixing layer at the various filling degrees. For illustration the 50% filled cavities are shown for each Reynolds number in figure 5.13. From all the situations, it was seen that the vortical structures for the lowest Reynolds number are all well-defined. From this it is suspected that the flow at the lowest Reynolds number is always in the shear layer mode. However, for the other two Reynolds numbers the vortical structures all begin as well-defined structures that become unstable and break up in ill-defined vortical structures. These are again more characteristic to the non-oscillating mode. For the middle Reynolds number this is in
5.4. PARTIALLY LIQUID FILLED CAVITY FLOW FOR LAMINAR BOUNDARY LAYERS

contrast to the empty cavity case. This shows that the presence of liquid in the cavity can alter the vortical structures over the cavity. It is supposed that destabilizing influences from the recirculation cell, originating from the air-water interaction (waves) in the cavity area are related to this. They affect the vortical structures in the mixing layer in similar way as the aforementioned turbulent influences such that the vortical structures become unstable. This effect did not occur earlier as the interaction (waves) for the lower Reynolds number was smaller. The well-defined vortical structures broke up in ill-defined vortical structures, but for these laminar cases no clear trend in the size of the ill-defined vortical structures was found upon varying the Reynolds number and the filling degree.

Moreover, these instantaneous vorticity distributions also tell that the shedding of vortical structures occurs over time for almost all measurements. The only exception is the case at a Reynolds number of \( \text{Re}_{L+r} = 2.7 \times 10^4 \) with a 90% water filling. Here the mixing layer stays continuous from the leading edge to the trailing edge of the cavity. Looking closer at this specific case tells that the present cavity is only 2 mm deep, which is smaller than the initial boundary layer thickness at this speed. It is thought that as a result of the lack of space on the bottom to expand in, the flow does not really see that cavity as an gap and just flows over it. The demonstration that the mixing layer stays continuous also backs up the findings of Rockwell et al. [53], who reported that for cavity depths smaller than the boundary layer thickness no vortex shedding took place.

VORTICITY THICKNESS

Next the vorticity thickness for the mixing layers spanning the partially liquid filled cavities is analysed. In figure 5.14 the development of the non-dimensional vorticity thickness for the empty and partially liquid filled cavities is depicted for the constant Reynolds number sets.

For the lower Reynolds number the peak vorticity thickness, is seen to vary with the water level.

Figure 5.14: Non-dimensional vorticity thickness; top left: \( \text{Re}_{L+r} = 2.7 \times 10^4 \), top right: \( \text{Re}_{L+r} = 4.0 \times 10^4 \), bottom left: \( \text{Re}_{L+r} = 5.3 \times 10^4 \) and bottom right: growth rate for various filling degrees at the three Reynolds numbers.
The mixing layer increases in spreading until a water level of 50%; after this point the maximum spreading will only decrease. The change in peak vorticity thickness over the cavity span is dependent on: the size of the linear growth rate regime and the magnitude of the linear growth rate. From the top left graph in figure 5.14 the linear growth rate regime is discerned to be largest in size for a 50% filled cavity and decreases in the sequence of 25%, 0%, and 75% filled. In the case of a 90% filling degree the linear growth rate regime is not even entered, as there is no amplification of disturbances and shedding. As the linear growth rate is larger than the exponential growth rate, this already explains for a large part why the peak vorticity thickness ranks in this order. The variation in the starting point of the linear growth rate regime can be related to the upstream influence of the positive vortical structures in the cavity area. As they influence the mixing layer further upstream, as seen for lower filling degrees, the mixing layer will transition earlier to the linear growth rate regime.

The linear growth rates for the partially liquid filled cavities are plotted in the bottom right graph of figure 5.14 and also show that the growth rate for the lowest Reynolds number ranks in decreasing order as: 50%, 25%, 0% and 75% filled. For this Reynolds number the changes in cavity depth initially (from 0% filled to 50% filled) positively affect the growth rate of the mixing layer and thereafter negatively affect the growth rate of the mixing layer. The reasons for these variations are thought to be related to the different recirculation area. Firstly, for higher speeds of the main recirculation cell, a lower pressure core is expected. A lower pressure will attract the mixing layer flow more downwards and thereby stimulate a higher linear growth rate of the mixing layer. Secondly, for higher water levels a higher located recirculation cell is obtained that counteracts downward expansion of the mixing layer. In response to this, a lower growth rate of the mixing layer is obtained. For filling the cavity until 50%, the main recirculation cell decreases in size. Though the flow velocity in the cell increases, inducing a lower pressure that explains the larger growth rate of the mixing layer until this filling degree. The effect of the higher located recirculation cell is not yet felt at this point. Thereafter the main recirculation cell decreases even more, but now the velocity in the cell decreases as well, leading to a lower growth rate of the mixing layer. It is thought that since the 75% filled cavity has a lower peak vorticity thickness than the empty cavity, but a similar growth rate, now the effect of the higher located recirculation cell is also felt.

For the higher two Reynolds numbers, the peak vorticity thickness overall decreased with increasing the filling degree. The only exception was the empty cavity at the middle Reynolds number. For these higher Reynolds numbers the upstream extent of the recirculation cells is relatively constant and the variation in peak vorticity thickness can mainly be explained by the variation in the linear growth rate.

Again looking at the bottom right graph of figure 5.14 shows that for the higher two Reynolds numbers, the growth rate of the 25% filled cavities is about equal to the growth rate of the empty cavities. For the highest Reynolds number this explains why the peak vorticity thickness in these cases do not differ by much. Though as the velocity in the main recirculation cell decreased and the liquid filling degree increased in both cases, the equal growth rates for these situations could not be explained from the previous arguments. It is thought that here also the changed recirculation zone by the changed cavity bottom plays a part. The far lower peak vorticity thickness for the middle Reynolds number flow over an empty cavity, that increased upon filling the cavity till 25%, was reasoned to be caused by the different nature of the vortical structures. Upon filling the cavity, the vortical structures transformed from well-defined to ill-defined. As the ill-defined vortical structures are spread over a wider spectrum, when compared to well-defined vortical structures, they also lead to a larger mixing layer spreading that saturates later on.

When further increasing the water level, decreases in the growth rate and the peak vorticity thickness were found. These changes were again conform to what was expected from the lower velocity in the recirculation cell and the higher liquid filling. The reduction in both the growth rate and the peak vorticity thickness became significant for the middle Reynolds number at 50% filling.
and for the highest Reynolds number at 75% filling. It is thought that from here on the higher located recirculation cell affected the mixing layer as well.

When comparing the different partially liquid filled cavity configurations to each other, the spreading increases with the Reynolds number, in both a non-dimensional and absolute sense. This is supposedly because the low pressure core of the recirculation zone becomes lower in magnitude for higher recirculation velocities.

**Turbulence Intensity**

Finally the turbulence intensities are presented for the partially liquid filled cavities in figure 5.15. Here only the 50% and 75% filled cavities are shown for the lowest and highest Reynolds number, the 25% filled cavities are, however, given in appendix C as these distributions were very similar to the 50% filled cavities. For a filling degree of 90% (also pictured in appendix C) no significant fluctuations are present and this configuration will therefore initially be disregarded. When looking at the turbulence intensity structures and comparing them to the ones found for empty cavities in figure 5.6, globally the same behaviour with large turbulence intensity in the mixing layer, at the cavity trailing edge and on the flat plate section behind the cavity, is demonstrated for all situations. Though as water is added, the turbulence intensity in and around the cavity becomes higher.

Concerning the upstream extent of the turbulence intensity structures for partially liquid filled cavities, it can be mentioned that the upstream increase is in accordance with the observation of a larger linear growth rate regime. This is logical, as from that point the large velocity fluctuations appear. Since for the lowest Reynolds number there are serious differences in the transition point to the linear growth rate regime, the size and beginning of those structures varies per water level. For the higher two Reynolds numbers the transition to the linear growth rate regime occurs almost

![Figure 5.15: Turbulence intensity; top left: \(\text{Re}_{L+r} = 2.7 \times 10^4\) at 50% filled, top right: \(\text{Re}_{L+r} = 2.7 \times 10^4\) at 75% filled, bottom left: \(\text{Re}_{L+r} = 5.3 \times 10^4\) at 50% filled and bottom right: \(\text{Re}_{L+r} = 5.3 \times 10^4\) at 75% filled](image-url)
always at the same point and therefore there are no significant differences in the structure size.

Upon increasing the length to depth ratio of the cavity by adding water, a different behaviour is found in the mixing layer for the lower Reynolds number and the higher two Reynolds numbers. For the lower Reynolds number, the peak turbulence intensity and the high turbulence intensity region in the mixing layer initially increases with depth and thereafter decreases. The higher two Reynolds numbers show a monotonic increase in the peak turbulence intensity and the high turbulence intensity region. For the peak turbulence intensity this behaviour is also graphically depicted in figure 5.16. From this it also seen that the magnitude of the peak turbulence intensity for the different Reynolds numbers is close to each other.

The behaviour described above and depicted in figure 5.16 can be explained from the nature of the shed vortical structures. For the lower Reynolds number only well-defined vortical structures are present. By lowering the depth, initially larger and then smaller vortical structures are formed. Initially this results in increased velocity fluctuations, that induce a higher peak turbulence intensity and a larger high turbulence intensity region in the mixing layer. Followed by a decrease in velocity fluctuations, that induce a lower peak turbulence intensity and a smaller high turbulence intensity region in the mixing layer.

For the higher Reynolds numbers all the vortical structures start at the same point and break up in smaller parts. These smaller structures are widely spread and can also end up in the cavity area without impinging on the cavity trailing edge. However, as the cavity depth decreases, the underlying space to spread in decreases as well. This gives a higher turbulence intensity in the area that is available for spreading and thereby leads to an increase of the peak turbulence intensity and the region of high turbulence intensity at the location of the mixing layer and the cavity trailing edge. It is also noted that, in contrast to the empty cavity with ill-defined vortical structures, the peak turbulence intensity that the mixing layer attains, is located more downstream. This also stimulates a higher turbulence intensity at the trailing edge.

For the 90% filled cavity at the lowest Reynolds number the flow hardly exhibits any fluctuations that may cause a significant turbulence intensity. Based on the vorticity thickness for this case, which stayed in the exponential growth rate regime, there was already an indication that the flow might still be laminar. From the turbulence intensity this statement is enhanced even more.

5.5. Empty and Partially Liquid Filled Cavity Flow for Turbulent Boundary Layers

In this section the empty and partially liquid filled cavities for turbulent incoming boundary layers will be investigated. For the turbulent inflow again measurements are performed at the same four cavity filling degrees and the same three Reynolds numbers.

5.5.1. Water Film

When we visually examine the water film for the low Reynolds number case, no difference in the global behaviour is seen for the partially liquid filled cavities at a 25% and a 50% water level in comparison to the laminar case. Above a filling degree of 50% the water film, however, demonstrates a different behaviour. At these filling degrees the flat surface disappears and makes room for a slightly inclined surface, as seen in the left picture in figure 5.17. In all configurations the surface
also features small waves, caused by the Kelvin-Helmholtz instability mechanism. For the middle and higher Reynolds numbers the characteristic shape of the laminar situation remained intact, but the ‘dent’ at the trailing edge was much smaller than for the laminar situation (see middle picture in figure 5.17). This profile was presented until a filling degree of 50%. For a filling degree of 75% again an inclined water surface appeared (right picture in figure 5.17) for both Reynolds numbers. Though since for the highest Reynolds number water was blown out of the cavity, measurements were not made in this case. Moreover, the above shows that also for the turbulent inflow the water surface contour is in agreement with the findings of Belfroid et al [6].

When shifting our attention towards the Reynolds number at which water droplets were blown out of the cavity, some surprising results were observed. Firstly, for the filling levels of 25% and 50%, water could not be blown out of the cavity. We should remark that the maximum Reynolds number to which the cavity was subjected, was around \( \text{Re}_{L+r} = 8.0 \times 10^4 \) (corresponding to a velocity of approximately 30 m/s), after this it was not deemed safe to continue. Secondly, for the 75% filling degree it was already mentioned that the highest Reynolds number at which PIV measurements were performed, would blow water droplets out of the cavity. From additional tests it was seen that the blow out Reynolds number was \( \text{Re}_{L+r} = 4.2 \times 10^4 \), which was only slightly higher than the middle Reynolds number at which we performed PIV measurements. Finally, for the highest filling degree of 90% the flow did not hold on for long again, as a Reynolds number of \( \text{Re}_{L+r} = 2.8 \times 10^4 \) was sufficient. For all these situations, with exception of the 90% filling degree, the water also left the cavity at the leading edge. Based on experience from the previous section, this denotes the presence of a recirculation cell in these instances.

So with respect to the laminar situation, the blow out Reynolds numbers severely increased till a filling degree of 50%, but after this it decreased. A clear explanation for this behaviour was not found, as the, to be discussed, turbulent internal cavity flows are globally the same to the laminar internal cavity flows.

5.5.2. AIR FLOW
In this subsection the turbulent air flow in and over empty and partially liquid filled cavities will be visualized. The focus will mainly be on the lower Reynolds number. Since the nature of the mixing layer is the same for all Reynolds numbers and because the higher two Reynolds numbers lead to a similar flow as observed for their laminar analogue. Figures of the higher two Reynolds number flows are therefore found in appendix C.

FLOW FIELD
Just as for a laminar inflow the cavity is characterized by: a mixing layer originating from the leading edge, a clockwise rotating recirculation cell in the cavity area and a jet-like flow on the cavity edges. Though also differences appear in the distribution of the components, with regards to the laminar measurements.

For the turbulent case the flow tends to follow the convex leading edge for a (slightly) longer distance, in comparison to the laminar situation. This is the result of the turbulent stream that contains more energy. Hence the adverse pressure gradient, that is paired with following the curved leading edge, is sustained for a longer time. After separation from the leading edge, a deeper located
mixing layer between the free-stream flow and the internal cavity flow is formed. How deep the mixing layer spreading is, depends on the filling degree of the cavity and the Reynolds number of the inflow. Though again no statements are made on the mixing layer spreading based on the flow field. Close to the trailing edge the mixing layer contains a large vertical velocity component, as can be seen from the streamlines. This upward movement is much larger than for the mixing layers originating from a laminar flow and is thought to be the result of the lower located mixing layer. Finally, the mixing layer still has a velocity between 0.4-0.6 times the free-stream velocity.

When we give a closer look at the underlying recirculation area for the lowest Reynolds number, more changes are noticed. Firstly, it is seen that upon increasing the water level the shape of the recirculation cell changes. The circular shape was already lost after a 25% degree of filling and replaced by a more skewed shape for the remaining configurations. The second remarkable fact is that until a 50% filling degree, the recirculation cell will fill up the whole cavity, in contrast to the laminar
case. It is reasoned that this is the result of the more energetic turbulent flow that is diverted into the cavity. Therefore the internal flow at the bottom edge is less prone to separation (e.g. smaller recirculation cells). Thirdly, for a filling degree of 75% another surprising phenomenon is seen. The spreading mixing layer now becomes so large that it severely interacts with the internal cavity flow. As a result the recirculation cell is shifted upstream, close to the cavity leading edge. Although the length to depth ratio of the cavity is not between $9 < \frac{(L+r)}{D} < 12$ and the mixing layer will not collide with the bottom corner of the cavity trailing edge, the appearing flow does look like a transitional cavity flow due to this upstream shifted recirculation cell. For the filling degree of 90%, the flow will again create no recirculating flow.

For the middle and higher Reynolds numbers roughly the same internal flow distribution is observed as for the laminar situation: slanted cavity bottoms and large skewed recirculation cells that almost always reach till close to the cavity leading edge. This is in accordance with the similar look-
ing water surface for these cases and suggests that the influence of the nature of the flow is very small at higher Reynolds numbers.

When comparing to the literature, the general consensus of the recirculation cell moving backwards with increasing water level (and thereby increasing length to depth ratio) was displayed until a 50% filling degree for the lowest and highest Reynolds number and a 75% filling degree for the middle Reynolds number. Since at a 75% degree of filling a transitional-like cavity was formed for the lowest Reynolds and water droplets were blown out of the cavity for the highest Reynolds number. Besides this, Grace et al. [18] confirmed our observation that generally the recirculation cell formed by a turbulent inflow is located further upstream than for a laminar inflow.

**VORTICITY**

The turbulent situation has peak vorticity magnitudes that are in general smaller than for the laminar situation, as the free shear layers for the laminar case are smaller in width and hence contain higher gradients. Albeit these differences, initially the mixing layer core vorticity still starts out the same and is affected when it flows over the recirculation area. This is seen by the varying attenuation of the mixing layer core vorticity for the lowest Reynolds number at the different filling degrees in figure 5.20. For the fillings degrees up to 50% the internal flow distribution is the same and the attenuation is similar. For the higher filling degrees the attenuation weakens (locally) as the main recirculation cell becomes smaller and eventually disappears.

Since the recirculation area, if present, always reaches the upstream edge, the positive vortical structures are immediately absorbed at the cavity leading edge. But after this, no initial period of forming well-defined vortical structures in the mixing layer is seen from the instantaneous distributions in figure 5.20. Instead the mixing layer always directly breaks up into ill-defined vortical

Figure 5.20: Top left: Non-dimensional mean mixing layer core vorticity at $Re_{L+r} = 2.7 \times 10^4$; non-dimensional instantaneous vorticity at 50% filled cavity, top right: $Re_{L+r} = 2.7 \times 10^4$, bottom left: $Re_{L+r} = 4.0 \times 10^4$ and bottom right: $Re_{L+r} = 5.3 \times 10^4$
structures that are transported downstream. This is in line with the non-oscillating mode. In this situation the size of the vortical structures seemed to increase with the Reynolds number, but the variation in size of the vortical structures with the filling degree was too difficult to assess visually.

The above happens up to a water level of 75%, hereafter no shedding appears. For the 90% filled cavity at $Re_{L+r} = 2.7 \times 10^4$ again the boundary layer was very large compared to the cavity depth. As a consequence the air just flowed over the cavity continuously, without the formation of vortical structures.

**Vorticity thickness**

Since the inflow to the cavity is already turbulent, the mixing layer will not display an exponential growth rate regime. Instead it will show a linear growth rate regime right from the start. Therefore the peak vorticity thickness for a constant Reynolds number set can be explained from the linear growth rate, a higher growth rate will thereby lead to a higher peak vorticity thickness.

The growth rates for the empty cavities are: $\frac{d\delta}{dx} = 0.31$ for $Re_{L+r} = 2.7 \times 10^4$, $\frac{d\delta}{dx} = 0.25$ for $Re_{L+r} = 4.0 \times 10^4$ and $\frac{d\delta}{dx} = 0.25$ for $Re_{L+r} = 5.3 \times 10^4$. As the cavity is filled with water, a general trend of a lower linear growth rate at higher water levels is found from the bottom right graph in figure 5.21.

From analysing figure 5.21 further, it can then be stated that: the mixing layer spreading decreased as the filling degree is increased. This is supposedly due to the lower velocities in the recirculation zone and the higher located recirculation zone, that appear for higher filling degrees. Both induce a smaller growth rate and expansion; the non-dimensional mixing layer spreading increases with the Reynolds number. The only outlier to this trend, is the empty cavity at the lowest Reynolds number. While the non-dimensional spreading
increases for higher Reynolds numbers, the absolute spreading decreases for higher Reynolds numbers. It is thought that this is due to the fact that the absolute spreading magnitude does not take the lower initial momentum thickness for higher Reynolds numbers into consideration. Therefore it is more difficult for the higher Reynolds numbers to obtain a larger absolute spreading.

When comparing the laminar and turbulent situations some changes were also seen in the growth rates and the corresponding peak vorticity thicknesses. For the lowest Reynolds number flow, the turbulent situation, in general, had a larger growth rate and a larger absolute spreading than in the laminar situation. Since the main recirculation cell has spread and filled up the complete cavity for the turbulent case, a larger low pressure area is created that is more favourable for a larger growth rate. Combining this with the lower located centre of the recirculation zone, eventually results into a larger peak magnitude. The non-dimensional spreading of the turbulent case was, however, lower, due to the different momentum thickness used for scaling. For the higher two Reynolds numbers the laminar and turbulent situation showed approximately the same absolute peak vorticity thickness, since the internal recirculating flow and the bottom contour were similar. The non-dimensional spreading of the turbulent case was again lower, due to the way of scaling. Regarding the linear growth rate, a smaller magnitude was found for the turbulent situation. This is explained by the vanishing of the exponential growth rate regime for the turbulent situations. Therefore the linear growth rate regime now covers the whole cavity span. Since the peak vorticity thickness is similar for both situations, a smaller growth rate then suffices for the turbulent situations.

For the 90% filled cavity at the lowest Reynolds number, again a deviated behaviour is discerned because of the high water level. The mixing layer showed repeatedly to vary between growth and no growth. The behaviour is linear between $0 < x/(L+r) < 0.2$, but as it is hindered by the water film it will not grow for a while after the point $x/(L+r) = 0.2$. Though as seen in the figure 5.18 the water level decreases shortly after this point, again allowing a linear growth rate till the point $x/(L+r) = 0.4$. In the region between $0.6 < x/(L+r) < 0.8$ a second decrease in the water level is found, which once more allows a linear growth over a small distance.

**Turbulence Intensity**

Finally the turbulence intensity for the turbulent boundary layer configurations are treated. Overall the same turbulence intensity structures are expected as for the laminar situation with ill-defined vortical structures. These have high turbulence intensity at the location of the mixing layer, the cavity trailing edge and the flat plate section behind the cavity. In figure 5.22 the turbulence intensity for the Reynolds number of $\text{Re}_{L+r} = 2.7 \times 10^4$ was presented at the filling levels of 0%, 50% and 75%. As the shape of the turbulence intensity structure for the 25% filling degree resembled the structure for the 50% filling degree, it was not pictured here but in appendix C. For the 90% filled cavity now some larger fluctuations were presented due to the turbulent inflow. The turbulence intensity, though, was still much smaller than for the other situations and hence again pictured in appendix C.

When looking at the upstream extent of the turbulence intensity structures, all of them reach the cavity leading edge. This can be explained from the turbulent inflow: since it already contains fluctuations, turbulence intensity is induced from the start.

From the laminar situation it is expected that if liquid is added to cavity flow with ill-defined vertical structures, the peak turbulence intensity and the high turbulence intensity region in the mixing layer will monotonically increase. This was proposed to be due to the smaller space to expand in. From the peak turbulence intensity behaviour and the high turbulence intensity regions, as shown in figure 5.22 and appendix C, such a clear trend was not found. Rather these turbulent situations showed variations over a small range without a clear trend.

Regarding the magnitude, it can again be concluded that the different Reynolds number flows had the same turbulence intensity. Though if one compares the values to the laminar situations, an
5.6. DISCUSSION

5.6.1. CAVITY FLOW

Concerning the question if the increased length to depth ratio of a cavity, by adding water, would affect the flow in and over the cavity, the answer is yes. Adding water to the cavity area makes the cavity less deep and some parallels can be drawn to empty cavities with the same length to depth ratio. Though changes arise as well, such as the different bottom contour: with increasing the Reynolds number of the air flow, the water surface modulation, with respect to the static distribution, becomes more severe. In effect this influences the internal cavity flow, giving differently sized and shaped recirculation cells. The recirculating flow again shows to affect the mixing layer. For laminar flows the internal recirculating flow controls the point at which the mixing layer shows linear growth (e.g. turbulent effects arise), the growth rate of the mixing layer and the nature of the vortical structures in the mixing layer. Whilst for turbulent flows only the growth rate of the mixing layer is affected by the internal flow.

For partially liquid filled cavities at a laminar, low Reynolds number inflow the mixing layer was suspected to be in the shear layer mode. In this case the liquid was always distributed uniformly on the bottom and the circular main recirculation cell decreased in size for higher filling degrees. The location where the transition to the linear growth rate regime took place depended on the size of the recirculation zone. Until filling the cavity halfway, an upstream located secondary recirculation cell appeared that led to an earlier transition. After passing this level, the secondary recirculation cell disappeared and the recirculation zone shrunk. This resulted in a postponed transition. The overall lower turbulence intensity is found for the turbulent situations. This is in agreement with the observations by Grace et al [18].

Figure 5.22: Turbulence intensity; top left: \( \text{Re}_{L+r} = 2.7 \times 10^4 \) at 0% filled; top right: \( \text{Re}_{L+r} = 2.7 \times 10^4 \) at 50% filled; bottom left: \( \text{Re}_{L+r} = 2.7 \times 10^4 \) at 75% filled and bottom right: Peak turbulence intensity behaviour for the turbulent situations.
spreading of the mixing layer was related to the size, the velocity and the height wise centre position of the recirculation zone. Until a 50% filled cavity the net effect of water addition was an increase in the growth rate and the spreading of the mixing layer, whereas a decrease occurred thereafter. The size of the vortical structures in the mixing layer (and thus the velocity fluctuations) are related to the upstream extent of the linear growth rate regime, therefore growing well-defined vortical structures and fluctuations were found until a 50% water level and decreasing structures and velocity fluctuations thereafter. This held until the cavity depth was smaller than the boundary layer height, since then no vortical structures occurred anymore. The magnitude of the pressure fluctuations is positively correlated to the magnitude of the velocity fluctuations. Therefore the noise level from the partially liquid filled cavity at this Reynolds number, is expected to initially increase and then decrease upon liquid filling.

For partially liquid filled cavities with laminar inflow at the higher two Reynolds numbers, the transition point to the linear growth regime was barely altered, as the main recirculation cell always filled the complete cavity. The mixing layer spreading in this case showed, under the influence of the recirculation zone, a monotonic decrease for higher water levels. Since the recirculation zone velocity decreased and the recirculation cell centre position increased in height. Another difference between these cases and the lower Reynolds number situation, were the ill-defined vortical structures in the mixing layer that origin from broken up well-defined vortical structures and are proposed to be more characteristic to the non-oscillating mode. As a configuration that initially showed well-defined vortical structures could present ill-defined vortical structures by increasing the Reynolds number or adding water (see figure 5.23). The breaking up is proposed to be caused by destabilizing influences in the flow. From measurements on the boundary layer of the highest Reynolds number flow, already a small turbulent component was seen. While in the case of water addition sufficiently strong fluctuations appear in the internal cavity area by the air-water interaction at the interface from a certain minimum Reynolds number. These influence the vortical structures such that they break up. The existence of these structures makes the periodic vortex-wall interaction at the trailing edge disappear and instead lead to a more random interaction. Likely reasons for this can be found in the largely varying size, distribution and impingement location of these structures, which differ more compared to well-defined vortical structures. Also a part of the structures may already be diverted into the cavity by the recirculating flow without colliding with the trailing edge. For this situation of ill-defined vortical structures, the velocity fluctuations at the trailing edge increase with water filling. This will lead to larger pressure fluctuations, that results in the emittance of higher levels of white noise.

For the turbulent cases the incoming flow already contains turbulent effects, that make the boundary layer break up into ill-defined vortical structures after separation from the leading edge. Since only ill-defined vortical structures are present, the mixing layer sometimes behaves just like the aforementioned laminar flow at higher Reynolds number. So does the mixing layer spreading also show an overall increase when filling the cavity with liquid. Though with regards to the turbulence intensity, changes are found. In contrast to the laminar situation, only variations and no clear trend in the turbulence intensity behaviour was found upon filling the cavity. As the variations were small, the noise level of the white noise is expected to stay constant. Regarding the recirculation zone in the cavity area, also a flow distribution similar to the laminar situation is found for the
higher two Reynolds numbers. But for the lower Reynolds number some changes appear in the internal cavity area. For filling degrees up to 50%, the recirculation cell will fill up the complete cavity area in contrast to the laminar situation. Whereas for a filling degree of 75%, the interaction of the mixing layer with the recirculation cell leads to a more transitional-like cavity. The 90% filled cavity again shows no recirculation cell.

5.6.2. Relation to Flexible Risers

For multiphase flow in flexible risers either annular flow or mist flow may be presented [9], depending on the flow rates of the natural gas and the liquid (oil, or any other liquid that may be injected). For high natural gas flow rates, the gas flow will shear liquid from the liquid film into droplets and create a mist flow. For lower natural gas flow rates and/or higher liquid flow rates annular flow is expected. Based on the above, three situation can be distinguished: annular flow where the liquid phase will completely fill up the corrugations, annular flow where the liquid phase will not completely fill up the corrugations and mist flow.

For the annular flow where the liquid phase will completely fill up the corrugations, no expectations can be drawn from this study. Since due to the large liquid film on the wall, the natural gas flow will no longer ‘see’ a corrugated pipe, but a smooth pipe. Consequently the whole idea of a cavity flow and the process of vortical structures that impinge on the trailing edge of the corrugations is no longer relevant. In this case therefore no tonal noise is expected. For the situation with annular flow, where the corrugations are not completely filled up, and mist flow, the flow of gas in the corrugations and the appearance of a liquid film in the corrugation area are still relevant. Therefore the remaining part of this subsection is devoted to these two situations and some expectations are given based on the present study.

As the flexible risers make tonal noise, the mixing layer is in the shear layer mode and has periodic vortex-wall interaction. This is thought to be best represented by the flow for the lower Reynolds number at all filling degrees and the middle Reynolds number in the empty cavity scenario. By increasing the Reynolds number and adding liquid to the cavity, the vortical structures are subjected to influences from the inflow and the recirculation cell, which may alter the periodic behaviour such that only ill-defined vortical structures arise. In this way the periodic vortex-wall interaction disappears, which should in principle lead to no tonal noise production, but white noise production. Though this behaviour cannot be directly extrapolated to the behaviour of flexible risers subjected to multiphase flow. Since the measurements were performed for a single cavity under laboratory conditions, which differ from the flexible riser in terms of the structure and the operating conditions. Therefore a translation step is made, to account for these changes and thereby give better expectations of the resulting flow and noise.

The laboratory experiments were performed at atmospheric pressure, whereas the flexible risers will transport multiphasic mixtures at pressures close to 100 bar. This leads to changed fluid properties between the performed measurements and the operation conditions. For the natural gas and the liquid the following trends are coupled to increased pressure. Firstly, regarding the density, the natural gas would show a large density increase [40], whereas for the liquids the density is, as a consequence of the incompressible nature, expected to stay approximately constant [37]. This promotes a more severe interaction between the natural gas and the liquid and may result into higher displacements of the liquid film. This higher interaction will also induce a larger tendency of the liquid to be blown out of the corrugation. Secondly, the viscosity of both the natural gas and the liquid (slightly) increases [21, 29]. As the viscosity ratio stays approximately constant, the attenuation of the waves at the interface is expected to be similar to the present measurements. Thirdly, the surface tension of the natural gas-liquid interface should show a lower magnitude at elevated pressure [52]. The inclusion of the surface tension is done now and not previously, since the corrugations of the flexible risers are about five to ten times smaller than in our cavity structure. This
will make the force induced by the surface tension larger and also leads to new effects such as the appearance of capillary waves instead of gravity waves.

The behaviour of the liquid film in the corrugations can in these two cases for the flexible risers that are locally oriented horizontal (only lower half of the pipe) and vertical, be assessed by the ratio between the dynamic pressure of the natural gas flow diverted into the corrugation and the surface tension of the liquid film $\gamma_l$ in the corrugation (left) and the ratio between the dynamic pressure of the natural gas flow diverted into the corrugation and the hydrostatic pressure theoretically needed to blow out the whole liquid film from the corrugation (right):

$$\frac{0.125 \rho_g U^2}{\gamma_l L + r} \quad \text{and} \quad \frac{0.125 \rho_g U^2}{\rho_l g D}$$

(5.1)

From an order of magnitude estimation it is found that the first ratio at an elevated pressure of around 100 bar will be larger than one, meaning that inertial forces are dominant at the interface. Since the dynamic pressure increase outweighs the surface tension increase due to the much larger gas density change [40]. This is partially in agreement with our study, since inertial forces were also dominant at the interface. The difference is that now the second ratio is smaller than one, whereas it was larger than one for our study.

For the lower half of the horizontally oriented riser sections, this means that most of the liquid is blown out of the corrugations by the inertial forces. In practice this prohibits high filling degrees in the corrugations. Though liquid accumulation in the corrugations is still expected to sustain in this case, but only in small amounts at the low velocity zone near the bottom leading edge corner. For the vertically placed risers sections the second ratio is not important due to the orientation. However the gravity, which is expected to be leading in this case, is foreseen to induce a similar distribution of the liquid film in the corrugations as described above. Even though liquid will escape from the corrugations in both cases, the behaviour inside the corrugation can be best compared to our measured internal cavity flow at higher Reynolds number and low filling degrees. Since here the liquid at the trailing edge was also displaced upstream, albeit to a much lesser extent.

For very small amounts of liquid accumulation in the corrugations and/or low natural gas flow rates, the velocity fluctuations created in the natural gas stream by interaction at the natural gas-liquid interface are not anticipated to be large. Therefore it is thought that the vortical structures in the mixing layer remain well-defined. But an increased noise production of the individual corrugations with well-defined vortical structures upon adding liquid, as was seen for the partially liquid filled cavities is not expected. Since the natural gas flow is highly turbulent and has high inertia, the recirculation zone is expected to fill the whole corrugation area. As a result, the mixing layer should always start close to the leading edge of the corrugation, independent of the water filling. Even larger well-defined vortical structures in the corrugation upon adding liquid, and thus higher noise levels of the individual corrugations, are hence not deemed possible. In practice, the situation of a too low liquid accumulation could be solved by injecting extra liquid in the flexible riser, whereas in the current time of increasing natural gas demands, the situation of a too low natural gas flow rate is not expected too often in practice.

From a certain minimum liquid accumulation in the corrugations and a certain natural gas flow rate, significant velocity fluctuations are created in the natural gas stream by interaction at the interface (waves). These fluctuations are foreseen to influence the well-defined vortical structures such that they break up in ill-defined vortical structures. Besides the interaction at the interface, the transportation of the sheared droplets from the natural gas-liquid interface to the mixing layer is anticipated to be an additional effect that promotes the destabilization of well-defined vortical structures. The transition of the vortical structures from well-defined to ill-defined by adding liquid, is paired with a slight increase in the velocity fluctuations (<4%). As this increase is very small, no significant changes upon adding liquid are foreseen in the noise production of individual corrugations with ill-defined vortical structures.
For the upper half of the horizontal oriented riser sections no liquid accumulation in the corrugations is expected to take place. Since in this gravity dominated situation, the much heavier liquid in the corrugation area will be pulled downwards. Also in the case of mist flow, the natural gas stream in the top half will have a much lower liquid content. As these factors make the flow in the corrugations better approach a single-phase gas flow, well-defined vortical structures are still expected here.

Another aspect is that we have simplified the flexible risers by only using a single two dimensional cavity. In reality a flexible riser is a concatenation of corrugations that are revolved around an axis. The incoming flow over the sequenced corrugations can then, under the influence of a sound wave, become synchronized and show periodic shedding with the same frequency in all the corrugations. It is, however, thought that as ill-defined vortical structures appear, the lock-in mechanism will have no significant effect. Since the distribution and the trailing edge impingement of such vortices has a wide spectrum, it cannot be regulated by a single frequency and no lock-in is expected to occur.

All in all, it can be derived that for multiphase flow in flexible risers from a certain gas flow rate and liquid flow rate, ill-defined vortical structures should appear on a large scale. These structures do not have a periodic interaction with the trailing edge and therefore, in the most convenient case, lead to the total disappearance of tonal noise. In return a white noise component is obtained. Though in practice it is not expected that tonal noise will be completely replaced by white noise. Since it is anticipated that in the mixing layer over some corrugated sections still well-defined vortical structures are formed, for example the upper half of a horizontal oriented flexible riser.

The white noise level, that is produced by the individual corrugations and partially replaces the tonal noise production, is similar in magnitude to the tonal noise level. In that sense there is thus no clear reduction in noise produced by the individual corrugations. However, as the ill-defined vortical structures prevent the lock-in mechanism to set in, no excessive levels of noise and mechanical vibrations can occur in the flexible riser.
CONCLUSIONS AND RECOMMENDATIONS

In the present study the cavity flow appearing upon filling the internal cavity area with water was studied. The report started with condensing the flexible riser problem to a more approachable cavity flow study. Thereafter a theoretical background was given that updated the reader on shear layer characteristics, which were important later on. The third chapter treated the cavity flow, such that the important features of the flow were covered. In the fourth chapter the experimental facility and the cavity model were discussed. Further on, the experimental techniques: Hotwire Anemometry and Particle Image Velocimetry were introduced and some light was shed on the procedures performed for both techniques before and after the measurements. Finally in the fifth chapter the results were treated, first for the base case of an empty cavity at laminar conditions and later on for a partially liquid filled cavity at laminar and turbulent conditions.

6.1. CONCLUSIONS

Looking back at the research objective of the present study, which was: ‘Investigate the effect of filling up the cavity volume with liquid in terms of the flow behaviour and the noise production potential’, the following conclusions can be drawn:

• Liquid addition to cavities will create a modulated ‘bottom’ contour for the gas flow that influences the internal cavity flow. For the low Reynolds number of \( \text{Re}_{L+r} = 2.7 \times 10^4 \), the water surface is mainly flat due to the low inertial force of the air flow. For the higher two Reynolds numbers at \( \text{Re}_{L+r} = 4.0 \times 10^4 \) and \( \text{Re}_{L+r} = 5.3 \times 10^4 \), the water at the trailing edge was displaced upstream. This is the result of the higher inertial force of the air flow.

• Overall the shape of the water surface showed the same trends as found by Belfroid et al. [6], only a trailing edge completely free from liquid was not seen owing to the too low inertial force of the air stream. For the flexible risers such a configuration is under certain circumstances, however, foreseen, due to the far greater inertial force of the gas stream. As a result, most of the liquid will be blown out of the cavity, but small amounts of liquid are expected to accumulate at the bottom leading edge corner of the cavity.

• The cavity flow of empty and partially liquid filled cavities for equal length to depth ratios contains the same components, but are not similar in distribution due to a difference in nature of the bottom edge (solid versus fluid). During inflow the bottom edge will interact with the flow or take in a different shape, which results into a changed internal flow, that subsequently also influences the mixing layer spanning the cavity.

• For all the cavity flow configurations an open cavity flow appeared, except for three cases: for the turbulent, low Reynolds number flow at a 75% filling degree the flow showed a more
transitional-like cavity flow, due to more interaction between the mixing layer and the internal cavity flow; for both the laminar and turbulent low Reynolds number flow at 90% filling no distinct internal cavity flow appeared, as the cavity depth was smaller than the boundary layer thickness.

- For the laminar low Reynolds number flow the recirculation zone contains a main recirculation cell near the trailing edge, that upon increasing the water level shrinks and eventually vanishes at 90% filling. This is probably caused by the low inertial force of the flow that is smaller than the adverse pressure gradient that prevails in the cavity area. For the higher Reynolds number flows and the turbulent flows, the main recirculation cell almost always reached the cavity leading edge and filled the complete cavity area. This is because the flow was more energetic, either by a higher inertial force or a turbulent nature, which made it possible to resist the adverse pressure gradient longer. The only exception was the middle Reynolds number at 75% filling.

- For laminar inflow the transition location of the mixing layer to the linear growth rate regime is affected by the size of the recirculation zone: the further upstream it reaches, the earlier the transition occurs by the transport of disturbances to the mixing layer. Though as the recirculation cell usually fills up the complete cavity, this effect will only be important at low Reynolds number flow.

- The linear growth rate of the mixing layer, which varied between $0.19 < \frac{d\delta}{dx} < 0.31$ in the present study, and the peak vorticity thickness, in general, decreased at higher filling degrees. This is because the mixing layer spreading showed dependency on the recirculation zone. For high water levels, lower velocities in the main recirculation cell appeared that were proposed to create a higher core pressure. This resulted into a smaller downward attraction and is therefore unfavourable for a larger growth rate and peak vorticity thickness. Next to this, the higher located recirculation cell, that appeared for higher water filling, hinders expansion of the mixing layer by its presence. Besides these two contributions, the nature of the vortical structures in the mixing layer also promoted the peak vorticity thickness. As ill-defined vortical structures show a wider spreading, a higher peak vorticity thickness was found in this situation than for well-defined vortical structures.

- Smaller well-defined vortical structures lead to a lower turbulence intensity in the mixing layer and at the trailing edge. This smaller size is coupled to a lower tonal noise level. Since the size of the well-defined vortical structures initially increased and then decreased upon water filling, the turbulence intensity and the tonal noise level showed the same trend. Ill-defined vortical structures for laminar inflow inflow, showed an increase in the turbulence intensity at the trailing edge for water addition. This leads to higher white noise levels upon increasing the water filling. This trend was thought to be the result of the smaller space of the vortical structures to expand in. For the turbulent inflow the turbulence intensity behaviour showed small variations without a clear trend. The noise level therefore seemed to be approximately constant with varying the filling degree.

- It was derived that the vortical structures in the mixing layer could be shifted to ill-defined by increasing the Reynolds number (from $Re_{L+r} = 4.0 \times 10^4$ to $Re_{L+r} = 5.3 \times 10^4$) or filling up the cavity with water (for $Re_{L+r} = 4.0 \times 10^4$ from 0% filling to 25% filling). It is proposed that this leads to destabilizing influences originating from either the incoming flow or the underlying recirculation zone. This is detrimental for the well-defined vortical structures, as they break up due to this. From a specific point, the multiphase flow in flexible risers will have a high
enough gas flow rate and sufficient liquid accumulation in the corrugation area. As a result, significant destabilizing influences are expected, that will lead to the appearance of ill-defined vortical structures on a large scale.

- As under certain conditions ill-defined vortical structure are expected in most of the mixing layers spanning the corrugations of the flexible riser, a lower tonal noise level should be produced by all the individual corrugations together. In return, the individual corrugations will, however, produce a larger white noise level. This keeps the total noise level produced by all the individual corrugations in both situations approximately the same. More important, the randomness of the ill-defined vortical structures is thought to be advantageous against the lock-in mechanism that appears in flexible risers, as no periodic component is present which can be regulated. This significantly lowers the maximum achievable pressure fluctuations in the flexible riser and thereby prevents excessive levels of noise and mechanical vibrations in the flexible risers.

- For partially liquid filled cavities at filling degrees higher than 90%, the mixing layer does not roll-up and form vortical structures. Instead it presents itself as a continuous streak from the leading edge to the trailing edge. From this situation small non-periodic pressure fluctuations are expected.

6.2. Recommendations

The present study gave more insight into multiphase flow in flexible risers by investigating flow over a partially liquid filled cavity. Though since this is the first time that experiments were performed on such a configuration and as this is a very fundamental representation of the multiphase flow in flexible risers, shortcomings still exist and create additional open questions. For future studies therefore some propositions are made by the author.

- As there is a possibility that the addition of liquid to a cavity may reduce the noise, a logical continuation may be to perform an investigation where the flow of both the gas and liquid phase and the noise level are measured simultaneously. Due to lack of time, efforts were not made to perform measurements on the liquid side and because of the absence of a horizontal open jet anechoic wind tunnel no sound measurements could be made. But in this way the interaction between the multiphase flow and the noise (level) can be better understood. Furthermore, more understanding can be gained in the transition of the vortical structures from well-defined to ill-defined. As it was only proposed to be related to destabilizing influences from the inflow and the recirculation area and at the moment no solid evidence is found for this. As a cavity model is already available, this research would be most suited to an experimental study. According to the author, the preferred method would be a measurement technique with a high temporal resolution such as time-resolved PIV.

- Since a flexible riser is a concatenation of cavities whose shedding period is locked-in, an interesting development would be to look at a tandem configuration of partially liquid filled cavities. In a further stage this could be extended to more cavities or even a whole pipe. In this way, it could be determined how the interaction would vary depending on the liquid filling and whether the distance between the corrugations has any effect on this. The nature of this study is more suited to numerical research. From experience the author suggests a method involving no turbulence modelling of the flow.

- Simplifying measures were also applied by using a larger shaped cavity structure than found in flexible risers, together with air and water as working fluids. A first act would hence be a smaller cavity structure. In this way the surface tension forces become larger and therefore the
dynamics of the interface better represents the dynamics of the interface in the corrugations of the flexible riser. Although it was shown in the previous chapter that the surface tension force would still be smaller than the inertial force of the gas stream in this case, unforeseen effects may still appear in the smaller cavity structure. A second act would be to do a parameter study on the effect of fluid properties and vary the density, the viscosity and maybe even the surface tension of the working fluids. This may help to get a better indication of the dynamics of the natural gas-liquid interface, that will appear in the flexible risers. A start to this was already made by performing PIV measurements in the current setup, using glycerine for the liquid phase. However, because of insufficient time it was chosen not to evaluate this data. The author thinks that for this study both experimental and numerical work are suitable.

- Finally the author would like to mention that, although partially liquid filled cavities are an important aspect of the mystery surrounding noise mitigation in flexible risers, one should not solely seek in this area for answers. Returning back to the introduction, Belfroid et al. [6] proposed three mechanisms that could possibly explain the noise mitigation. As of now, two of the three mechanisms were tested in the literature. Thereby it only seems relevant to quantify the remaining mechanism which states that: ‘The thicker upstream boundary layer of the gas flow, resulting from liquid deposition on the wall, could possibly reduce the source strength of the corrugations’. This is worth investigating, because the present study showed that a thicker boundary layer will lead to an altered cavity flow that, at a certain moment, does not include shedding of vortical structures.
## Nomenclature

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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
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<tr>
<td>$c$</td>
<td>Polynomial constant</td>
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<tr>
<td>$c_{\infty}$</td>
<td>Speed of sound</td>
<td>[m/s]</td>
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<tr>
<td>$D$</td>
<td>Cavity depth</td>
<td>[m]</td>
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<tr>
<td>$D$</td>
<td>Total differential</td>
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<td>$D_{\text{plate}}$</td>
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<td>$d$</td>
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<td>$d_r$</td>
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### A. NOMENCLATURE

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B

FIGURES FROM REFERENCES


Figure B.1: Mean velocity magnitude, left: $Re_L = 2.4 \times 10^4$ and right: $Re_L = 3.0 \times 10^4$

Figure B.2: Streamlines at $Re_L = 3.0 \times 10^4$
Figure B.3: Instantaneous vorticity distribution, left: $Re_L = 2.4 \times 10^4$ and right: $Re_L = 3.0 \times 10^4$

Figure B.4: Turbulence intensity, left: $Re_L = 2.4 \times 10^4$ and right: $Re_L = 3.0 \times 10^4$
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B.3. PARKHI [49]

Figure B.6: X-component of velocity at $Re_L = 2.0 \times 10^4$

Figure B.7: Streamlines at $Re_L = 2.0 \times 10^4$

Figure B.8: Instantaneous vorticity at $Re_L = 2.0 \times 10^4$
C.1. Empty and partially liquid filled cavity flow for laminar boundary layers

Flow field

Figure C.1: Non-dimensional mean velocity magnitude and streamlines at $Re_{L+r} = 2.7 \times 10^4$; first row: 0% filled (left) and 25% filled (right), second row: 50% filled (left) and 75% filled (right) and third row: 90% filled
Figure C.2: Non-dimensional mean velocity magnitude and streamlines at $\text{Re}_{L+r} = 4.0 \times 10^4$; top left: 0% filled, top right: 25% filled, bottom left: 50% filled and bottom right: 75% filled
Figure C.3: Non-dimensional mean velocity magnitude and streamlines at $Re_{L+r} = 5.3 \times 10^4$; top left: 0% filled, top right: 25% filled, bottom left: 50% filled and bottom right: 75% filled
Vorticity

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Figure C.5: Non-dimensional mean vorticity at $\text{Re}_{L+r} = 4.0 \times 10^4$; first row: 0% filled (left) and 25% filled (right), second row: 50% filled (left) and 75% filled (right) and third row: non-dimensional mean mixing layer core vorticity
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VORTICITY THICKNESS

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Turbulence intensity

Figure C.8: Turbulence intensity at $Re_{L+r} = 2.7 \times 10^4$; first row: 0% filled (left) and 25% filled (right), second row: 50% filled (left) and 75% filled (right) and third row: 90% filled
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C.1. EMPTY AND PARTIALLY LIQUID FILLED CAVITY FLOW FOR LAMINAR BOUNDARY LAYERS

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C.2. EMPTY AND PARTIALLY LIQUID FILLED CAVITY FLOW FOR TURBULENT BOUNDARY LAYERS

FLOW FIELD

Figure C.11: Non-dimensional mean velocity magnitude and streamlines at $Re_{L+T} = 2.7 \times 10^4$; first row: 0% filled (left) and 25% filled (right), second row: 50% filled (left) and 75% filled (right) and third row: 90% filled
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Figure C.13: Non-dimensional mean velocity magnitude and streamlines at $\text{Re}_{L+r} = 5.3 \times 10^4$; top left: 0% filled, top right: 25% filled and bottom: 50% filled
VORTICITY

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Figure C.16: Non-dimensional mean vorticity at $\text{Re}_{L+r} = 5.3 \times 10^4$; top left: 0% filled, top right: 25% filled, bottom left: 50% filled and bottom right: non-dimensional mean mixing layer core vorticity.
Vorticity thickness

Figure C.17: Non-dimensional vorticity thickness; top left: $Re_{L+r} = 2.7 \times 10^4$, top right: $Re_{L+r} = 4.0 \times 10^4$, bottom left: $Re_{L+r} = 5.3 \times 10^4$ and bottom right: growth rate for various filling degrees at the three Reynolds numbers
Turbulence intensity

Figure C.18: Turbulence intensity at $Re_{L+r} = 2.7 \times 10^4$; first row: 0% filled (left) and 25% filled (right), second row: 50% filled (left) and 75% filled (right) and third row: 90% filled.
Figure C.19: Turbulence intensity at $Re_{L+T} = 4.0 \times 10^4$; top left: 0% filled, top right: 25% filled, bottom left: 50% filled and bottom right: 75% filled.
Figure C.20: Turbulence intensity at $Re_{L+r} = 5.3 \times 10^4$; top left: 0% filled, top right: 25% filled and bottom: 50% filled.


