

DRIVING MECHANISMS OF GAP FLOW  
BETWEEN TWO SIDE-BY-SIDE  
CIRCULAR CYLINDERS

Michael David Atkins

A dissertation submitted to the faculty of Engineering and Built Environment, University of the Witwatersrand, Johannesburg, in fulfilment of the requirements for the degree of Master of Science in Engineering.

Johannesburg, 2014

## **DECLARATION**

I declare that this dissertation is my own unaided work. It is being submitted to the degree of Master of Science to the University of the Witwatersrand, Johannesburg. It has not been submitted before any degree or examination to any other university.

.....  
(Signature of Candidate)

..... day of .....year.....

## ABSTRACT

We present the driving mechanism of gap flow between two stationary side-by-side circular cylinders immersed in the sub-critical Reynolds number regime and its variation with gap spacing, based on steady flow characteristics. A series of experiments and numerical simulations have been performed. Results reveal that gap flow, the flow passing between the cylinders, can be classified broadly into *pressure gradient* and *momentum driven* regime, depending on the spacing ratio ( $T/D$ ) where  $T$  is the transverse center-to-center spacing between the cylinders and  $D$  is the cylinder diameter. The pressure gradient driven regime occurs roughly  $T/D < 1.25$  where the mean velocity of the gap flow increases as the spacing ratio ( $T/D$ ) increases. The momentum driven regime follows with a monotonic decrease in the mean velocity as the spacing ratio increases when  $T/D > 1.25$ . Within the pressure gradient driven regime, the gap flow undergoes transitions resulting from distinct changes in the circumferential static pressure distribution, velocity profile, the transverse pressure gradient and mean velocity at the throat of the two cylinders.

## **ACKNOWLEDGEMENTS**

I acknowledge the valued assistance and mentorship of my dissertation supervisor Professor Tongbeum Kim. His guidance and direct style of supervision has ensured that I have been continually challenged during my time at Wits and I have thoroughly enjoyed the experience. I am also grateful for the financial support from Eskom TESP 2014 and the Postgraduate Merit Award from the University of the Witwatersrand, Johannesburg.

| <b>CONTENTS</b>                                                           | <b>Page</b>   |
|---------------------------------------------------------------------------|---------------|
| DECLARATION                                                               | 2             |
| ABSTRACT                                                                  | 3             |
| ACKNOWLEDGEMENTS                                                          | 4             |
| LIST OF FIGURES                                                           | 6             |
| LIST OF TABLES                                                            | 7             |
| LIST OF SYMBOLS                                                           | 8             |
| <br><b>1. INTRODUCTION</b>                                                | <br><b>10</b> |
| <br><b>2.2. EXPERIMENTAL DETAILS</b>                                      | <br><b>15</b> |
| 2.1. Test rig and instrumentation                                         | 15            |
| 2.2. Velocity field measurement using Particle Image<br>Velocimetry (PIV) | 17            |
| 2.3. Data reduction parameters and measurement<br>Uncertainties           | 19            |
| <br><b>3. NUMERICAL SIMULATION</b>                                        | <br><b>21</b> |
| <br><b>4. DISCUSSION OF RESULTS</b>                                       | <br><b>26</b> |
| 4.1. Overall classification                                               | 26            |
| 4.2. Poiseuille flow regime ( $1.005 \leq T/D < 1.011$ )                  | 35            |
| 4.3. Pressure distribution transition ( $1.005 < T/D \leq 1.011$ )        | 39            |
| 4.4. Velocity profile transition ( $1.025 < T/D \leq 1.05$ )              | 41            |
| 4.5. Transverse pressure gradient transition ( $1.1 < T/D < 1.25$ )       | 43            |
| 4.6. Momentum driven regime ( $T/D > 1.25$ )                              | 45            |
| <br><b>5. CONCLUSION</b>                                                  | <br><b>50</b> |
| <br><b>REFERENCES</b>                                                     | <br><b>52</b> |

## LIST OF FIGURES

| Figure                                                                                     | Page |
|--------------------------------------------------------------------------------------------|------|
| 1.1 Schematic representation of distinct flow regimes                                      | 11   |
| 2.1 Experimental setup                                                                     | 16   |
| 3.1 Computational domain                                                                   | 23   |
| 4.1 Instantaneous PIV streamlines                                                          | 27   |
| 4.2. Schematic illustrating the position of the stagnation points                          | 28   |
| 4.3. The variation of the stagnation azimuth angle                                         | 29   |
| 4.4. Variation of the area ratio                                                           | 29   |
| 4.5. Classification of flow regimes of the gap flow                                        | 34   |
| 4.6. Pressure coefficient ( $C_p$ ) contour in the gap region                              | 35   |
| 4.7. Circumferential pressure distribution and velocity profile                            | 38   |
| 4.8. Circumferential pressure distribution                                                 | 39   |
| 4.9. Steady velocity and non-uniform pressure coefficient profiles                         | 43   |
| 4.10. The variation of the area weighted average...<br>of the transverse pressure gradient | 45   |
| 4.11. Time averaged PIV flow image                                                         | 46   |
| 4.12. The circumferential static pressure distribution                                     | 47   |
| 4.13. Schematic of the streamlines in the stagnation affected region                       | 47   |
| 4.14. Highly non-uniform pressure coefficient profile...<br>and developed velocity profile | 49   |

## LIST OF TABLES

| Table                                          | Page |
|------------------------------------------------|------|
| 4.1 New classification of the gap flow regimes | 34   |

## LIST OF SYMBOLS

|                                                      |          |
|------------------------------------------------------|----------|
| Flow area ( $\text{m}^2$ )                           | $A$      |
| Diameter (m)                                         | $D$      |
| Coefficient of pressure                              | $C_p$    |
| Base pressure coefficient                            | $C_{pb}$ |
| Factor of safety                                     | $F_s$    |
| Gap spacing between cylinder surfaces (m)            | $G$      |
| Inflection point                                     | $I$      |
| Reynolds number                                      | $Re$     |
| Non dimensional shedding frequency/ Separation point | $S$      |
| Stagnation point                                     | $St$     |
| Transverse center-to-center spacing (m)              | $T$      |
| Turbulence intensity                                 | $Tu$     |
| Axial velocity (m/s)                                 | $U$      |
| Particle image diameter (pixels)                     | $d_\tau$ |
| Shedding frequency (Hz), Friction factor             | $f$      |
| Static pressure (Pa)                                 | $p$      |
| Dynamic pressure (Pa)                                | $q$      |
| Radial distance from cylinder center (m)             | $r$      |
| Axis coordinate variable (m)                         | $x$      |
| Axis coordinate variable (m)                         | $y$      |
| Non-dimensional wall distance                        | $y^+$    |

## Greek Symbols

|                                                           |           |
|-----------------------------------------------------------|-----------|
| Boundary layer thickness (m)                              | $\delta$  |
| Fluid density ( $\text{kg}/\text{m}^3$ )                  | $\rho$    |
| Dynamic viscosity ( $\text{kg}/(\text{m}\cdot\text{s})$ ) | $\mu$     |
| Kinematic viscosity ( $\text{m}^2/\text{s}$ )             | $\nu$     |
| Azimuth angle (deg)                                       | $\varphi$ |
| Wall shear stress ( $\text{N}/\text{m}^2$ )               | $\tau_w$  |

### **Subscripts**

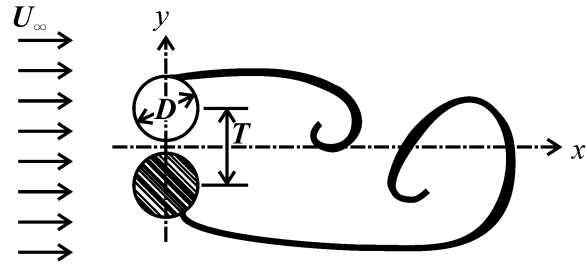
|                         |          |
|-------------------------|----------|
| Narrow wake             | $n$      |
| Wide Wake               | $w$      |
| Freestream              | $\infty$ |
| Stagnation/Entry region | $0$      |
| Throat                  | $T$      |

## 1. INTRODUCTION

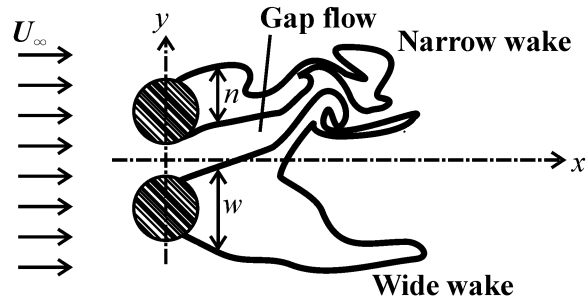
The behavior of flow around two stationary circular cylinders arranged in a side-by-side configuration strongly depends on the transverse center-to-center gap spacing ratio ( $T/D$ ) where  $D$  is the cylinder diameter (Fig. 1.1). Based on the observed wake structure [1-5], the flow pattern that varies with the gap spacing may be classified into three flow regimes: single bluff body (Fig. 1.1(a)), biased flow (Fig. 1.1(b)), and symmetric flow (Fig. 1.1(c)).

The single bluff body flow regime is defined when the gap spacing is *small* e.g.,  $T/D < 1.1 \sim 1.2$  [3-6]. The wake pattern is an asymmetric single vortex street with vortices shed alternately from the outer surfaces of the cylinders (Fig. 1.1(a)). The non-dimensional shedding frequency, i.e., Strouhal number  $S = (f(2D)/U_\infty)$  is approximately 0.2 where  $f$  is the shedding frequency and  $U_\infty$  is the free-stream velocity. In this regime, the characteristic length of  $2D$  agrees with the Strouhal number that is typically calculated for single bluff bodies with a diameter of  $2D$  [7].

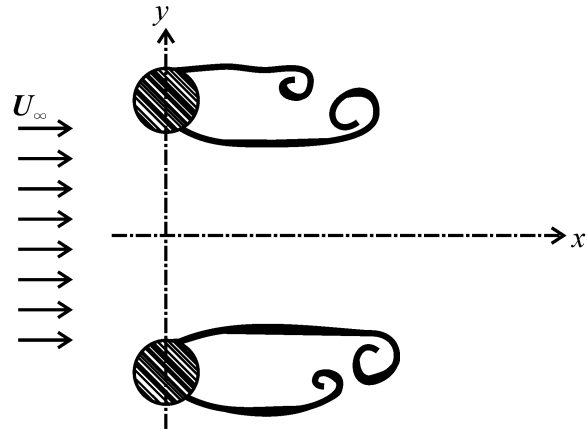
The biased flow regime is observed at an *intermediate* gap spacing ranging between e.g.,  $1.1 \sim 1.2 \leq T/D \leq 2.0 \sim 2.2$  [4,8,9]. The flow passing between the



(a)



(b)



(c)

**Figure 1.1. Schematic representation of distinct flow regimes at different gap spacing [3]: (a)  $T/D < 1.1 \sim 1.2$ , (b)  $1.1 \sim 1.2 \leq T/D \leq 2.0 \sim 2.2$  and (c)  $T/D > 2.2 \sim 2.0$ .**

cylinders - *gap flow* - is biased towards one cylinder (Fig. 1.1(b)). The wake pattern behind that cylinder has a narrow near-wake ( $n$ ) with a higher vortex shedding frequency  $S_n = (f_n D/U_\infty) \sim 0.3$  while the other cylinder has a wider near-wake ( $w$ ) and a lower shedding frequency  $S_w = (f_w D/U_\infty) \sim 0.1$  [3-5, 7-15]. The wake is bi-stable because the gap flow switches direction at random intervals between the two side-by-side cylinders [8-10,12,15,16]. However, the time intervals between the switching (or flopping) events are several orders of magnitude greater than the vortex shedding period and instability of the separating shear layers [12,16]. Sumner *et al.* [3,12,17] have argued that bi-stable flow is an example of “quasi-steady behavior” due to the relatively large time scale between switches.

The role of the gap flow is considered to behave like a jet flow [7]; higher momentum penetrating into the wake region through the gap. As a result, inner vortices begin to roll up in the wake region. Hence, the gap flow divides the single large scale wake and enables the formation of two vortex streets. Zhou *et al.* [16] has speculated that the generation of inner gap vortices by gap flow is necessary for the formation of two vortex streets. The angle of the gap flow deflection relative to the  $x$ -axis that is coincident with the nominal free-stream direction also depends on the spacing ratio. The deflection is at its greatest at the beginning of the biased regime and decreases monotonically to zero when the spacing ratio increases, approaching a symmetric wake structure as illustrated in Fig. 1.1(c) [3,15].

The symmetric flow regime occurs at large gap spacing e.g.,  $T/D > 2.2 \sim 2.0$ . The wake pattern is predominantly symmetric with respect to the centerline axis (the  $x$ -axis) as illustrated in Fig. 1.1(c). Two separate but coupled vortex streets are shed from each cylinder; each with the same shedding frequency  $S = (fD/U_\infty) \sim 0.21$ , which is equal to a single cylinder [1-11,13-21].

It has been argued in previous studies that gap flow is a possible mechanism that can alter the wake behavior; therefore, leading to the classification of these three flow regimes illustrated in Fig. 1.1. Sumner *et al.* [3,8,22] argued that for the small gap spacing (i.e., single bluff body flow regime); higher momentum gap flow bleeding between the cylinders alters vortex formation in the wake. Gap flow bleeding into the wake extends the vortex formation position further downstream of the cylinders. Here, the vortex formation region is the location in the wake where the periodic vortex street first develops [23].

For the biased flow regime, Spivack [7] and Bearman *et al.* [8] reported that the gap flow leads to the formation of gap vortices on the inner cylinder surface. Furthermore, Agrawal *et al.* [17] numerically observed that the gap flow interacts strongly with the wake behind the cylinders and, therefore, could be responsible for the bi-stable behavior of the wake.

For the large gap spacing, Wang *et al.* [5] inferred, based on their flow visualization results, that the interaction of the outer vortices with the shed gap vortices emanating from the gap flow is a possible cause of wake instability

resulting in the distinct wake pattern that has been studied in the symmetric flow regime.

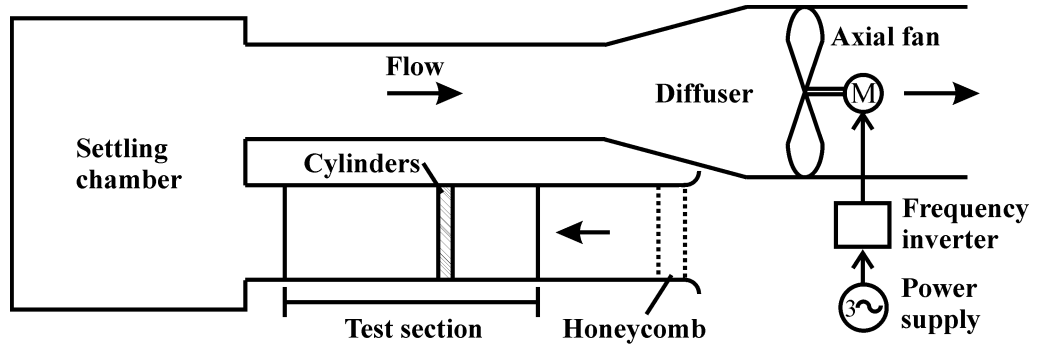
Despite the previous studies including the above-mentioned works that have identified gap flow behavior as a probable cause of the distinct types of wake behavior, the physical characteristics of gap flow and their variation with gap spacing have not yet been understood. This study squarely aims to demonstrate (a) driving mechanism of gap flow at each regime and (b) more detailed classification of flow regimes based on newly identified gap flow behavior and its driving force with varied gap spacing. To this end, a series of steady state experiments and numerical simulations has been conducted, that focuses on the time-averaged flow field formed by two side-by-side circular cylinders of different gap spacing. The fundamental relationship between static pressure in the flow-field and fluid acceleration in the gap is obtained. In contrast, previous studies classify the wake behavior by unsteady, instantaneous experimental and numerical methods. In this study, the new classification of the gap flow is made, based on steady flow behavior, providing further physical insight into the nature of gap flow at different gap spacing.

## 2. EXPERIMENTAL DETAILS

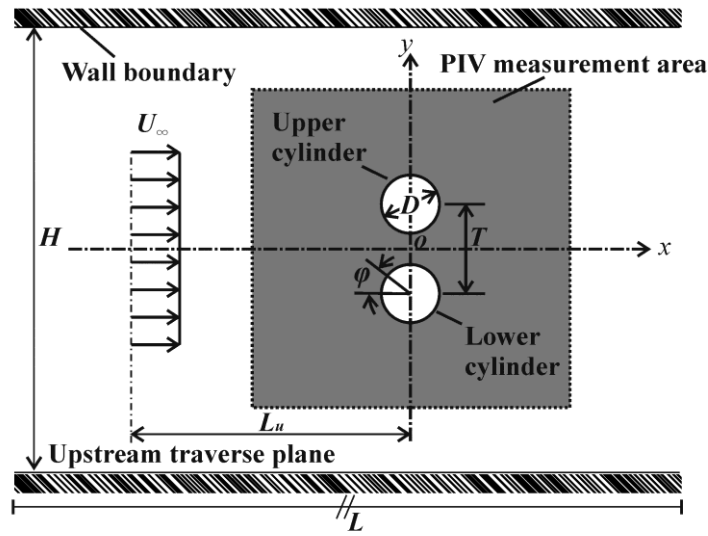
### 2.1. Test rig and instrumentation

Figure 2.1 shows a schematic of the present experiment setup. The experiments were conducted in an open suction type wind-tunnel with test section dimensions of  $W \times H \times L = 0.312\text{m} \times 0.32\text{m} \times 1.0\text{m}$ . A single layer of honeycomb is placed ahead of the test section to improve the uniformity of the free-stream flow. The flow velocity in the test section is controlled by a frequency inverter that adjusts the axial fan speed that drives the flow through the wind tunnel (Fig. 2.1 (a)). The flow velocity was set to maintain a subcritical Reynolds number based on the single cylinder diameter of  $Re_D = 6000$ . Prior to the actual testing the free stream turbulence intensity was measured to be  $Tu = 1.1\%$  (Dantec MiniCTA 54T42).

Two circular cylinders were mounted in the test section. Each cylinder was made from precision ground silver steel with a diameter of 15.0mm and the mean surface roughness was of  $< 6.0\mu\text{m}$ . The resulting area blockage was 9.4%. The lower cylinder had a single static pressure tapping that was electric discharge machined (EDM) into the mid-span of the cylinder with a circular hole diameter of 0.25mm. The lower cylinder was also connected to a rotary table that was mounted outside of the wind-tunnel. The cylinder was rotated to cover up to 360 degrees. The zero azimuth angle was set for each gap spacing so that the pressure



(a)



(b)

**Figure 2.1. Experimental setup; (a) Schematic of the open suction type wind tunnel used for the experiments (plan view); (b) Schematic of the test section with two stationary side-by-side circular cylinders where the lower cylinder is rotated to measure the circumferential pressure distribution (side view).**

hole would coincide with the geometric stagnation point for a single cylinder. The cylinder was rotated at a 5 degree increment whilst between the azimuth angles of 80 and 110 degrees, a 2 degree increment was used. Ten spacing ratios,  $T/D = 1.005, 1.011, 1.025, 1.1, 1.3, 1.7, 2.1, 3.0, 4.0$ , and  $6.0$  were selected. Among these

spacing ratios,  $T/D = 1.005, 1.011, 1.025$  were chosen based on preliminary numerical simulation results showing transition in the circumferential static pressure distribution. The other spacing ratios were based on the values considered in previous studies including [13].

To measure the free stream flow conditions, the velocity profile was traversed at a distance  $L_u = 15.0D$  upstream of the cylinders (see, Fig. 2.1(b)). The axial velocity was measured using a Pitot probe mounted on a linear traverse system (along the  $y$ -axis) that was positioned at the mid-span of the cylinders. The pressure data from the Pitot probe and static pressure tappings were read by a differential pressure transducer (DSA 3217, Scanivalve Inc).

## **2.2. Velocity field measurement using Particle Image Velocimetry (PIV)**

The upstream and downstream velocity field around the cylinders including the gap region was mapped using particle image velocimetry (PIV). This method allowed instantaneous and time averaged flow field measurements to be obtained within the measurement area shown in Fig. 2.1(b).

The flow field was seeded with an atomized mineral oil with a mean particle diameter of  $1\mu\text{m}$ . It is assumed that the trajectory of the seeding particles is the same as the actual flow field and the particles do not interact with one another [24]. The effect of particle “lag” on the measurement accuracy is considered to be

small due to the relatively low flow velocity [25]. The particles carried by the flow are then illuminated by a pulsed laser light sheet that has a 1.0mm thickness. The green light-sheet is generated from a frequency doubled, dual cavity Nd:YAG laser (New Wave Research), with a light wavelength of 532nm. The light sheet is perpendicular to the center axis of the cylinders and located at the mid span. It is further assumed that the measured flow field is predominately two dimensional, so that the majority of particles entering into the thin light sheet remain within the light sheet during the measurement period. The measurement period is defined by two sequential pulses of the laser light sheet that are separated by a finite time interval (33 $\mu$ s).

The recorded flow field is represented by the random pattern of particle images that are mapped onto the image plane of the CCD sensor. The images are recorded on the sensor frame that has a 2048 x 2048 pixel resolution and a pixel pitch of 7.4 $\mu$ m. The field of view that is setup within the light sheet is shown in Fig. 2.1(b), where the PIV measurement area is 175mm  $\times$  175mm. The optical axis of the camera is perpendicular to the laser light sheet.

Acquisition of the velocity field depends on first estimating the displacement field. This is achieved when the PIV images are interrogated using the Dantec (DynamicStudio V1.45) imaging systems software. The PIV images are divided into sub regions referred to as interrogation areas that are 32  $\times$  32 pixels in size and overlapped by 50%. Within each interrogation area the average velocity is estimated based on the assumption that the sub regions are small enough so that

the local velocity gradients within the interrogation areas are insignificant. Hence, each particle-image appearing within a sub region has uniform velocity. However, this assumption leads to inaccuracies in regions of the flow field that experience high levels of shear such as within the boundary layer [26,27].

The evaluation of the vector field between two successive frames yields an instantaneous vector map. The time averaged velocity field for a specific location in the flow field is then evaluated over an ensemble of eighty instantaneous vector maps sampled at a frequency of 5Hz.

### 2.3. Data reduction parameters and measurement uncertainties

The Reynolds number based on the single cylinder diameter  $D$  and the upstream mean flow velocity  $U_\infty$  is defined as:

$$\text{Re}_D = \frac{\rho_\infty U_\infty D}{\mu_\infty} \quad (1)$$

where  $\rho_\infty$  and  $\mu_\infty$  are the density and dynamic viscosity of air, respectively. The Reynolds number was fixed at 6000 (sub-critical flow regime) for all spacing ratios.

The circumferential static pressure  $p(\varphi)$  on the lower cylinder measured at an arbitrary azimuthal angle  $(\varphi)$  is expressed as a non-dimensional pressure coefficient defined as:

$$C_p = \frac{p(\varphi) - p_\infty}{\rho_\infty U_\infty^2 / 2} \quad (2)$$

where  $p_\infty$  is the free stream static pressure. The uncertainty associated with the azimuth angle was found to be within  $\pm 0.03^\circ$ . The experimental uncertainty of the Reynolds number, axial flow velocity and the static pressure coefficient were estimated using a method reported by Holman [28] based on 20:1 odds and to be within 1.1%, 1.1% and  $\pm 0.04$  respectively. Uncertainty of the cylinder dimensions and the transverse center-to-center cylinder spacing are within  $\pm 0.015$ mm. Pitot probe traverse positions along the y-axis are measured using a digital caliper with a resolution of 0.01mm.

The uncertainty of the PIV velocity measurements are primarily related to the estimation of the average particle displacement within an interrogation area. The uncertainty regarding the timing of the light sheet pulses, camera synchronization and particle lag are not considered as significant sources of error. The determination of measurement uncertainty relating to particle displacement have been quantified analytically and by the generation of synthetic images with known parameter values [24,26,29]. In order to provide a reasonable estimate of measurement error, the results given in Westerweel [29] are considered to be applicable to the PIV algorithm used in this investigation. The displacement measurement error based on the mean particle image diameter ( $d_r$ ) of 2-3 pixels is 0.05 pixels. Therefore, the full scale relative measurement error is 0.6%. The error of the time averaged velocity estimates are determined from the method given in Wernet [30] and are within 2.3%.

### 3. NUMERICAL SIMULATION

Numerical simulations were performed to gain physical insight into the behavior of certain flow parameters such as the gap velocity profile and gap static pressure profile, which were experimentally inaccessible. This is primarily due to the extremely close spacing of the cylinders that were considered. The experimental data of the circumferential pressure coefficient for  $T/D = 1.005, 1.011, 1.025, 1.1, 1.3, 2.1, 3.0, 4.0, \text{ and } 6.0$  and the mean velocity profile between the cylinders for  $T/D = 1.7$  and  $2.1$  was used to validate the numerical solution. To make direct comparison between experimental and numerical results, the setup of the numerical simulation was based extensively on the experimental configuration.

A commercial computational fluid dynamics (CFD) solver, Fluent version 14.5 was employed. The two-dimensional Reynolds Averaged Navier-Stokes (RANS) governing equations were solved by implementation of a finite volume discretization of the fluid domain [31]. A steady-state, pressure-based solver was employed to be consistent with the time-averaged experimental data. The discretized governing equations were solved using the Coupled algorithm. Spatial discretization of the transport equations yields diffusion terms that are second order accurate and convective terms evaluated at the cell face are interpolated from the cell centers using a second order upwind scheme. The required gradients for interpolation are computed by way of the Green-Gauss Node-Based scheme. The flow Courant number had a value of unity, which was necessary to ensure

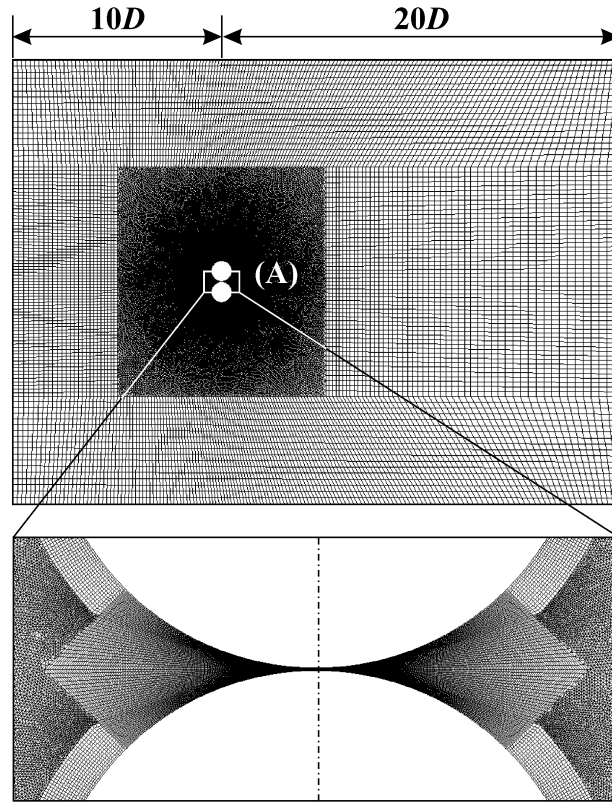
adequate stability of the numerical solution. The Explicit relaxation factors for momentum and pressure were both maintained at 0.75. The calculations were performed using double precision.

Amongst other turbulence models, the Spalart- Allmaras turbulence model was found to produce results in best agreement with the experimental data and thus was employed.

The height of the computational domain was made the same as the test section and extended  $10D$  and  $20D$  fore and aft of the cylinders respectively (Fig. 3.1). The domain was discretized using a Hybrid mesh strategy whereby the computational domain was split into an outer and inner block. The outer block used structured, quadrilateral elements. The inner block used predominantly unstructured triangular elements, the interface between the outer and inner blocks were conformal. Within the inner block is the gap region, which was split into multiple blocks. These blocks used refined quadrilateral elements to resolve the gap region with adequate resolution when the spacing was very close. The method of generating the mesh for the gap region was consistent for all the spacing ratios (i.e.,  $1.005 \leq T/D \leq 6$ ) that were simulated numerically.

The meshing of the boundary layer is arguably the most critical feature of the grid that can affect the accuracy of the solution. Consequently, an inflation layer was inserted onto the cylinder surfaces. The inflation layer consisted of 15 layers where the overall boundary layer thickness ( $\delta$ ) was determined from  $\delta/R(U_\infty$

$D/\nu)^{0.5} = 3.6F_s$ ;  $R = D/2$ ,  $F_s = 2^{1/2}$  and  $\nu$  is the kinematic viscosity [32]. This estimation was sufficient to ensure that the grid resolution was high enough to capture the development of the separated flow regions. Within the inflation layer, the viscous sub-layer needed to be resolved, which has a dimensionless wall distance  $y^+ < 5$  where  $y^+ = \rho_\infty(\tau_w/\rho_\infty)^{0.5}(y/\mu_\infty)$  and  $\tau_w$  is the shear stress at the wall [31,32]. The first layer thickness is calculated based on the criteria that  $y^+ \sim 1$  [31,33]; the simulation setup used  $y^+ = 0.75$ , which resulted in a first layer thickness 0.33% of the cylinder diameter. The maximum cell aspect ratio is 1.3.



**Figure 3.1. Computational domain for  $T/D = 1.005$  where sub-domain (A) shows the meshing of the gap region.**

The far field inlet boundary was setup as the *velocity inlet* condition and assumed

the incoming flow to be uniform. The inlet turbulence intensity is  $Tu = 1.1\%$ , which was experimentally measured. The outlet boundary was set to the *pressure outlet* condition. All wall surfaces including the boundaries representing the wind tunnel's upper and lower walls had the *no slip* condition.

There were two iterative convergence criteria that were applied globally to all simulated spacing ratios: (a) The normalized residuals relating to the momentum and continuity equation must decrease by at least 4 orders of magnitude [34,35]. (b) The iteration error must be at least 2 orders less than the numerical error. This was evaluated over the last 3000 iterations of each simulation using the centerline axial velocity as the critical parameter at distances  $0D$ ,  $3D$  and  $10D$  downstream of the cylinders. The iterative error is within 0.04%.

The Least Squares version of the Grid Convergence Index (LSGCI) method was used for estimating the discretization error for all of the simulations [34,36]. The LSGCI method was used as the conventional GCI method based on Richardson extrapolation is only valid in the case where the numerical solutions on three grids of different levels of refinement show monotonic convergence. It is acknowledged in the literature that the numerical solution may have non-monotonic convergence, thus the GCI method is not suitable [34-37]. The LSGCI procedure required that four grids were produced of varying amounts of refinement for each spacing ratio [36]. For the above grids the grid refinement factor was greater than 1.3. The critical parameters were the axial velocity components of the velocity wake

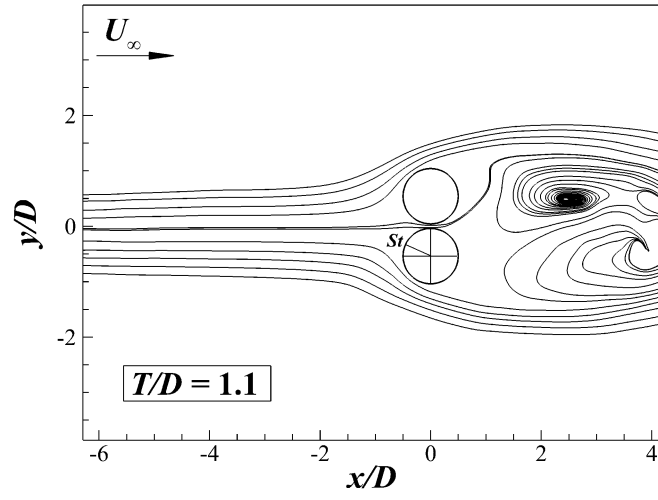
profile taken at 10 diameters downstream of the cylinders and the flow separation angle on the outer cylinder surface. The numerical uncertainty is within 2.5%.

## 4. DISCUSSION OF RESULTS

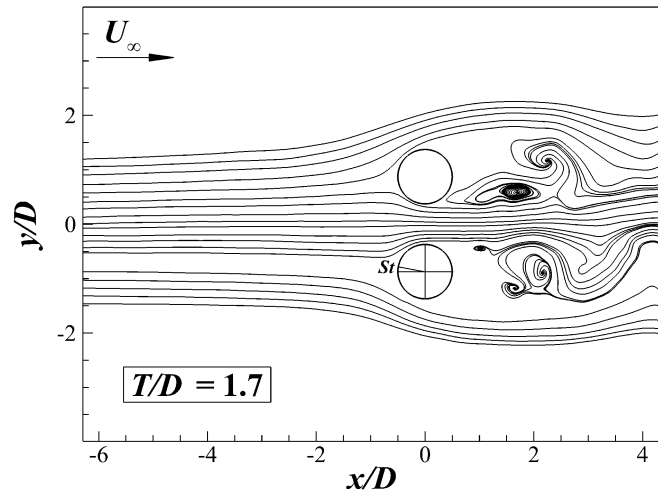
### 4.1. Overall classification

The overall flow behavior around two side-by-side circular cylinders including the gap region is shown in Figs. 4.1 (a,b), obtained experimentally for  $T/D = 1.1$  and 1.7. From these PIV flow images it is observed that the free stream ahead of the cylinders is divided either side of the stagnation point ( $St$ ). The stagnation point is also the termination point of so-called *dividing* or *stagnation* streamlines that originate far upstream of the cylinders and divide the free stream into *external* and *internal* flow. The external flow is directed around the outer surface of the cylinders, whereas the internal flow passes between the cylinders, hence gap flow is also internal flow.

The stagnation streamlines are illustrated schematically in Fig. 4.2 and are perpendicular to the cylinder wall surface. These dividing streamlines are analogous to the outer edges of a stream tube where the mass flow rate is conserved throughout a stream tube. From the continuity equation, the flow area ( $A_\infty$ ) between the two stagnation streamlines in the free-stream flow is related to the mean axial velocity ( $U_{T,ave}$ ) measured at the *throat* ( $A_T$ ); where the throat is the smallest flow area between the cylinders. If the flow area between the stagnation streamlines ( $A_\infty$ ) is known then the mean velocity at the throat can be estimated as the free-stream velocity ( $U_\infty$ ) and the gap spacing at the throat are generally known parameters.

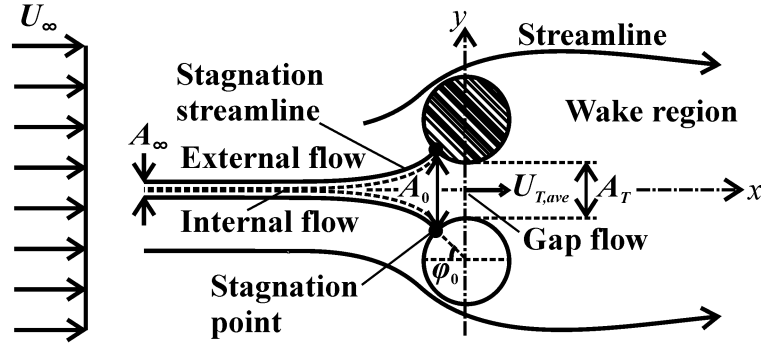


(a)



(b)

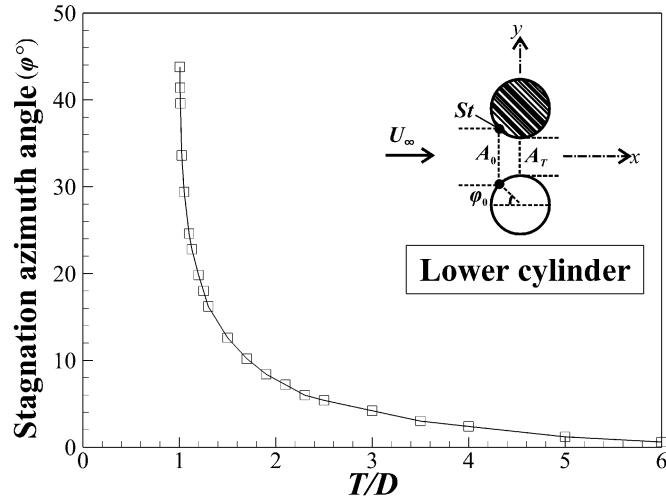
**Figure 4.1** Instantaneous PIV streamlines of the flow field around two side-by-side circular cylinders. (a) Stagnation azimuth angle  $\varphi_0 \approx 24.6^\circ$  for  $T/D = 1.1$  (b) Stagnation azimuth angle  $\varphi_0 \approx 10.2^\circ$  for  $T/D = 1.7$ .



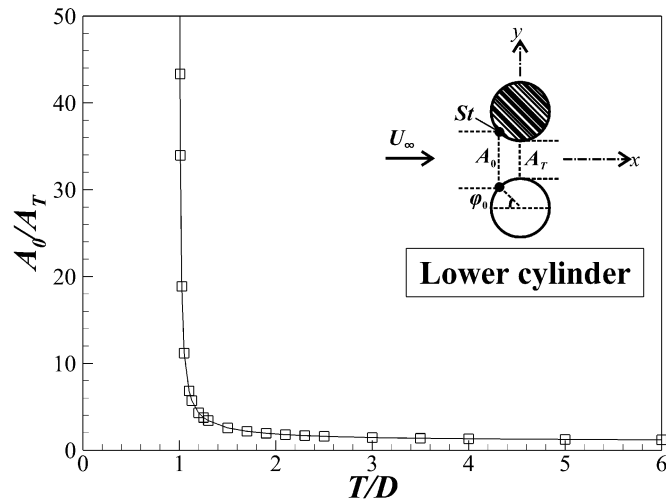
**Figure 4.2. Schematic illustrating the position of the stagnation points that separate the flow into external and internal flow. The location of the free-stream flow area between the stagnation streamlines ( $A_\infty$ ), the entry ( $A_0$ ) and the throat region ( $A_T$ ) of the gap flow is also shown.**

Figure 4.3 shows the variation of the azimuth angle ( $\phi_0$ ) of the stagnation point on the lower cylinder that decreases monotonically as the spacing ratio increases. Here, the stagnation point is the location on the cylinder surface that has the maximum static pressure. When the spacing ratio is large, the stagnation point approaches the azimuth angle of the geometric stagnation point of a single cylinder (i.e.,  $\phi_0 \rightarrow 0^\circ$ ). However, when the cylinders are close, such as  $T/D = 1.005$ , the stagnation point azimuth angle becomes  $\phi_0 \approx 43.8^\circ$ .

The stagnation point also defines the *entry region* ( $A_0$ ) of the gap flow immediately between the cylinders (Fig. 4.2). An area ratio is defined between the entry region and the throat ( $A_0/A_T$ ). The variation of ( $A_0/A_T$ ) with the spacing ratio is shown in Fig. 4.4. The entry region is dependent on the location of the stagnation points (Fig. 4.3). As the gap spacing increases, the azimuth angle of the



**Figure 4.3. The variation of the stagnation azimuth angle ( $\phi_0$ ) with spacing ratio.**



**Figure 4.4. Variation of the area ratio ( $A_0/A_T$ ) between the entry region ( $A_0$ ) and the throat ( $A_T$ ) with spacing ratio ( $T/D$ ).**

stagnation point decreases, causing the area ratio, ( $A_0 / A_T$ ) to decrease monotonically.

Flow images from the PIV measurements (Fig. 4.1) also show that as the internal flow approaches the cylinders, the internal streamlines *disperse* ahead of the entry

region. When these streamlines disperse, the local flow velocity reduces and static pressure increases. This creates a *stagnation affected region* ahead of the gap, which is a local area of the flow field with an adverse pressure gradient, where the static pressure increases in the direction of flow.

The degree of dispersion of the internal streamlines can be quantified according to the ratio of the flow area ( $A_\infty$ ) between the two stagnation streamlines in the free-stream flow and the entry region ( $A_0$ );  $A_\infty/A_0$ . If  $A_\infty/A_0 \ll 1$  then there is a high level of dispersion of the internal streamlines. Conversely, when  $A_\infty/A_0 = 1$  then the streamlines of the internal flow has no dispersion.

Generally, when the internal streamlines are highly dispersed the flow ahead of the cylinders decelerates and static pressure increases dramatically. The implication of the dispersed internal streamlines is a large proportion of the free-stream fluid momentum is converted into localized high static pressure ahead of the cylinders.

The internal streamlines have the highest levels of dispersion i.e.,  $(A_\infty/A_0) \ll 1$  when the gap spacing is small. In contrast, when the cylinders have a large spacing the dispersion of the internal streamlines is negligible i.e.,  $A_\infty/A_0 = 1$ .

The degree of dispersion of the internal streamlines is related to the level of curvature of the stagnation streamline that bounds the internal streamlines. We can surmise that the curvature of the stagnation streamlines is governed by; (1) the

continuity equation, where uniform flow entering between the stagnation streamlines at ( $A_\infty$ ) must leave between the cylinders ( $A_T$ ), (2) The stagnation streamline terminates normal to the cylinder wall, and (3) The mass flow rate between the stagnation streamlines and the cylinders is controlled by the level of resistance of the flow at the throat. As the flow passes through the entry and throat region, the internal streamlines that were dispersed ahead of the cylinders now begin to bunch together, indicating the flow velocity increases in the gap region.

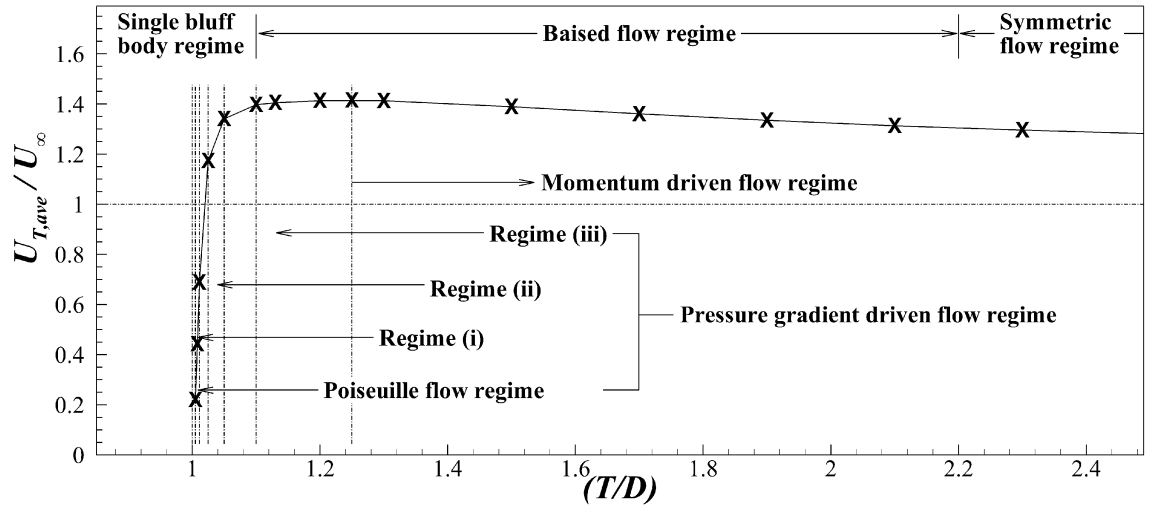
The gap flow is accelerated through the gap region and introduces higher fluid momentum into the wake region. The gap flow interacts strongly with the fluid in the wake, resulting in distinct wake behavior depending on the spacing ratio. The two  $T/D$  cases shown in Figs. 4.1(a,b) have been previously classified into the biased flow regime (Fig. 1.1(b)). Despite both cases having biased and bi-stable gap flow behavior, the resulting wake patterns differs significantly. For  $T/D = 1.1$ , the wake pattern is comparable to that of a single bluff body wake structure. Generally, the gap flow is highly deflected towards the upper cylinder creating a narrow near wake. Behind the lower cylinder a wide near wake is observed, which appears to dominate the wake structure. Considering the wake pattern of the larger spacing ratio  $T/D = 1.7$ , the flow image highlights the case where the gap flow and wake pattern were momentarily symmetrical about the  $x$ -axis. The wake structure has two separate but highly coupled vortex streets behind each cylinder. Following the symmetrical wake pattern, the gap flow then became deflected toward either cylinder but with a lesser degree of gap flow deflection compared to  $T/D = 1.1$ .

As it has been previously discussed, the mean velocity at the throat is related to the free-stream flow area ( $A_\infty$ ) between the stagnation streamlines, by the continuity equation. Key to understanding the driving mechanism of the gap flow, is to determine how the mean axial velocity ( $U_{T,ave}$ ) at the throat varies with spacing ratio. The mean axial velocity at the throat is determined by numerically integrating the velocity profile as:

$$U_{T,ave} = \frac{1}{G_T} \int_{G_T} U_T(y) dy \quad (3)$$

where  $U_T(y)$  is the axial velocity component across the throat (along the  $y$ -axis in Fig. 4.2) and  $G_T$  is the gap spacing at the throat. The variation of the mean velocity with the spacing ratio is plotted in Fig. 4.5 that is based on the local velocity field obtained from the present numerical simulations. This leads to a new classification of the gap flow, based on steady flow behavior. The classification of the regimes is based upon the dominant mechanism that is driving the gap flow at the throat, which will be discussed in the sections to follow. It should be noted that there is an approximate correlation between these newly defined steady gap flow regimes and the previously defined unsteady regimes based on the wake pattern (Fig. 1.1). However, in this study, the behavior of the gap flow has also been evaluated for a very small range of spacing ratios, which has not been previously studied. Hence, a new physical interpretation of the existing single bluff body regime is provided. Gap flow has been classified broadly into either *pressure gradient* or *momentum driven* regime. Within the pressure gradient driven regime (roughly  $T/D < 1.25$ ), the mean velocity ( $U_{T,ave}$ ) at

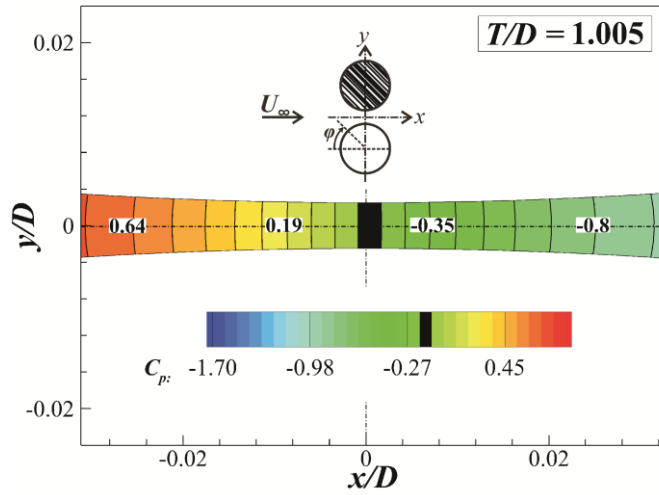
the throat increases as the spacing ratio ( $T/D$ ) increases (Fig. 4.5). Conversely, at  $T/D > 1.25$ , the mean velocity decreases, following the same trend as the variation of the area ratio ( $A_0/A_T$ ) with gap spacing shown in Fig. 4.4 - the momentum driven regime. Within the pressure gradient driven regime, the gap flow has newly identified *intermediate* regimes that are associated with transitions that occur at certain gap spacing. These transitions are distinguished by changes in the circumferential static pressure distribution, velocity profile, transverse pressure gradient and mean velocity. At  $1.005 \leq T/D < 1.011$ , the gap flow is driven purely by a favorable axial pressure gradient, the so-called “Poiseuille flow regime.” After which, the gap flow makes a transition that affects the pressure distribution in the vicinity of the throat. The second transition is observed from a distinct change in the velocity profile within the range of spacing ratio  $1.025 < T/D \leq 1.05$ . The change of the velocity profile coincides approximately with the biased flow regime. The third transition occurs between  $1.1 < T/D < 1.25$ ; the magnitude of the transverse (or radial) pressure gradient across the throat begins to decrease. The final transition is seen at  $T/D > 1.25$ ; as the mean velocity decreases; the gap flow transitions from predominantly pressure gradient to the momentum driven regime. These classified sub-regimes are summarized in Table 4.1 and are discussed in detail, separately.



**Figure 4.5. Classification of flow regimes of the gap flow from the variation of the normalized mean axial velocity at the throat ( $U_{T,ave}/U_{\infty}$ ) with spacing ratio ( $T/D$ ).**

**Table 4.1. New classification of the gap flow regimes.**

| Spacing ratio ( $T/D$ )  | Driving force     | Description of transition                                        |
|--------------------------|-------------------|------------------------------------------------------------------|
| $1.005 \leq T/D < 1.011$ | Pressure gradient | Poiseuille flow                                                  |
| $1.005 < T/D \leq 1.011$ | Pressure gradient | Pressure distribution transition                                 |
| $1.025 < T/D \leq 1.05$  | Pressure gradient | Velocity profile transition                                      |
| $1.1 < T/D < 1.25$       | Pressure gradient | Radial pressure gradient transition                              |
| $T/D > 1.25$             | Momentum          | Mean velocity $U_{T,ave}$ and axial pressure gradient transition |



**Figure 4.6. Pressure coefficient ( $C_p$ ) contour in the gap region at the throat showing parallel isobars normal to the flow direction with no transverse pressure gradient.**

#### **4.2. Poiseuille flow regime ( $1.005 \leq T/D < 1.011$ )**

The Poiseuille flow regime is defined when the gap flow is driven by a favorable axial pressure gradient where the static pressure in the flow field decreases in the flow direction. This regime occurs when the gap spacing is very small i.e.,  $1.005 \leq T/D < 1.011$ . Figure 4.6 shows the contour plot of the pressure coefficient in the gap region where the isobars in the immediate vicinity of the throat are equally spaced and fit a straight line, implying the constant and favorable axial pressure gradient. Almost no transverse pressure gradient along the  $y$ -axis appears to form.

Due to the close spacing of the cylinders to each other, the influence of the curved walls in the flow direction may be neglected and hence the walls are treated as straight and parallel. Therefore, the Poiseuille flow can be evaluated at the throat to provide an analytical solution of a *fully developed*, parabolic velocity profile

[38,39] as:

$$U_T(y) = -\frac{1}{8\mu_\infty} (G_T^2 - 4y^2) \left( \frac{dp}{dx} \right)_T \quad (4)$$

where the subscript,  $T$  indicates evaluated at the throat. To analytically evaluate the velocity profile from Eq. (4), the axial pressure gradient  $(dp/dx)_T$  was derived from the circumferential static pressure distribution on the cylinder. As shown in Fig. 4.7(a), the pressure gradient on the cylinder surface forms as a result of the stagnation affected region where the stagnation point is a center of high static pressure located approximately at  $\varphi_0 \approx 43.8^\circ$ . On the aft of the cylinder across the range of azimuth angles ( $160^\circ \leq \varphi \leq 200^\circ$ ), a low static pressure region is formed, the so-called “base pressure region,  $(C_{pb})$ ” [13,40]. The relatively low base pressure results primarily from the external flow that separates from the outer cylinder surface. The base pressure is indicative of the “dead” region behind a bluff body and has a typically constant static pressure around the rear of the cylinder [8,32].

The circumferential pressure gradient  $(dp/d\varphi)_T$  is evaluated at the azimuth angle ( $\varphi_T = 90^\circ$ ) which coincides with the  $y$ -axis. It is assumed due to the parallel and straight isobars that the axial pressure gradient across the throat can be determined from the pressure gradient on the cylinder wall if a coordinate’s transformation is performed. The transformation of the cylindrical coordinate to the Cartesian coordinate (particularly along the  $x$ -axis) may be given by:

$$x = -D / (2 \cos(\phi)) \quad (5)$$

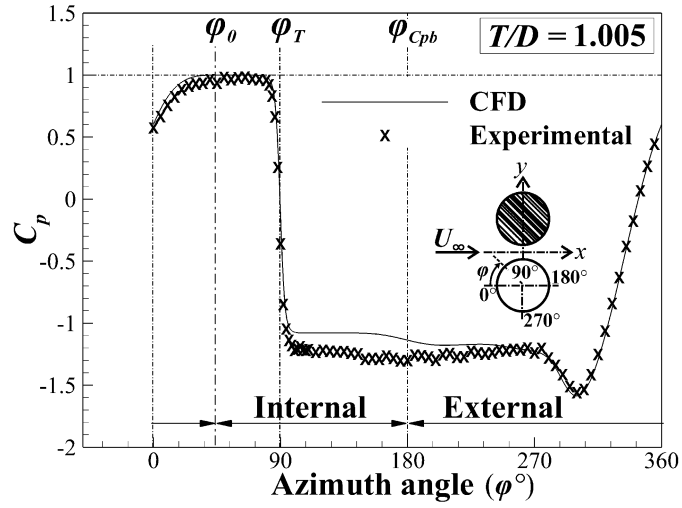
$$\left( \frac{dp}{dx} \right)_T = \frac{2}{D \sin(\phi_T)} \left( \frac{dp}{d\phi} \right)_T \quad (6)$$

From Eqs. (5,6), the circumferential pressure gradient can be transformed into the required axial pressure gradient, which can then be substituted into Eq. (4), the Poiseuille flow velocity profile equation. A comparison between the above mentioned analytical solution and the numerically estimated velocity profile is made in Fig. 4.7(b). Good agreement obtained indicates that the driving mechanism of the gap flow for the very small gap spacing is predominantly a favorable axial pressure gradient.

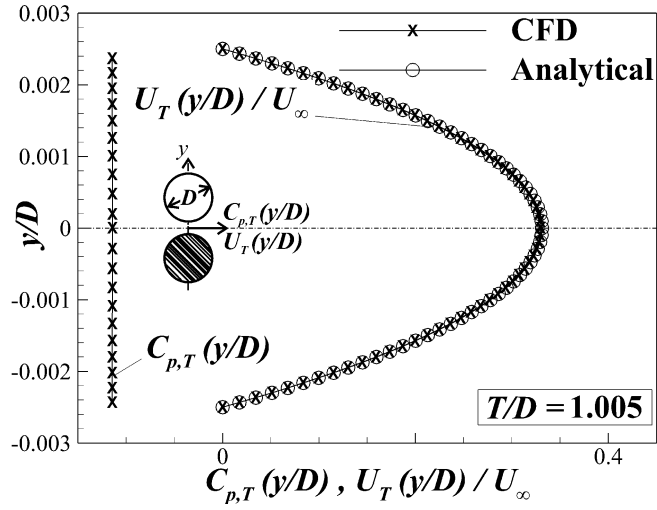
The mean axial velocity for the Poiseuille flow regime can be calculated by direct integration of the Poiseuille flow velocity profile, Eq. (4), yielding:

$$U_{T,ave} = -\frac{G_T^2}{12\mu_\infty} \left( \frac{dp}{dx} \right)_T \quad (7)$$

The mean axial velocity ( $U_{T,ave}$ ) is two-third of the axial centerline velocity ( $x$ -axis) of the gap flow; which is the location in the throat with the maximum axial velocity ( $U_{max}$ ) (i.e.,  $U_{T,ave} = 2/3 U_{max}$ ). In the Poiseuille flow regime, the transverse pressure gradient in the immediate vicinity of the throat is small enough to be considered as insignificant.

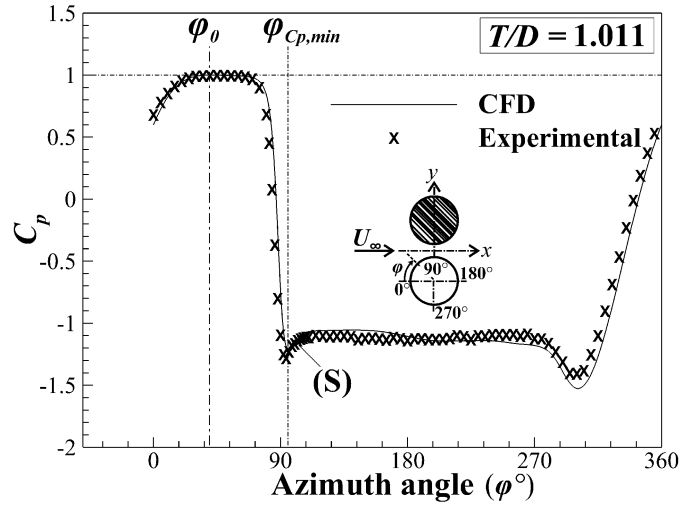


(a)



(b)

**Figure 4.7. Circumferential pressure distribution and velocity profile at the throat at  $T/D = 1.005$ ; (a) pressure coefficient distribution; (b) velocity and uniform pressure coefficient profile of the gap flow at the throat.**



**Figure 4.8. Circumferential pressure distribution at  $T/D = 1.011$ , showing the development of local minimum pressure at ( $\phi_{Cp,min} \sim 95.2^\circ$ ) where (S) indicates flow separation.**

#### **4.3. Pressure distribution transition ( $1.005 < T/D \leq 1.011$ )**

Within the spacing ratio range of  $1.005 < T/D \leq 1.011$ , a transition of the static pressure distribution occurs (Fig. 4.8). When the spacing increases from  $T/D = 1.005$  to  $T/D = 1.011$ , the fundamental change is the formation of a local minimum static pressure ( $C_{p,min}$ ) at the azimuth angle  $\phi_{Cp,min} \sim 95.2^\circ$ . An adverse pressure gradient is formed that causes an inflection point in the static pressure distribution at  $\phi \sim 98.3^\circ$ . The inflection point (S) is an indication that flow separation within the gap region takes place. With the increase in gap spacing, the stagnation point has also shifted slightly towards the geometric stagnation point and the favorable pressure gradient across the throat has reduced.

The onset of flow separation has been triggered by the increased fluid momentum of the gap flow at the larger spacing (i.e.,  $T/D = 1.011$ ). The gap flow momentum

at the throat is directly related to the mean axial velocity. It is shown in Fig. 4.5, that as the spacing ratio is increased from  $T/D = 1.005$  to  $T/D = 1.011$ , the mean velocity increases steeply. With the increased mean velocity, the fluid momentum in the gap region has possibly reached a critical value, where there is sufficient kinetic energy to initiate the roll-up of inner vortices.

The question as to why there is a dramatic increase in the fluid momentum with a small increase in spacing ratio is possibly explained by the fluid friction factor ( $f$ ) defined by [39]:

$$f = \frac{48\mu_{\infty}}{\rho_{\infty}U_{T,ave}G_T} = \frac{48}{\text{Re}_T} \quad (8)$$

The fluid friction factor is a non-dimensional parameter that relates flow resistance to the Reynolds number ( $\text{Re}_T$ ) based on the mean axial velocity ( $U_{T,ave}$ ) and the gap spacing at the throat ( $G_T$ ). In the Poiseuille flow regime, the Reynolds number for the spacing  $T/D = 1.005$  is low (i.e.,  $\text{Re}_T \approx 7$ ). With an increase of spacing to  $T/D = 1.011$ ;  $\text{Re}_T \approx 21$ . Therefore, from Eq. (8) it is apparent that the friction factor decreases.

The dominant factors that contribute to flow resistance, is shown by substituting the analytical expression Eq. (7) into Eq. (8) hence it is expressed as:

$$f \propto \frac{1}{G_T^3} \quad (9)$$

Equation (9) indicates that with a small incremental increase of spacing there is a large reduction of flow resistance. Thus, there is a dramatic increase in the mean velocity of the gap flow, despite the favorable pressure gradient across the throat having decreased, as indicated in Fig. 4.8.

#### 4.4. Velocity profile transition ( $1.025 < T/D \leq 1.05$ )

When the spacing ratio has been further increased within the range  $1.025 < T/D \leq 1.05$ , a transition is observed in the velocity profile. By comparing Figs. 4.9(a,b), it can be seen that the position of the maximum axial velocity at the throat ( $U_T$ ) is no longer found along the centerline ( $x$ -axis). The maximum axial velocities are now located away from the centerline; the peak velocity is reached on the periphery of the boundary layer. Inflection points (I) in the velocity profile develop at this transition phase (as indicated in Fig. 4.9(b)).

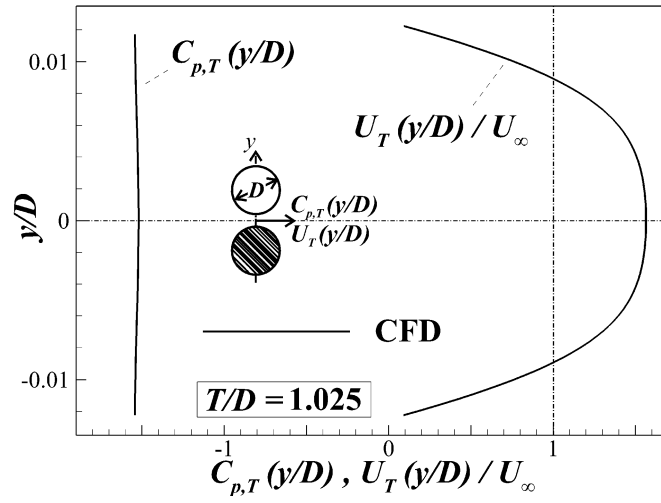
In this transition, the transverse (or radial) pressure gradient becomes substantial where the static pressure at the cylinder wall is less than at the centerline ( $x$ -axis) as shown in Fig. 4.9(b); a highly *non-uniform* pressure coefficient profile along the  $y$ -axis. The relationship between the transverse (or radial) pressure gradient due to the cylinder radius being coincident to the  $y$ -axis and the axial velocity along the  $y$ -axis is expressed as:

$$\left( \frac{\partial p}{\partial r} \right)_T = \rho_\infty \left( \frac{U_T^2}{r} \right)_T \quad (10)$$

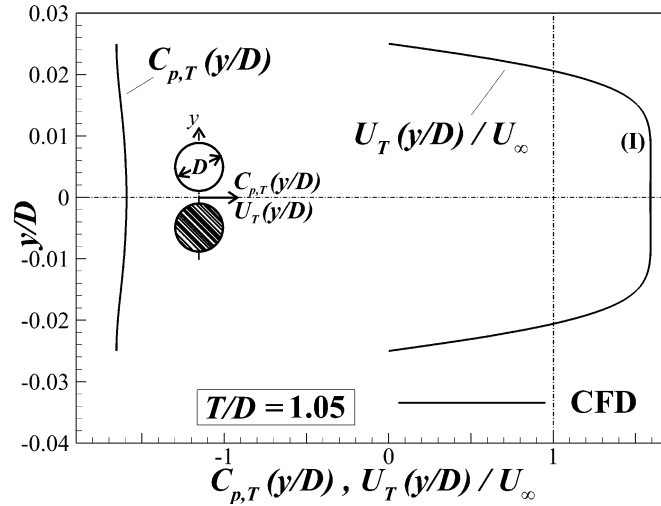
where  $r$  is the radial distance along the  $y$ -axis from the cylinder center up to the

centerline ( $x$ -axis). The transverse pressure gradient generates *centripetal forces*, causing the fluid elements to follow a curved path [41-43]. Here, flow velocity may be maintained if there is no strong axial pressure gradient. The effect of the transverse pressure gradient referring to (Eq. 10) is to accelerate the fluid closest to the cylinder wall more than the fluid near to the centerline ( $x$ -axis), the velocity profile will therefore have velocity peaks located away from the centerline as shown in Fig. 4.9 (b). The thinning of the boundary layer,  $\delta/G_T$  is also noticed in Fig. 4.9 as the gap spacing is increased.

The transition of the velocity profile coincides approximately with the onset of the unstable Biased flow regime,  $1.1-1.2 \leq T/D \leq 2-2.2$  [4,8]. This is consistent with an argument made by Schlichting [32] that “velocity profiles with a point of inflection are unstable.” In this context, “unstable” means that disturbances occurring in the flow field are naturally amplified. The development of a transverse or (radial) pressure gradient in the gap region may have an influence on the nature of the previously observed unstable wake behavior, specifically regarding the bi-stable nature of the gap flow.



(a)



(b)

**Figure 4.9. Steady velocity and non-uniform pressure coefficient profiles of the gap flow at the throat; (a)  $T/D = 1.025$ , maximum axial velocity along centerline ( $x$ -axis); (b)  $T/D = 1.05$ , maximum axial velocity near the cylinder wall and the velocity profile with inflection points (I).**

#### 4.5. Transverse pressure gradient transition ( $1.1 < T/D < 1.25$ )

A transverse or (radial) pressure gradient develops from a non-uniform pressure profile at the throat where the static pressure near the cylinder wall is decreasing

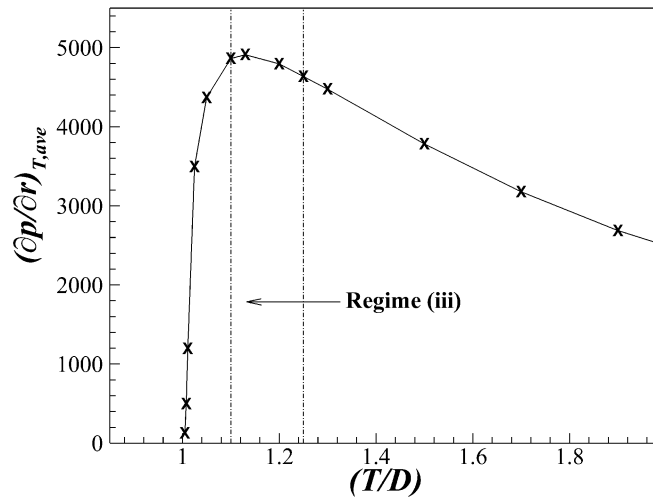
relative to the centreline ( $x$ -axis). From Eq. (10), the effect of the transverse pressure gradient is to accelerate the fluid near to the cylinder wall, causing peak axial velocities to be located away from the centreline. The transverse pressure gradient is, therefore, introduced as another gap flow driving mechanism.

From the numerical simulation results, the area weighted average or mean transverse (radial) pressure gradient across the throat that was averaged over half the gap spacing is defined as:

$$\left(\frac{\partial p}{\partial r}\right)_{T,ave} = \frac{2}{G_T} \int_{G_T/2} \left(\frac{\partial p}{\partial r}\right)_T dG_T \quad (11)$$

The variation of the mean transverse pressure gradient with spacing ratio is plotted in Fig. 4.10; which increases steeply until  $T/D = 1.13$ . At  $T/D = 1.13$ , a maximum is reached, followed by a monotonic reduction of the mean transverse pressure gradient when  $T/D > 1.13$ ; following a similar trend to the variation of the mean axial velocity with spacing ratio (Fig. 4.5). Based on the relationship between the transverse pressure gradient and the axial velocity along the  $y$ -axis from Eq. (10), we observe a correlation between the mean transverse pressure gradient and the mean axial velocity. Within the transitional range of gap spacing for Regime (iii) (i.e.,  $1.1 < T/D < 1.25$ ), the mean transverse pressure gradient reduces, consequently the mean axial velocity no longer increases as steeply compared to the smaller spacing values (Fig. 4.5).

Within Regime (iii) shown in Figs. (4.5, 4.10), we observe that the mean axial velocity continues to increase (at a reduced rate) despite the mean transverse gradient now decreasing. We can attribute the small increase of the mean axial velocity with increasing gap spacing to: (1) Reduced flow resistance due to increased gap spacing, (2) A weak favourable axial pressure gradient along the entire length (y-axis) of the throat region, (3) The degree of dispersion of the internal streamlines ahead of the cylinders continuously reduces as the spacing increases, allowing more of the free stream to enter into the gap region.

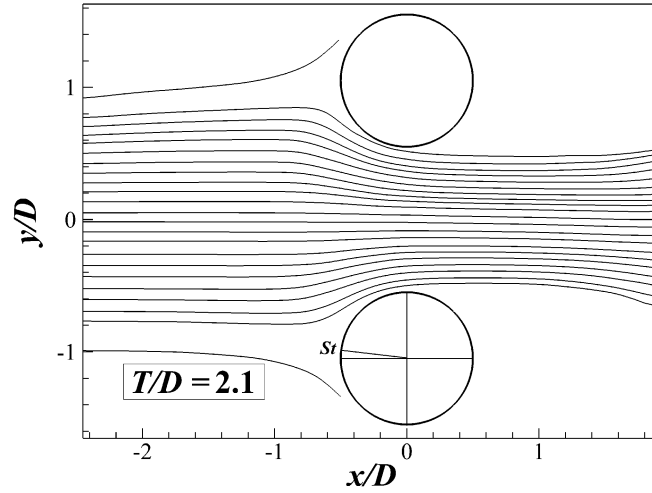


**Figure 4.10. The variation of the area weighted average of the transverse pressure gradient with spacing ratio.**

#### **4.6. Momentum driven regime ( $T/D > 1.25$ )**

The gap flow is predominantly momentum driven when the gap flow is maintained by the fluid inertia of the free-stream. With decreased dispersion of the internal streamlines the free-stream fluid momentum is no longer converted into a high static pressure region ahead of the cylinders (Fig. 4.11). The momentum

driven flow is identified as the dominant driving mechanism of the gap flow, when the mean axial velocity decreases monotonically with gap spacing (i.e.,  $T/D > 1.25$ ), following a similar trend to the variation of area ratio ( $A_0/A_T$ ) that was shown in Fig. 4.4.



**Figure 4.11. Time averaged PIV flow image showing the internal streamlines ahead of the two side-by-side cylinders, including the gap region,  $T/D = 2.1$ ,  $\varphi_0 = 7.2^\circ$ .**

The transition from pressure to momentum driven flow is apparent when the circumferential static pressure distribution on the cylinder surface is considered for the spacing ratios  $T/D > 1.25$ . In Fig. 4.12, it is noticeable how the azimuth angle of the stagnation point ( $\varphi_0$ ) and minimum pressure ( $\varphi_{Cp,min}$ ) has decreased as the spacing is increased. At  $T/D \leq 1.25$ , the location of the center of low pressure is aft of the throat i.e.,  $\varphi_{Cp,min} > 90^\circ$ . For larger spacing values (i.e.,  $T/D > 1.25$ ), the angle is reduced (shifting towards the fore side of the cylinders), for example when  $T/D = 1.3$ ;  $\varphi_{Cp,min} = 89.5^\circ$  (Fig. 4.12). This indicates that the axial pressure

gradient near to the wall has transitioned from favorable to adverse as schematically illustrated in Fig. 4.13. The adverse pressure gradient acts to counter the prevailing gap flow, thus the mean velocity is prevented from increasing and possibly slowing down the bulk of the gap flow.

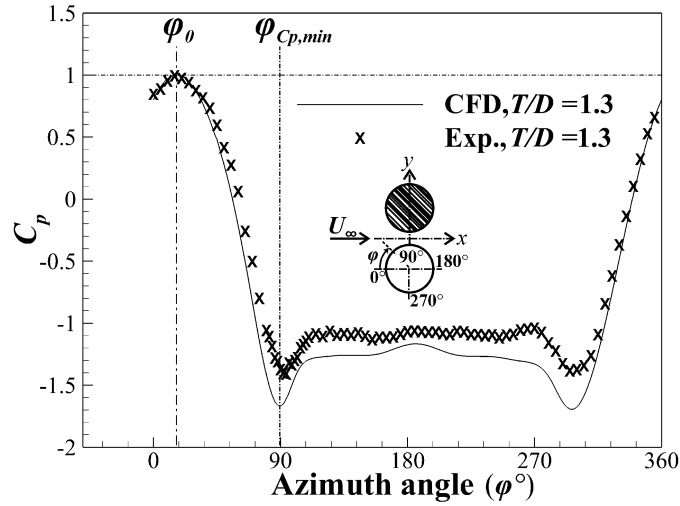


Figure 4.12. The circumferential static pressure distribution around the lower cylinder. Local minimum pressure at the throat;  $\phi_{Cp,min} \sim 89.5^\circ$  at  $T/D = 1.3$ .

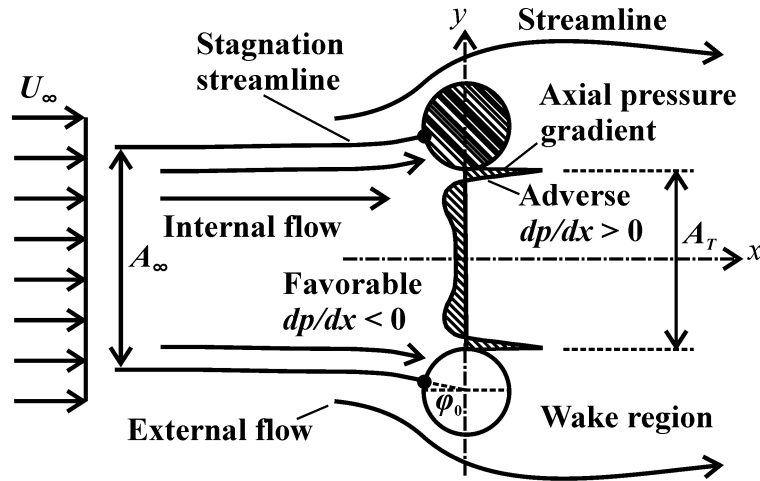
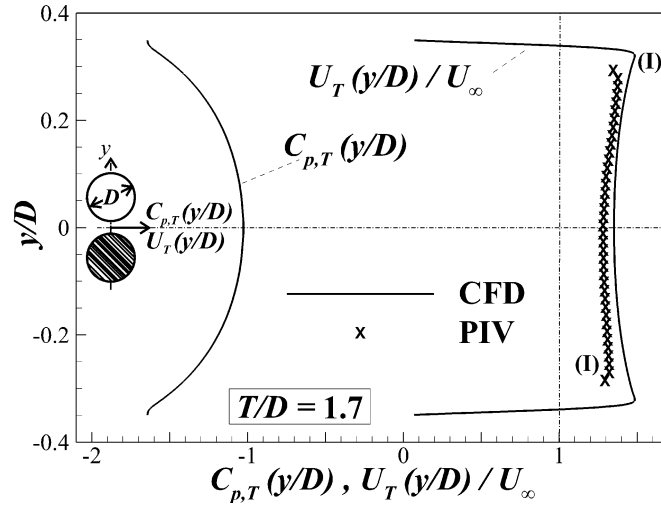


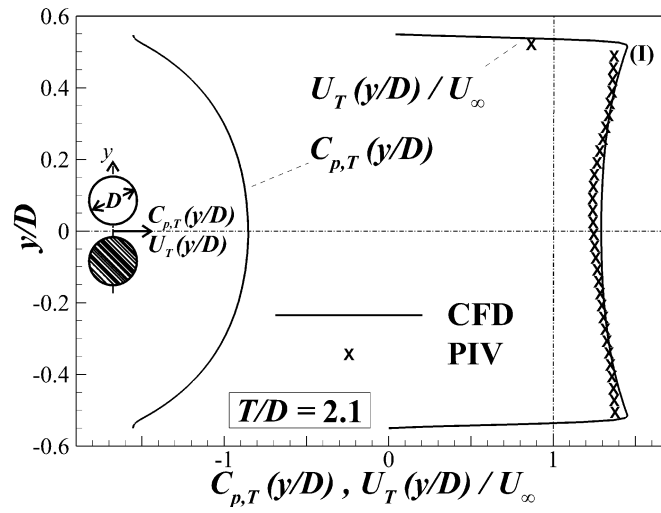
Figure 4.13. Schematic of the streamlines in the stagnation affected region and the axial pressure gradient along the throat.

The gap flow is predominantly driven as a result of the upstream flow maintaining a certain amount of fluid momentum into the entry area ( $A_0$ ). This is related to the observed stagnation points shifting towards the geometric stagnation point of a single cylinder ( $\phi_0 \rightarrow 0^\circ$ ) when the spacing is sufficiently large. When the stagnation point approaches the zero azimuth angle, the degree of dispersion of the internal streamlines is reduced. Hence, a greater portion of the upstream flow velocity ( $U_\infty$ ) is carried into the gap region so that the mean velocity at the entry, ( $U_{0,ave}$ ) approaches the free-stream velocity ( $U_\infty$ ). The increase in the flow velocity between the entry region and the throat is caused from the reduction of flow area between these two regions so that when the spacing is large, the normalized mean velocity at the throat approaches the area ratio ( $U_{T,ave}/U_{0,ave} \rightarrow A_0/A_T$ ).

The velocity profile at the large gap spacing is similar to that shown in Figs. 4.14(a,b). The steady numerical results are compared to the time averaged PIV measurements for the spacing ratio  $T/D = 1.7$  and  $T/D = 2.1$  and there is good agreement. The axial velocity close to the wall is significantly higher than at the centerline and the boundary layer is relatively small compared to the overall gap spacing ( $G_T$ ).



(a)



(b)

**Figure 4.14. Highly non-uniform pressure coefficient profile and developed velocity profile with inflection points (I), (a)  $T/D = 1.7$ , (b)  $T/D = 2.1$ .**

## 5. CONCLUSION

The present study considered steady behavior of the gap flow at the throat ( $A_T$ ) between two side-by-side circular cylinders with the gap spacing ratios ranging from  $T/D = 1.005$  to  $T/D = 6.0$ . At small gap spacing (pressure gradient driven) leads to highly dispersed internal streamlines that is related to the reduction of free-stream fluid momentum and the formation of a local high static pressure region ahead of the cylinders. For a large gap spacing (momentum driven) there is no dispersion of the internal streamlines, which indicates that the free-stream fluid momentum is maintained into the gap region.

A new further classification of the gap flow based on steady state velocity and pressure characteristics was established as:

- (1) The Poiseuille flow regime ( $1.005 \leq T/D < 1.011$ ) occurs, driven solely by a favorable axial pressure gradient.
- (2) A pressure distribution transition is found within  $1.005 < T/D \leq 1.011$ , that is associated with flow separation on the inner cylinder surface.
- (3) The velocity profile transition occurs when the spacing ratio is between  $1.025 < T/D \leq 1.05$ . The position of the maximum velocity transitions from the centerline to a location that is closer to the walls. As a consequence, inflection points are present in the velocity profile. The transition of the velocity profile coincides approximately with the beginning of the biased flow regimes ( $1.1 \sim 1.2 \leq T/D \leq 2.0 \sim 2.2$ ).

- (4) The mean transverse pressure gradient across the throat reaches a maximum and then reduces when the spacing ratio is between  $1.1 < T/D < 1.25$ , the effect is that the mean axial velocity at the throat no longer increases as steeply with spacing ratio.
- (5) The momentum driven regime is observed when  $T/D > 1.25$ . In this regime, the mean flow velocity now reduces monotonically as the area ratio decreases. Upstream of the cylinders the strength of the stagnation affected region reduces, with increasing gap spacing so that the mean velocity in the entry region approaches the free-stream velocity.
- (6) At sufficiently large spacing, the stagnation point becomes the geometric stagnation point of a single cylinder, which is indicative that each cylinder behaves independently. This is approximately consistent with the behavior of the wake structure related to the symmetric flow regime,  $T/D > 2.2$ .
- (7) Further work is required to determine how the driving mechanisms of gap flow are related to the distinct flow regimes based on wake behavior.

## REFERENCES

- <sup>1</sup>M. M. Zdravkovich, "Flow induced oscillations of two interfering circular cylinders," J. Sound and Vibration **101**, (4), 511 (1985).
- <sup>2</sup>M. M. Zdravkovich, "The effects of interference between circular cylinders in cross flow," J. Fluids Struct. **1**, 239 (1987).
- <sup>3</sup>D. Sumner, S. S. T. Wong, S. J. Price, and M. P. Paidoussis, "Fluid behavior of side-by side circular cylinders in steady cross-flow," J. Fluids Struct. **13**, 309 (1999).
- <sup>4</sup>D. Sumner, "Two circular cylinders in cross-flow: A review," J. Fluids Struct. **26**, 849 (2010).
- <sup>5</sup>Z. J. Wang, Y. Zhou, "Vortex interactions in a two side-by-side cylinder near-wake," Int. J. Heat and Fluid Flow **26**, 362 (2005).
- <sup>6</sup>M. M. Alam, and Y. Zhou, "Flow around two side-by-side closely spaced circular cylinders," J. Fluids Struct. **23**, 805 (2007).
- <sup>7</sup>H. M. Spivack, "Vortex frequency and flow pattern in the wake of two parallel cylinders at varied spacing normal to an air stream," J. Aero. Sci **13**, 289 (1946).
- <sup>8</sup>P. W. Bearman, and A. J. Wadcock, "The interaction between a pair of circular cylinders normal to a stream," J. Fluid Mech. **61**, part 3, 499 (1973).
- <sup>9</sup>M. M. Zdravkovich, "REVIEW-Review of flow interference between two circular cylinders in various arrangements," ASME J. Fluids. Eng. **99**(4), 618 (1977).

- <sup>10</sup>S. Ishigai, E. Nishikawa, K. Nishimura, and K. Cho, “Experimental study on structure of gas flow in tube banks with tube axes normal to flow,” *Bulletin of the Japan Society Mechanical Engineers* **15**, 86, 949 (1972).
- <sup>11</sup>C. H. K. Williamson, “Evolution of a single wake behind a pair of bluff bodies,” *J. Fluid Mech.* **159**, 1 (1985).
- <sup>12</sup>H. J. Kim, and P. A. Durbin, “Investigation of the flow between a pair of circular cylinders in the flopping regime,” *J. Fluid. Mech.* **196**, 431 (1988).
- <sup>13</sup>Y. Zhou, Z. J. Wang, R. M. C. So, S. J. Xu, and W. Jin, “Free vibrations of two side-by-side cylinders in a cross-flow,” *J. Fluid Mech.* **443**, 197 (2001).
- <sup>14</sup>S. J. Xu, Y. Zhou, and R. M. C. So, “Reynolds number effect on the flow structure behind two side-by-side cylinders,” *Physics. Fluids* **15**, 1214 (2003).
- <sup>15</sup>I. Afgan, Y. Kahil, S. Benhamadouche, and P. Sagaut, “Large eddy simulation of the flow around single and two side-by-side cylinders at subcritical Reynolds numbers,” *Phys. Fluids* **23**, 075101 (2011).
- <sup>16</sup>Y. Zhou, H. J. Zhang, and M. W. Yiu, “The turbulent wake of two side-by-side circular cylinders,” *J. Fluid Mech.* **458**, 303 (2002).
- <sup>17</sup>A. Agrawal, L. Djenidi, and R. A. Antonia, “Investigation of flow around a pair of side-by-side square cylinders using the lattice Boltzmann method,” *Computers & Fluids* **35**, 1093 (2006).
- <sup>18</sup>Y. Zhou, R. M. C. So, M. H. Liu, and H. J. Zhang, “Complex turbulent wakes generated by two and three side-by-side-cylinders,” *Int. J. Heat Fluid Flow* **21**, 125 (2000).

- <sup>19</sup>M. M. Alam, M. Moriya, and H. Sakamoto, “Aerodynamic characteristics of two side-by-side circular cylinders and application of wavelet analysis on the switching phenomenon,” *J. Fluids Struct.* **18**, 325 (2003).
- <sup>20</sup>V. Kolar, D.A. Lyn, and W. Rodi, “Ensemble-averaged measurements in the turbulent near wake of two side-by-side square cylinders,” *J. Fluid. Mech.* **346**, 201 (1997).
- <sup>21</sup>C. Farell, and J. Blessmann, “On critical flow around smooth circular cylinders,” *J. Fluid Mech.* **136**, 375 (1983).
- <sup>22</sup>P. W. Bearman, “The effect of base bleed on the flow behind a two-dimensional model with a blunt trailing edge,” *Aero. Quart.* **18**, 207, (1967).
- <sup>23</sup>S. Bloor, “The transition to turbulence in the wake of a circular cylinder,” *J. Fluid Mech.*, **19**, part 2, 290 (1964).
- <sup>24</sup>J. Westerweel, “Fundamentals of digital particle image velocimetry,” *Meas. Sci. Technol.* **8**, 1379 (1997).
- <sup>25</sup>R. J. Adrian, “Particle-imaging techniques for experimental fluid mechanics,” *Annu. Rev. Fluid Mech.* **23**, 261 (1991).
- <sup>26</sup>J. Westerweel, “On velocity gradients in PIV interrogation,” *Exp. Fluids* **44**, 831 (2008).
- <sup>27</sup>B. M. Wilson, and B. L. Smith, “Uncertainty on PIV mean and fluctuating velocity due to bias and random errors,” *Meas. Sci. Technol.* **24**, 035302, (2013).
- <sup>28</sup>J. P. Holman & W. J. Gajda, Jr., *Experimental Methods For Engineers* (McGraw-Hill , New York, 1978) p.7.

- <sup>29</sup>J. Westerweel, “Theoretical analysis of the measurement precision in particle image velocimetry,” *Exp. Fluids* [Suppl.] S3-S-12 (2000).
- <sup>30</sup>M. P. Wernet, “Application of DPIV to study both steady state and transient turbomachinery flows,” *Optics & Laser Technology* **32**, 497 (2000).
- <sup>31</sup>ANSYS Inc. ANSYS Fluent theory guide. Release 14.5. 2012. §4.1.1 *Reynolds (Ensemble) Averaging*, §4.13 *Near-wall treatments for wall-bounded turbulent flows*, and §4.2 *Spalart –Allmaras Model*.
- <sup>32</sup>H. Schlichting and K. Gersten, *Boundary Layer Theory* (Springer, Heidelberg, 2000).
- <sup>33</sup>ANSYS Inc. ANSYS Fluent user’s guide. Release 14.5. 2012. §12.2.1.9 *Grid resolution for RANS models*.
- <sup>34</sup>C. J. Freitas, “Editorial Policy Statement on the Control of Numerical Accuracy,” *ASME J. Fluids Eng.*, 115, 339 (1993).
- <sup>35</sup>F. Stern, R. V. Wilson, H. W. Coleman, and E. G. Paterson, “Comprehensive Approach to Verification and Validation of CFD Simulations – Part 1: Methodology and Procedures,” *ASME J. Fluids Eng.*, **123**, 793 (2001).
- <sup>36</sup>L. Eca and M. Hoekstra, “Discretization Uncertainty Estimation based on a Least Squares version of the Grid Convergence Index,” 2<sup>nd</sup> Workshop on CFD Uncertainty Analysis, Lisbon, October, (2006).
- <sup>37</sup>I. B. Celik and J. Li, “Assessment of numerical uncertainty for the calculations of turbulent flow over a backward-facing step,” *Int. J. Numer. Meth Fluids* **49**, 1015 (2005).
- <sup>38</sup>J. D. Anderson, Jr., *Fundamentals of Aerodynamics*, (McGraw–Hill International, Boston, 2007) p.863.

- <sup>39</sup>F. M. White, *Fluid Mechanics* (McGraw-Hill Book Company, New York, 1979) p.343.
- <sup>40</sup>A. Roshko, “Experiments on the flow past a circular cylinder at very high Reynolds number,” *J. Fluid Mech.*, **10**, (3), 345 (1961).
- <sup>41</sup>E. L. Houghton and P. W. Carpenter, *Aerodynamics for Engineering Students* (Elsevier Butterworth Heinemann, Oxford, 2003) p.136.
- <sup>42</sup>J. K. Vennard, *Elementary Fluid Mechanics* (Wiley & Sons, New York, 1961) p.119.
- <sup>43</sup>E. M. Greitzer, C. S. Tan, and M. B. Graf, *Internal Flows Concepts and Applications* (Cambridge University Press, Cambridge, 2004) p.173.