

**ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES**

PhD THESIS

Göktürk Memduh ÖZKAN

**SUPPRESSION OF VORTEX SHEDDING IN THE WAKE OF A CIRCULAR
CYLINDER BY ATTACHED PERMEABLE PLATES IN DEEP WATER**

DEPARTMENT OF MECHANICAL ENGINEERING

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ABSTRACT

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The aim of present study is to control the vortex shedding downstream of a circular cylinder by means of attached permeable plates. Experiments were performed in deep water using Particle image velocimetry (PIV) and flow visualization techniques. The permeable plate was made of a chrome-nickel screen which was attached on the cylinder surface along its midspan. The main parameters of the study are porosity, β of the permeable plate, the plate angle, θ and the plate length, L/d . Within this context, six porosity values, eighteen plate angles and two plate lengths were defined and studied in order to examine their effects on vortex shedding control. The results showed that the use of a permeable plate with various plate angles suppresses the vortex shedding downstream of the circular cylinder by reducing the fluctuations in the wake, elongating the vortex formation region and attenuating the vortex shedding frequency. The porosity values of $\beta=0.4$ and 0.5 with plate angles of $45^\circ \leq \theta \leq 135^\circ$ were found to be effective on suppressing the vortical flow structure around the cylinder. Within these effective intervals, this study brings out a new approach on controlling the unsteady flow structure around bluff bodies and could be suggested for engineering applications concerning the vortex shedding and consequent Vortex-Induced Vibrations for deep water flow conditions.

Key Words: Cylinder, flow control, PIV, VIV, vortex shedding

ÖZ

DOKTORA TEZİ

DERİN SUDA DAİRESEL SİLİNDİR ART İZİNDE OLUŞAN GİRDAP KOPMASININ GEÇİRGEN PLAKALAR KULLANILARAK BASTIRILMASI

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Bu çalışmanın amacı dairesel bir silindir arkasında oluşan girdap kopmasının geçirgen plakalar yardımıyla kontrol edilmesidir. Deneyler derin suda, parçacık görüntülemeli hız ölçüm (PIV) ve akış görüntüleme teknikleri kullanılarak gerçekleştirilmiştir. Geçirgen plakalar krom-nikel tellerden örülü eleklerden elde edilmiş olup, merkezi eksenini doğrultusunda silindire eklenmiştir. Çalışmanın ana parametreleri plaka geçirgenlik oranı, β , plaka açısı, θ ve plaka uzunluğu L/d 'dir. Bu kapsamda altı farklı geçirgenlik oranı, on sekiz farklı plaka açısı ve iki plaka uzunluğu belirlenmiş ve bu parametrelerin girdap kopmasına etkileri deneysel olarak incelenmiştir. Elde edilen sonuçlar, çeşitli açılar verilerek kullanılan geçirgen plakaların art izi bölgesindeki çalkantıları azalttığı, bu bölgeyi akım yönünde uzattığı ve girdap kopma frekansının düşürmek suretiyle girdap kopmasını bastırdığını göstermektedir. $\beta=0.4$ ve 0.5 açıklık oranlarının $45^\circ \leq \theta \leq 135^\circ$ plaka açısı aralığında silindir ardında girdaplı akış yapısını kontrol etmede etkili olduğu sonucuna varılmıştır. Bu çalışma, elde edilen etkili parametreler içerisinde küt cisimler ardında daimi olmayan akış yapısının kontrol edilmesi bağlamında yeni bir yaklaşım getirerek, girdap kopması ve buna bağlı olarak oluşan girdap kaynaklı titreşimlerin mevcut olduğu mühendislik uygulamaları için tavsiye edilebilir nitelik taşımaktadır.

Anahtar Kelimeler: Akış kontrolü, girdap kopması, PIV, silindir, VIV

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LIST OF ABBREVIATONS

d	: Cylinder diameter
β	: Porosity
L	: Plate length
θ	: Plate angle
h_w	: water height
$\langle U \rangle$	: Time averaged streamwise velocity
$\langle V \rangle$	: Time averaged velocity vectors
$\langle y \rangle$	: Time averaged streamline
$\langle w \rangle$	: Time averaged vorticity
U_∞	: Free stream velocity
$\langle u\phi \rangle$	: Reynolds shear stress
TKE	: Turbulent Kinetic Energy
FFT	: Fast Fourier Transform

1. INTRODUCTION

1.1. Definition of Bluff Body and Flow around a Cylinder

If a body generates flow separation over its surface when subjected to a fluid stream, the term bluff is used and has been widely investigated for more than a century. The separation point depends on the shape of the body, i.e. it is fixed at salient edges on sharp edged bodies, whereas for the bodies having continuous surface curvature, separation depends both on the shape of the body and the state of the boundary layer. Separation results in vortex formation in the wake of bluff bodies and after a critical velocity of the flow, shedding of these vortices starts. Most of the studies including vortex shedding are carried out on the cylinders having circular, square and rectangular cross-section but flow around circular cylinder is still the principal subject of researchers due to its engineering significance and in part due to its tempting simplicity in setting up arrangements in an experimental or computational laboratory.

1.1.1. Vortex Shedding and Flow Regimes

Vortex shedding around bluff bodies is a widely investigated phenomenon since the observation of Strouhal who studied vortex shedding around circular cylinder (Strouhal, 1878). The alternate vortex shedding of vortices in the near wake leads to a large fluctuating pressure forces which may cause structural vibrations and resonance resulting in failure of even large structures. Because of this failure potential, bluff body wake flows have direct engineering significance. Some practical examples are; flow past an automobile and a submarine, wind blowing past a high-rise building and a bridge, heat exchanger tubes, cooling towers, power lines, trees, and underwater pipelines, the lift developed by airplane wings, upward draft of rain, snow, hail, and dust particles in high winds; the transportation of red blood cells by blood flow, the entertainment and disbursement of liquid droplets by sprays, the vibration and noise generated by bodies moving in a fluid etc. (see Figure 1.1).



Figure 1.1. Some engineering applications concerning the flow around bluff bodies

Cross-flow over a cylinder exhibits complex flow patterns. If a circular cylinder comes across by a flow, flow separation, vortex shedding and shear layer instability etc. take place in different Reynolds numbers and as a result of these there would be unavoidable lift and drag forces exerted to the cylinder. Dynamics of flow around a cylinder depends only on the Reynolds number ($Re = U_{\infty} D / \nu$ where U_{∞} is the free stream velocity far from the cylinder, D is the diameter of the cylinder and ν is

the kinematic viscosity). From creeping flows ($Re \ll 1$) to fully turbulent flows ($Re \gg 10^7$) there are different flow regimes which can be classified as follows:

The fluid approaching the cylinder branches out and encircles the cylinder, forming a boundary layer that wraps around the cylinder. At very low upstream velocities ($Re < 6$) as shown in Figure 1.2 (a) the fluid completely wraps around the cylinder and the two arms of the fluid meet on the rear side of the cylinder in an orderly manner. Thus, the fluid follows the curvature of the cylinder. At higher velocities, i.e. for the range $6 < Re < 47$, the fluid still hugs the cylinder on the frontal side, but it is too fast to remain attached to the surface as it approaches the top (or bottom) of the cylinder, therefore the flow separates from the trailing edge of the cylinder and stationary vortex pair is generated in the wake whose recirculation length grows as the Reynolds number increases. As a result, the boundary layer detaches from the surface, forming a separation region behind the cylinder as seen by Figure 1.2 (b). In this range of Reynolds number, the flow in the wake region is characterized by periodic vortex formation and pressures much lower than the stagnation point pressure. At the critical Reynolds number ($Re \gg 47$), even though all the imposed conditions are being held steady, instability increases suddenly and vortex shedding appears behind the circular cylinder, resulting in the well-known von Karman vortex street which is presented in Figure 1.2 (c). In the case of higher Reynolds values, i.e. $Re > 200$, the flow becomes three dimensional and turbulent.

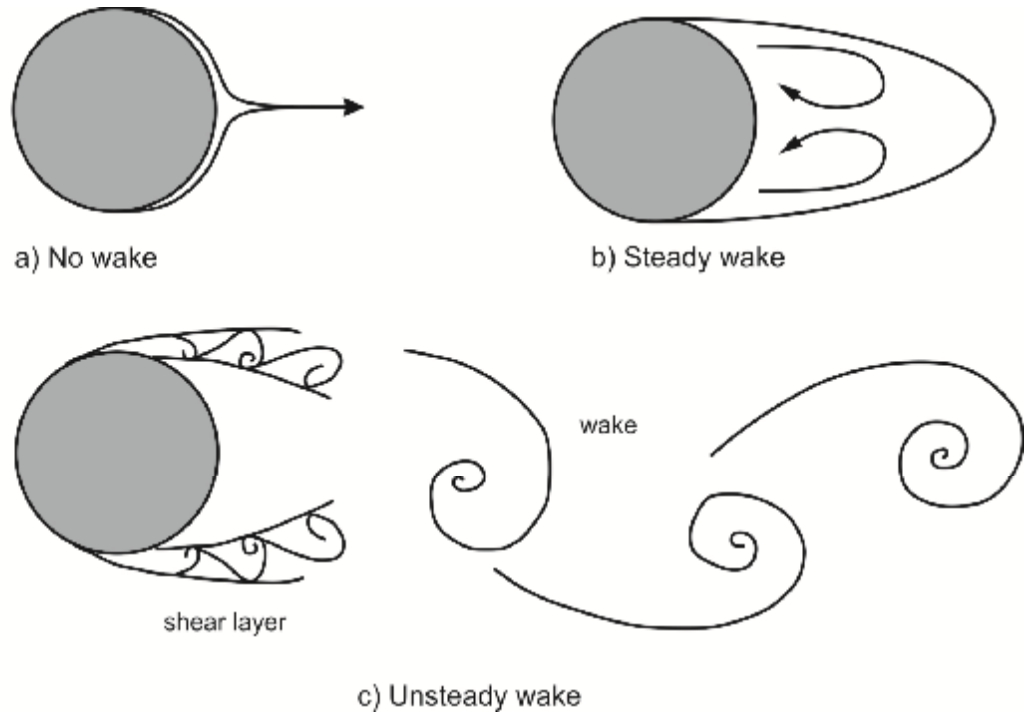


Figure 1.2. Wakes of the cylinder for different Reynolds intervals a) Creeping flow for $Re < 6$, b) The vortical wake of the cylinder for $6 < Re < 47$, c) Unsteady wake forming von Karman vortex street for $47 < Re < 200$

Figure 1.3 illustrates the mechanism of vortex shedding proposed by Gerrard (1964) and related an instantaneous PIV data showing the vorticity contours downstream of a circular cylinder. When a vortex starts growing (say the upper vortex) being fed with vorticity from the respective shear layer, it draws the shear layer on the opposite side across the wake (lower). This drawn shear layer (forming lower vortex) carries fluid with oppositely signed vorticity along with it and thus annihilates and cuts the vorticity supply to the upper vortex resulting in the shedding of the upper vortex. In the next turn, this lower vortex when grown sufficiently, would pull the upper shear layer across the wake (carrying fluid with oppositely signed vorticity) and due to vorticity annihilation, lower vortex would be shed downstream. This process repeats itself leading to alternate shedding of vortices from either side forming a vortex street downstream of the body.

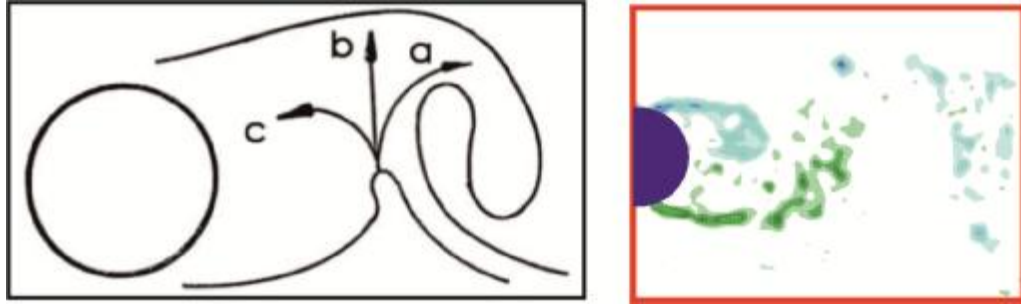


Figure 1.3. The vortex shedding mechanism by Gerard (1964) and instantaneous vorticity contours of PIV data downstream of a circular cylinder

1.1.2. Drag Force on a Cylinder

The nature of the flow across a cylinder or sphere strongly affects the total drag coefficient C_D . Both the friction drag and the pressure drag can be significant. The high pressure in the vicinity of the stagnation point and the low pressure on the opposite side in the wake produce a net force on the body in the direction of flow. The drag force is primarily due to friction drag at low Reynolds numbers ($Re < 10$) and to pressure drag at high Reynolds numbers ($Re > 5000$). Both effects are significant at intermediate Reynolds numbers. The average drag coefficients, C_D for cross-flow over a smooth single circular cylinder is given in Figure 1.4. The curves exhibit different behaviors in different ranges of Reynolds numbers:

For $Re \leq 1$, creeping flow occurs and the drag coefficient decreases with increasing Reynolds number. There is no flow separation in this regime. At about $Re=10$, separation starts occurring on the rear of the body with vortex shedding starting at about $Re \approx 90$. The region of separation increases with increasing Reynolds number up to about $Re = 10^3$. At this point, the drag is mostly (about 95 percent) due to pressure drag. The drag coefficient continues to decrease with increasing Reynolds number in this range of $10 < Re < 10^3$ (a decrease in the drag coefficient does not necessarily indicate a decrease in drag. The drag force is proportional to the square of the velocity, and the increase in velocity at higher Reynolds numbers usually more than offsets the decrease in the drag coefficient). In the moderate range of $10^3 < Re < 10^5$, the drag coefficient remains relatively constant. This behavior is characteristic of blunt bodies. The flow in the boundary

layer is laminar in this range, but the flow in the separated region past the cylinder or sphere is highly turbulent with a wide turbulent wake.

There is a sudden drop in the drag coefficient somewhere in the range of $10^5 < Re < 10^6$ (usually, at about 2×10^5). This large reduction in C_D is due to the flow in the boundary layer becoming turbulent, which moves the separation point further on the rear of the body, reducing the size of the wake and thus the magnitude of the pressure drag. This is in contrast to streamlined bodies, which experience an increase in the drag coefficient (mostly due to friction drag) when the boundary layer becomes turbulent. Flow separation occurs at about 80° (measured from the front stagnation point of a cylinder) when the boundary layer is laminar and at about 140° when it is turbulent. The delay of separation in turbulent flow is caused by the rapid fluctuations of the fluid in the transverse direction, which enables the turbulent boundary layer to travel farther along the surface before separation occurs, resulting in a narrower wake and a smaller pressure drag. Keep in mind that turbulent flow has a fuller velocity profile as compared to the laminar case, and thus it requires a stronger adverse pressure gradient to overcome the additional momentum close to the wall. In the range of Reynolds numbers where the flow changes from laminar to turbulent, even the drag force F_D decreases as the velocity (and thus the Reynolds number) increases. This results in a sudden decrease in drag of a flying body (sometimes called the drag crisis) and instabilities in flight (Çengel and Cimbala, 2007).

After this sudden reduction, an increase in drag coefficient exists at higher Reynolds numbers as seen by Figure 1.4. This can be explained by the increase in frictional drag, i.e. the fluid contacts with larger surface area when the separation angle gets larger values such as 140° . According to this larger surface contact between the fluid and the body surface, the frictional drag slightly increases by increasing Reynolds number, and this increases the overall drag force exerted on the body.

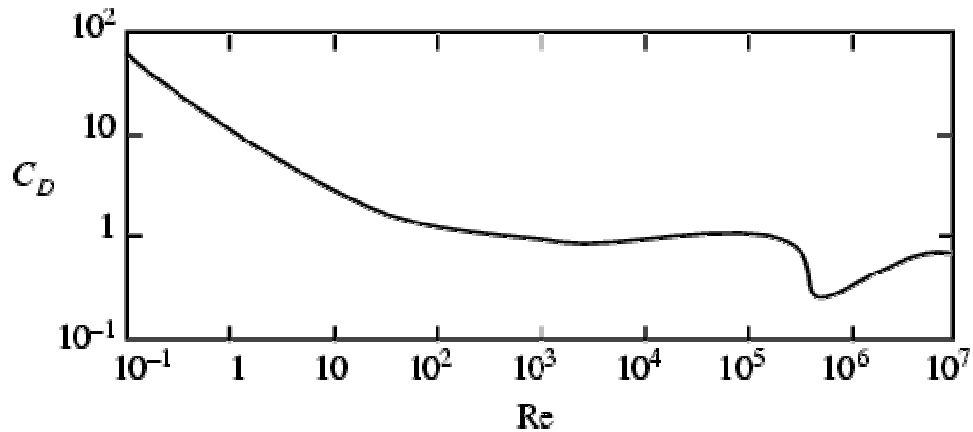


Figure 1.4. Change in drag coefficient, C_D with Reynolds number

1.1.3. Karman Vortex Street and Kelvin-Helmholtz Vortices

Two types of streamwise vortices have been reported in the laminar and transitional wakes of cylinders. At a sufficiently high value of Reynolds number, so-called “transition waves” appears in the shear layer separating from the surface of the cylinder. These waves are detectable in the region of the wake extending from the separation point on the body to the first appearance of the large-scale Karman vortex as seen on Figure 1.5. As a result, a vortex is shed from the cylinder when the local pressure changes each time. Therefore, the cylinder is subjected to a fluctuating drag and lift in addition to the mean drag and lift. The drag, lift and vortex shedding frequency are normally expressed in non-dimensional forms as the drag coefficient C_D , the lift coefficient C_L , and the Strouhal number St . These three non-dimensional parameters depend on the nature of both the approaching flow and the cylinder. When the cylinder is located near a plane boundary, the bed proximity and the velocity gradient in the boundary layer, as well as the turbulent characteristics of the boundary layer, also have some influence on these parameters.

When the value of Reynolds number is sufficiently high, then these transition waves take the form of train of Kelvin-Helmholtz vortices shown in Figure 1.5. In the bluff body community, these small-scale vortices are often referred to as Bloor-Gerrard vortices. The onset of “transition waves” is expected to be strong function of the disturbance level of the free stream, in view of the fact that they arise from convective-type instability, as described by Ho and Huerre (1984). Bloor (1964)

detected the small-scale vortices for the range of Reynolds number $Re > 1300$, however, Ünal and Rockwell (1988) first noted these vortices at $Re_d = 1900$. Wu et al. (1994) discussed the possible influence of background disturbance levels and observed these waves over the range of Reynolds numbers $1000 \leq Re \leq 3000$. Some natural instances of Karman Vortex Street and the Kelvin- Helmholtz vortices are shown in Figure 1.6.

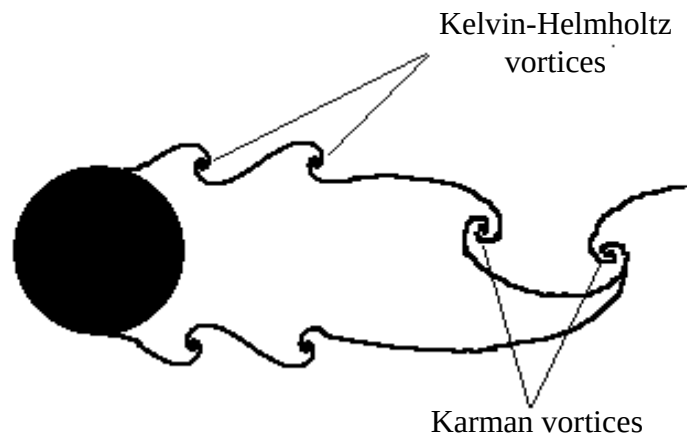


Figure 1.5. The Kelvin-Helmholtz (secondary) vortices and the Karman vortices from circular cylinder



Figure 1.6. Natural samples of Karman Vortex Street (left) and Kelvin-Helmholtz Vortices (right)

(<http://www.wired.com/2011/06/weird-world-of-fluids/?viewall=true>)

1.1.4. Strouhal Instability and Vortex Lock-in

Vortex lock-in is traditionally described as the coalescence of the Strouhal shedding frequency ($St = fD/U_\infty$ where f : frequency, D : cylinder diameter and U_∞ : free stream velocity) and the structural frequency, such that for a range of free stream velocities, the shedding frequency tunes or locks itself to the natural frequency of the structure. It is believed that the motion of the body affects the vortex shedding pattern such that the vortices are shed at the frequency of structural motion. The phenomenon is thought to allow a wider range of velocities for a system resonance with possible catastrophic consequences (Gharib, 1999).

Tacoma Narrows Bridge disaster as a result of development of Strouhal instability. The wake of the bridge because of wind created vibrations at a particular frequency and structural vibrations got into resonance. This resonant mode eventually leads to a catastrophic vibration that destroys the bridge. The oscillations of the bridge may amplify the vortex shedding and force the vortex street to occur with large amplitude. As a result, total collapse of the bridge occurred as seen in Figure 1.7.

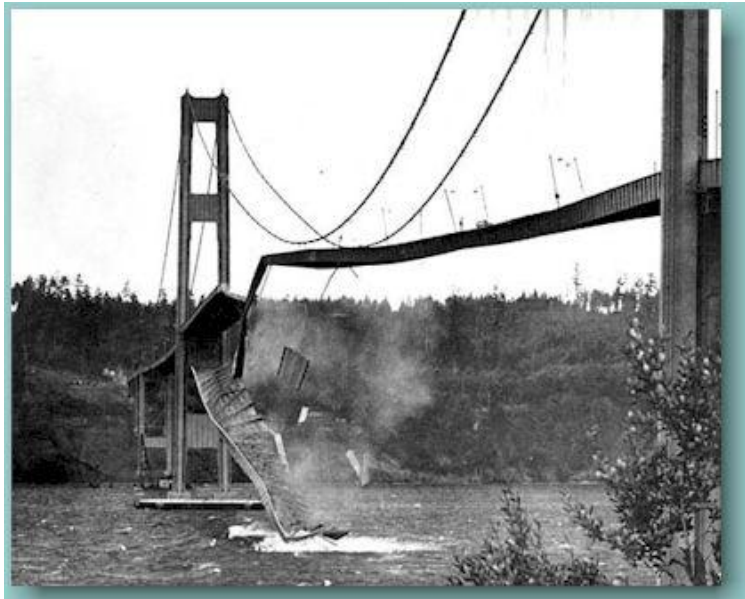


Figure 1.7. The collapse of Tacoma Narrows Bridge because of vortex lock-in (<http://www.lib.washington.edu/specialcollections/collections/exhibits/tnb>)

1.2. Flow Control around Bluff Bodies

Due to the adverse effects of vortex shedding as depicted in previous sections, the control of flow around bluff bodies is significantly required and has been studied extensively in the past decades. In general, control of flow can be divided into two groups as passive and active control methods. The passive technique requires no external energy, producing desired effects on flow by changing structural geometries, adding grooves, shrouds or near-wake stabilizers to structures. Some examples of passive control methods are dimple, surface modification, helical wire, splitter plate, small secondary control cylinder. The active technique involves energy input via the use of actuators to bring about desirable changes to the fluid–structure system using either an independent external disturbance, i.e. the open-loop (feed forward) control or i.e. the closed-loop control (feedback system). Applications of these techniques are oscillation of bluff body, rotating cylinder, electromagnetic forcing, blowing /suction, distributed forcing, synthetic jet for active open-loop and optimal control theory, reduced order model (ROM), single and multiple sensor for feedback control respectively.

A new insight into the classification of control methods come into the literature by Gad-el-Hak (2000) in which he classified the recent bluff-body flow controls into three groups (passive, active open-loop, and active closed-loop controls, respectively) for cases of actuator(s) without power input, actuator(s) with power input but no sensor, and sensor(s) and actuator(s) with power input. Furthermore, Choi et al. (2008) went a step further in case of this classification; they addressed the topic from a different perspective and in addition to the classification of Gad-el-Hak (2000), two different categories have been taken into consideration: (a) 2D versus 3D forcing and (b) boundary-layer control versus direct wake control. Here, 3D forcing is the control method in which the actuation property varies along the spanwise (or azimuthal) direction, whereas it does not vary in 2D forcing. A schematic presentation of this categorization on flow control is shown in Figure 1.8.

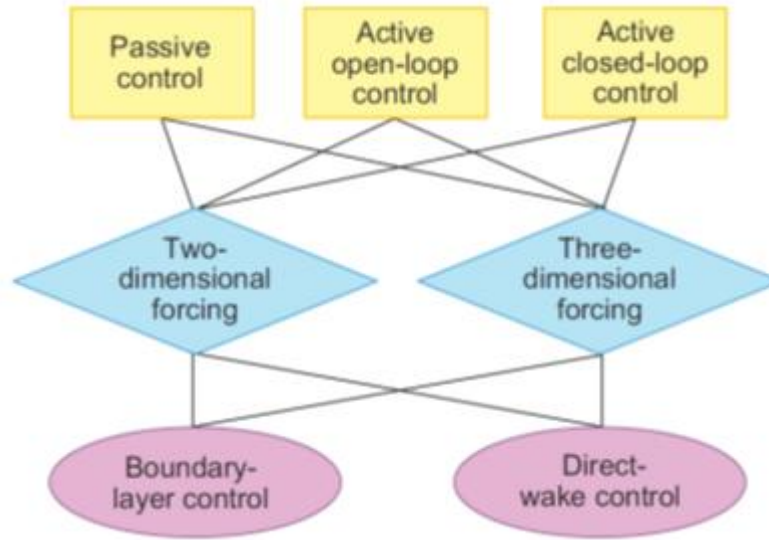


Figure 1.8. Classification of control methods flow over a bluff body (Choi et al. 2008)

1.2.1. Passive Control Techniques

Kumar et al. (2008) has taken out an impressive review including the passive control of vortical flow structure around bluff bodies and possible Vortex Induced Vibrations (VIV). They pointed out that the vortex suppression can be achieved by various passive control means such as by increasing damping, by avoiding resonance, by streamlining the structural geometry, by providing add-on devices and through body surface modifications. Some examples are of passive control methods are shown in Figure 1.9. The following sections include the findings of Kumar et al. (2008) in their passive control review.

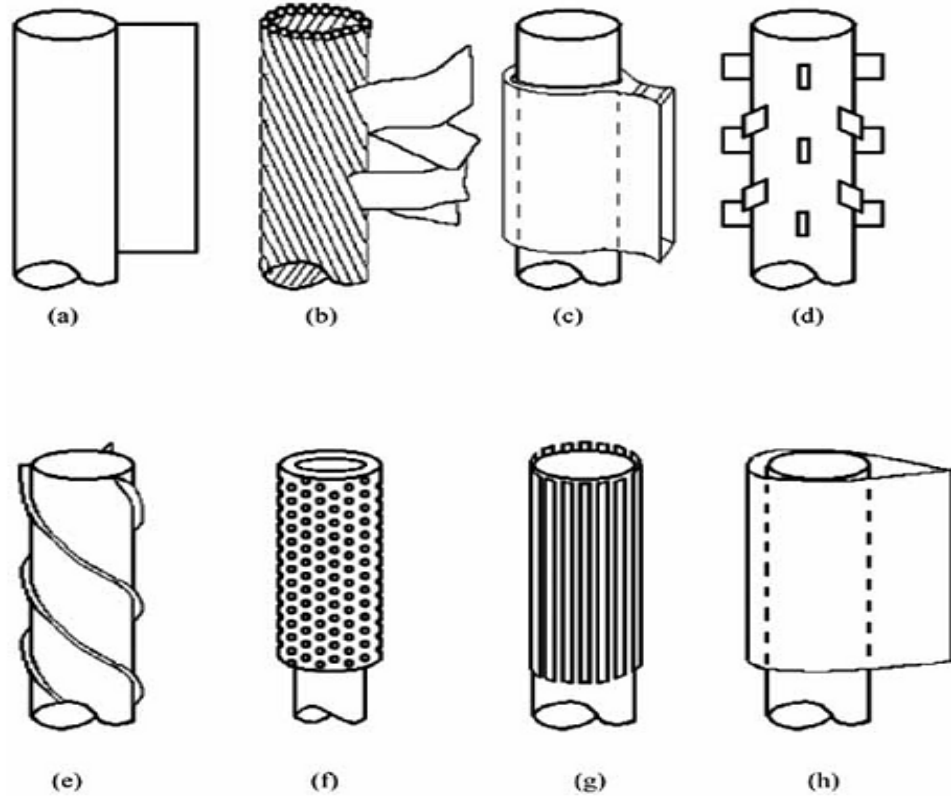


Figure 1.9. Passive control devices for suppressing VIV; (a) splitter plate, (b) ribbons, (c) guide vanes, (d) small-size tabs, (e) helical strake, (f) perforated shroud, (g) axial slats, (h) streamline fairings (Blevins, 1990, Kwon, 2002)

1.2.1.1. Increasing Damping

One of the means is to increase the reduced damping (Scruton number, k_s) of the fluid-structure system. If the reduced damping exceeds a value of 64, then the peak amplitudes at resonance would come down to within 1% of the cross stream dimension of the body. The Scruton number is given by, $k_s = 2.m.\delta/\rho D^2$, where m is the mass per unit length of the structure, δ is the logarithmic decrement, ρ is the fluid density and D is the cross stream dimension of the body. Hence, reduced damping could be varied by varying either one or more of these parameters. Additionally, motion-dependent fluid damping (viscous damping) could also influence the structural vibratory response quite significantly, which arises out of the viscous shearing of the fluid at the surface of the structure and flow separation

(Zhang et al., 2003). Fluid damping becomes predominant at low reduced velocities ($U/(fD) < 3.0$, where U is the approach flow velocity and ' f ' is the structural natural frequency).

To weaken or even to suppress VIV, higher damping could be introduced by utilizing scrapings of materials having high internal damping such as visco-elastic materials between structural members or by using external dampers (Blevins, 1990). A good example for the applications of flow control by increasing damping is the Stockbridge damper which has been used to reduce VIV of power transmission lines (Hagedorn, 1982). Dulhunty (2004) has patented an invention on Stockbridge vibration damper for power lines as seen in Figure 1.10. Stockbridge damper provides damping through two mechanisms (a) by exerting a force out-of-phase with the power line vibrations, and (b) by dissipating the vibrational energy through inter-strand friction in the messenger cable.

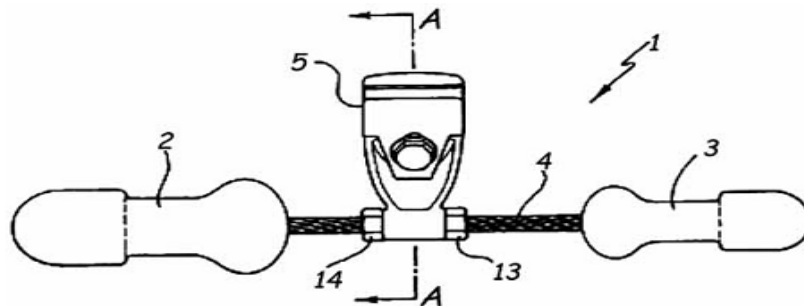


Figure 1.10. Stockbridge damper for overhead power lines invented by Dulhunty, 2004; 1-Stockbridge damper, 2&3-damping weights, 4-stranded steel cable, 5-clamp 13&14-claming points on the cable

1.2.1.2. Stream-Lining the Geometry

The vortex formation and shedding giving rise to pressure fluctuations is the primary cause for the occurrence of VIV and occurs in structures with bluff sections. Hence, stream-lining the structure (reducing the bluffness) is another option for controlling VIV. When vortex shedding is weakened or suppressed, the drag also would be reduced. For streamlining, a tapering of 1:6 is normally provided to the structure in the flow direction. Streamline fairings (also called tear-drop fairings) are

one of the effective streamlining devices generally used to mitigate vortex-induced vibrations. They are found to reduce VIV by about 80% with a drag coefficient of about 0.1 to 0.3. They are short, aerofoil shaped sections that swing freely about the body of the structure and they align with the currents to streamline the water flow around the structure so that the forces induced due to vortex shedding are minimized (Figure 1.9-h). Streamline fairings are also used in applications such as piers, oil pipes to reduce vortex-induced vibrations. But, fairings even though effective in reducing vibrations, have a disadvantage that they are expensive (large installation costs), quite large in size and difficult to handle. Also, they could be subjected to a change of geometry due to the buildup of marine growth (such as barnacles) and thus, could lose their streamline shape. There is a possibility of increased surface roughness also due to which the effectiveness of fairings could reduce with service. Streamlining is most effective in cases where the direction of flow is fixed relative to the structure and structure has sufficient stiffness to avoid flutter. But, in practical situations, change in the flow direction could be expected to occur due to which the effectiveness of fairings would be adversely affected, for fixed fairings. Rotatable fairings (adjustable with flow incidence angle) developed later compensates for this lacunae.

1.2.1.3. Introducing Add-on Devices

Introducing helical strakes one example of a 3D geometric modification or ribs is one of the proved and widely used control measures in various industrial and offshore applications to contain VIV. Figure 1.9-e shows typically how the helical strakes are mounted on to a cylindrical structure; strakes are helically wound on the structural member in a definite fashion. Basically, they adversely affect the shear layer roll up and disrupt the vortex formation and shedding process. Thus, the induced vibrations are curtailed. An engineering application of helical strakes used around a pipe used in underwater is shown in Figure 1.11.



Figure 1.11. Helical strakes used around a pipe
(<http://web.mit.edu/shear7/experiments.html>)

Introducing a perforated shroud is another way to attenuate vortex shedding. Basically, this device is affecting the entrainment layers and thus the vortex shedding phenomenon. Typically, Figure 1.9-f shows perforated shroud mounted on to a cylindrical body. It consists of a relatively thin metal cylinder that is held off the main cylinder by struts.

The main advantage of perforated shroud (with circular holes) is the lower drag it contributes. A modified version of shrouds called ‘axial slats’ Figure 1.9-g are also effectively used for reducing VIV in marine structures. The slat shroud outside diameter is generally kept at nearly $1.3D$ (D is the main cylinder diameter) with 40% percentage of open area. When compared to perforated shroud and helical strakes, slats are found to be better performing as a vortex suppression device. The main disadvantage is that it is comparatively more expensive.

Splitter plate shown in Figure 1.9-a is another efficient way to control vortex shedding and attenuate VIV in offshore engineering structures. It is a flat plate attached on to the rear end of the body extending along its length. This device prevents the switch of the confluence point, thus act as a near wake stabilizer. A

stream wise length of 4 to 5D is found to be effective for a wake splitter. For stabilizing the near-wake, methods like providing base bleed and introducing slits across the body could also be adopted to curb vortex-excitation. Even though wake splitters are effective, they suffer from a disadvantage that, they could be used only in those situations where the fluid flow direction does not vary significantly. Flags which are very similar to wake splitters (but flexible) could also be used for VIV suppression. Actually flags are not as effective as wake splitters, but they could be easily wrapped around the cylinder and remain somewhat effective with varying flow directions. However, they are not much used in subsea applications primarily due to the fact that they would eventually get completely wrapped around the cylinder and become ineffective. Another disadvantage is the difficulty and expense in attaching them to the structure, along the length of the structure.

Another cheaper as well as effective method for controlling VIV is to provide ribbons or hair cables Figure 1.9-b. Generally, a ribbon width of 1 to 2D is employed with a length of about 6 to 10D, thickness of 0.05D and spacing of about 1 to 3D. Ribbons (add-on ribbons) made of polyurethane film are utilized in practice. This type of vortex suppression device has many advantages over other devices like helical strakes. They include simple design, ease of fabrication, and convenience of handling the assembled structure, low maintenance and overall reduction in cost. The authors have also reported that a reduction in the aerodynamic drag could be achieved apart from suppressing VIV.

To mitigate VIV in structures, guide vanes could be used Figure 1.9-c. It is similar to a stream line fairing but with the trailing edge relatively bluff, also having a shorter body. The design of a typical guide vane would include a plate length (beyond the rear of the main cylinder) of 1D with a lateral separation between trailing edges of 0.9D. Such guide vanes are found to be effective in completely suppressing vortex-induced vibrations. Yet another passive device is an array of spoiler plates mounted on to the cylindrical member Figure 1.9-d. They disrupt vortex formation and shedding and thus, suppress the induced vibrations

1.2.1.4. Modifications on Body Surface

Another effective way of controlling the vortex formation and consequent vortex shedding is to apply some modifications on body surface. This method is different than the introduction of add-on devices since its application takes place the whole structure during construction process, not later on. As effective control methods, passive and active open-loop controls varying along the spanwise (or azimuthal) direction, called 3D forcing, have been applied near the separation point to control nominally 2D wake (Choi, 2008). Some examples are shown in Figure 1.12.

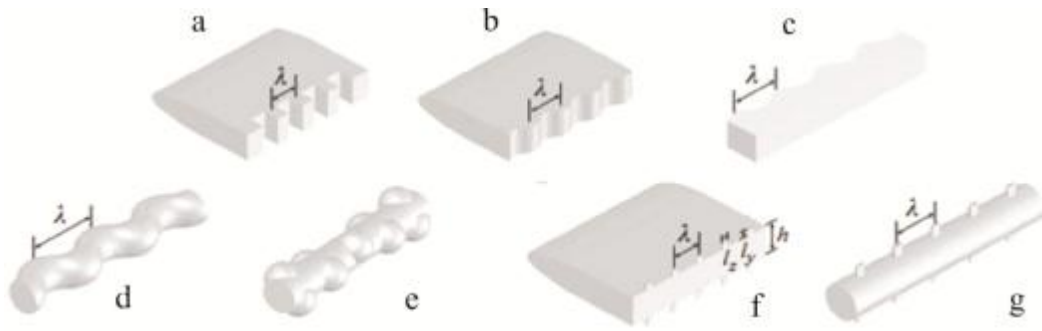


Figure 1.12. 3D forcing by passive means: (a) segmented trailing edge, (b) wavy trailing edge, (c) wavy stagnation face, (d) sinusoidal axis, (e) hemispherical bump, and (f, g) small-size tab (Choi, 2008)

1.2.1.5. Using a Small Rod

Vortex shedding past a circular cylinder is controlled over a limited range by the proper placement of a smaller control cylinder close to the main cylinder. There exists a domain close to the main cylinder where the placement of a control cylinder can completely suppress the vortex shedding for flows at a Reynolds number of 80 or less. The actual extent of this domain depends on the Reynolds number of the flow and the ratio of the diameter of the both two cylinders. Even though the flow remains unsteady for Reynolds numbers larger than 80, the presence of the control cylinder has a significant effect on the flow. It has also been reported that in certain cases

suppression of vortex shedding is accompanied by a significant reduction in the mean drag coefficient (Strykowski and Sreenivasan, 1990).

1.2.2. Active Control Techniques

1.2.2.1. Electro-Magnetic Control

The main advantage of electro-magnetic control strategy for electrically conducting fluids is that Lorentz force acts as a volume force inside the fluid, and it may be tuned to act in specific region of the flow. Many techniques using electric and magnetic fields have been developed in weakly conducting fluids since 1960s so as to achieve turbulent boundary layer control, transition delay and drag reduction (Chen, 2005).

The cylinder body is covered with electrodes and the permanent magnets side by side in the order positive electrode, N-pole magnet, negative electrode, S-pole magnet, a sequence which is repeated once more ending with a positive electrode in the axial direction of the cylinder (Figure 1.13). If it is desired Lorentz force's direction can be changed by altering the poles of each electrode.

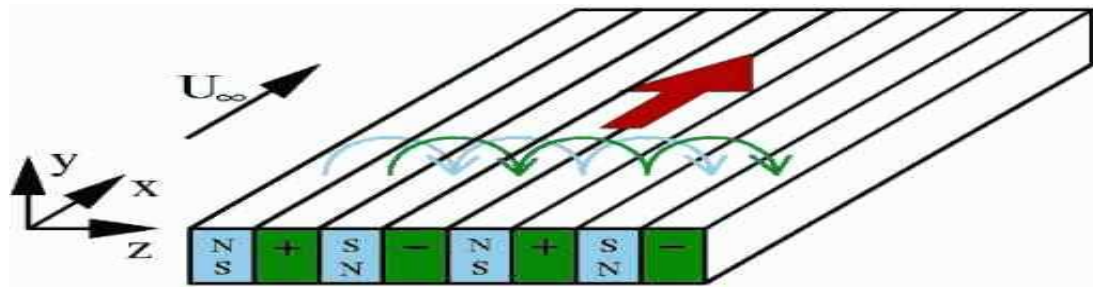


Figure 1.13. Sketch of the electric (green) and magnetic (blue) field lines and the resulting Lorentz force (red arrow) over a flat plate (www.hzdr.de)

1.2.2.2. Rotary Oscillations of Cylinders

The method uses small amplitude transverse cylinder oscillations with displacement of a few percent of the cylinder diameter. The degree of flow control

that can be achieved by utilizing rotating control cylinders depends on variety of parameters. In this technique, an oscillating rotational motion is applied to the circular cylinder around which the fluid is flowing (Figure 1.14). The resulting mutual interaction between the rotationally oscillating body and adjacent fluid modifies the wake flow pattern by accelerating and decelerating the flow around the upper and lower parts of the cylinder.

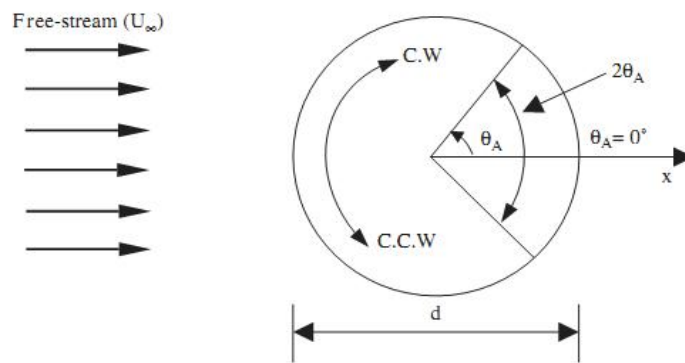


Figure 1.14. Schematics of the rotationally oscillating motion of a circular cylinder (Lee et al., 2006)

1.2.2.3. Rotating Circular Cylinder

The flow is controlled by a rotating circular cylinder which is placed in the flow field. At this control method, the forcing frequency and natural shedding frequency are observed: when the forcing frequency is lower than the natural shedding frequency an initial clockwise vortex is formed on the lower half of the cylinder when the cylinder is rotated in the counter-clockwise direction and a counter-clockwise vortex is formed on the upper half when the clockwise rotation starts. This leads to a non-synchronized vortex formation mode which cannot lead to the suppression of Karman vortex shedding (Akar, 2008).

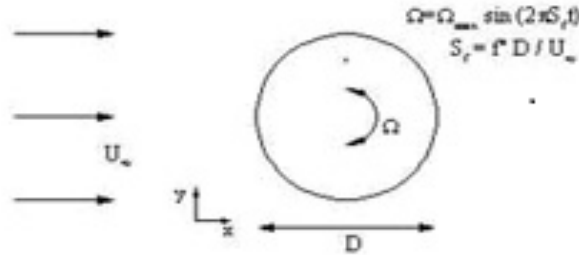


Figure 1.15. Configuration of the rotating circular cylinder

(http://flow.kaist.ac.kr/bbs/board.php?bo_table=databasinturbulen&wr_id=4)

1.2.2.4. Suction and Blowing

When wake flows controlled by suction and blowing, the flow's global stability state can be interpreted by considering the modification of the near wake profiles, and its implications on the flow's stability properties. There are two main effects of suction applied to cylinder base: the suction increases the magnitude of negative velocities, which acts in favor of instability increase in terms of local temporal growth rate increase. The second effect of suction is that it decreases the length of the recirculation zones and thus decreases the streamwise extent of absolute instability.

On the other hand, the effects of blowing are opposite to suction's effects: linear temporal growth rates in the near wake are expected to decrease, due to decreased back flow, extent of absolute instability is expected to increase. Therefore, in both cases, one has to rely phenomenological observations, to determine which action prevails for a given combination of the problem parameters (Akar, 2004).

1.2.2.5. Wake Heating

The control of vortex shedding downstream of a horizontal cylinder or a flat ribbon can be easily realized using heat input to the cylinder as seen in Figure 1.16. In an air flow, increase of the heat input to the cylinder or ribbon leads to a suppression of the vortex shedding while in a liquid flowing around a heated cylinder

or ribbon the effect is the opposite. However in liquids the range of heating conditions where buoyancy effects can be ignored is more limited.

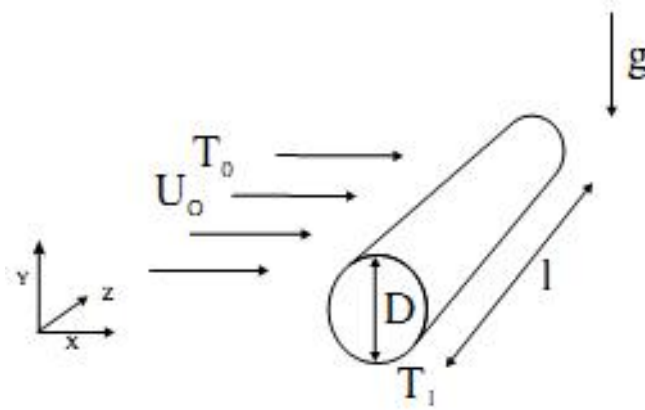


Figure 1.16. The problem configuration showing the horizontal cylinder exposed to a uniform cross flow. The gravity vector is perpendicular to the flow direction resulting in an asymmetry in the problem after inducing heat via the cylinder wall (Kieft et al., 2007)

1.2.2.6. Synthetic Jet

The synthetic jet generates vortices at the edge of an orifice by the time-periodic motion of a flexible diaphragm in a sealed cavity. The net mass flux of the synthetic jet is zero, but nonzero linear momentum is transferred to the flow (Choi, 2008). Working principle of this technique is using an oscillatory surface within a cavity to generate a jet from the flow that is being controlled without the need for mass injection. The jet shown in Figure 1.17 was generated with a piezoelectric diaphragm in a periodic manner. Flow enters and exits the cavity through an orifice. On the intake stroke, fluid is drawn into the cavity from the area surrounding the orifice. As this fluid is driven out of the cavity, a shear layer is formed between the expelled fluid and the surrounding fluid. This layer of vorticity rolls up to form a vortex ring. By the time the diaphragm begins to move away from the orifice to pull fluid back into the cavity, the vortex ring has moved far enough away that it is virtually unaffected. Thus a train of vortex rings is created by the actuator (Smith and Glezer, 1998).

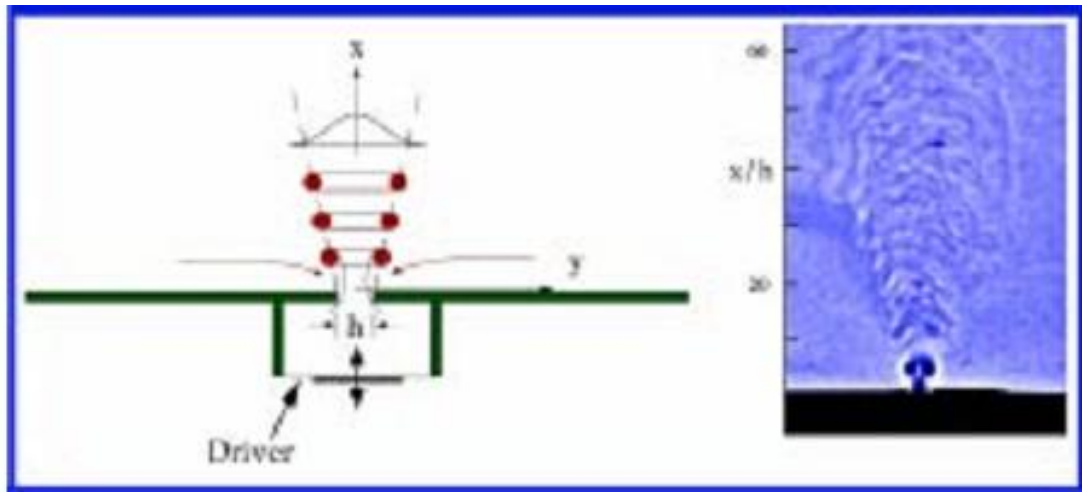


Figure 1.17. Synthetic jet actuator and flow patterns: (a) schematic and (b) schlieren flow visualization (Smith and Glezer, 1998)

1.3. Flow through Screens

The review of Laws and Livesey (1978) pointed out that the control of the velocity distribution of a fluid flow is a fundamental problem in engineering fluid mechanics. The possible consequences for the operation and efficiency of downstream components were emphasized in their work. Ability to control the flow is also necessary in component testing where, for results to be meaningful, test conditions must reproduce the composite flow situation. A screen may be used in both these operational modes to remove or create time-mean velocity nonuniformities, change the flow direction, and reduce or increase the scale and intensity of turbulence in a controlled manner. A screen may be thought of as any distributed resistance that effects a change in flow direction and a reduction in pressure. Common examples of screens for aerodynamic applications are arrays of parallel rods, honeycombs, perforated plates, and wire-gauze screens.

The wire- gauze screens (Figure 1.18), in other words the woven metal screens constructed from wires have been widely used in applications such as solar-receiving devices, high-efficiency heat exchangers, energy-storage units, and regenerators in Stirling cryocoolers, electronic coolers, catalytic reactors, and others. In filtering industry, woven metal screens are also widely used in the filtering of fluids due to their good resistance to chemical, thermal, and mechanical influences in

comparison with textile and paper made filters. Other advantages of woven metal screens include easy arrangement high permeability and relatively small deviation of the pore size from the mean value.

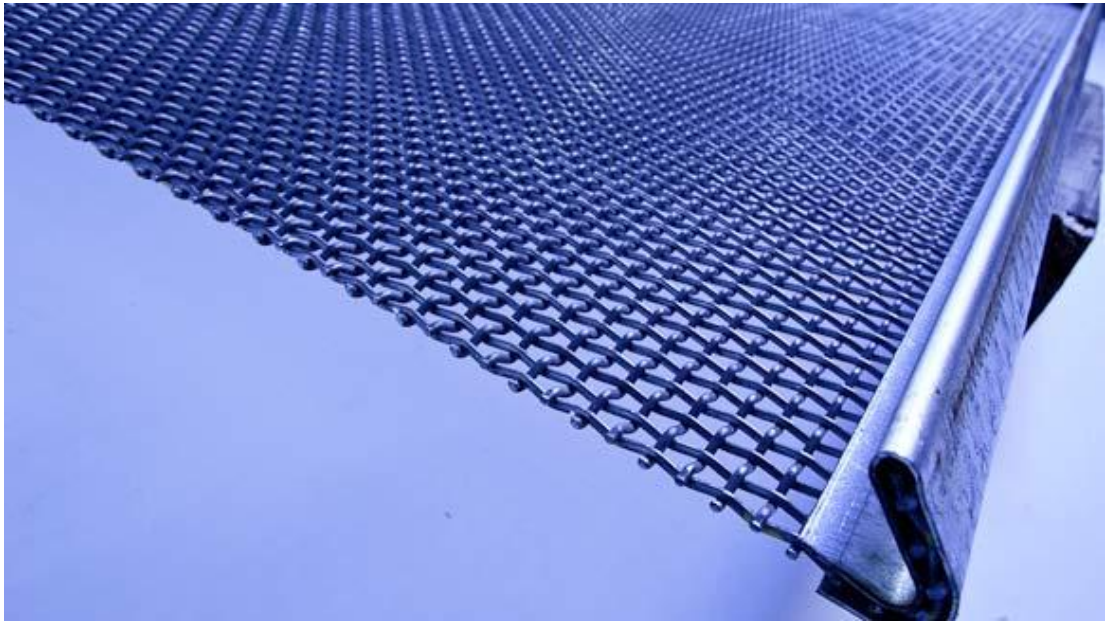


Figure 1.18. A picture of woven metal screen
(<http://samscreen.com/product/woven-wire-trommel-screens>)

Although woven metal screens offer the aforementioned merits, information on hydraulic resistance of woven metal screens is required when using these types of materials in any systems. Woven metal screens are usually classified by weaving types; there are four common types: the plain weaves, twill weaves, fourdrinier weaves, and dutch weaves.

- Ø *Plain weaves*: This is the most popular wire cloths weave. Each weft wire passes alternately over and under each warp wire and each warp wire passes alternately over and under each weft wire. Warp and weft wire diameters are generally the same.
- Ø *Twill weaves*: Each weft wire alternately passes over two, then under two successive warp wires and each warp wire passes alternately over two and under two successive weft wires, in a staggered arrangement.

Ø *Fourdrinier weaves*: This kind of weave is a mixture of plain and twill weaves and is also called semi-twill weaves.

Ø *Dutch weaves*: The warp wires remain straight; the weft wires are woven to lie as close as possible against each other in a linen weave forming a dense strong material with small, irregular and twisting passageways that appear triangular when diagonally viewing the weave.

The schematic diagram of these four types of woven metal screens is shown in Figure 1.19 (Wu et. al, 2005).

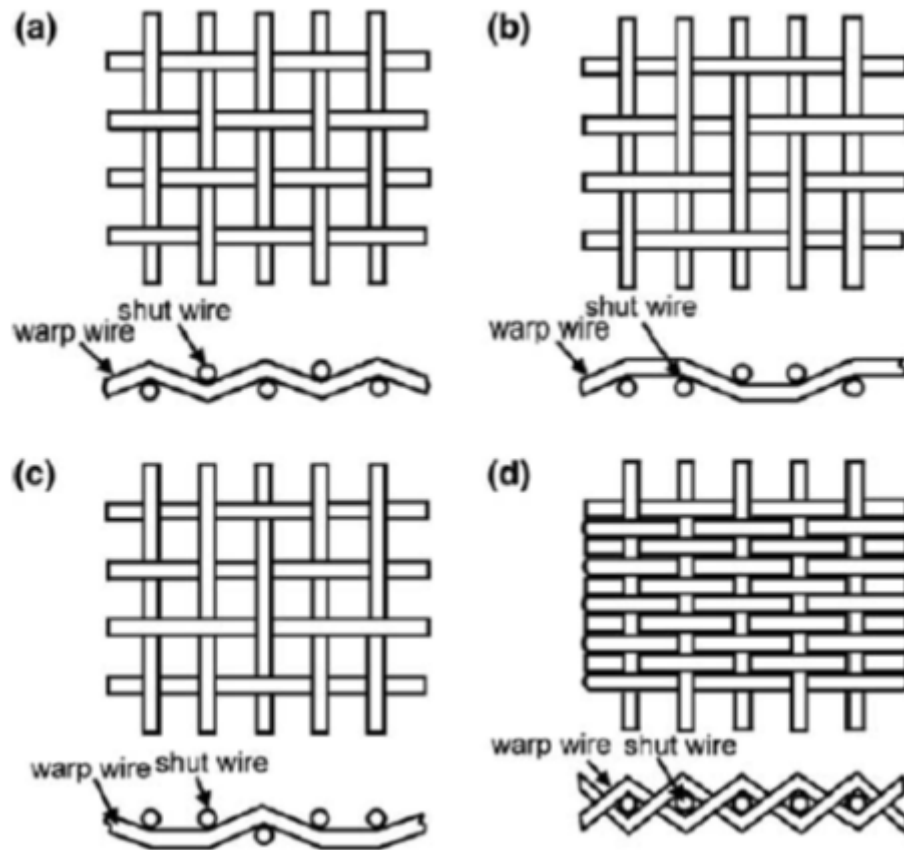


Figure 1.19. Schematic diagram of four common types of woven metal screens: (a) plain weaves, (b) twill weaves, (c) fourdrinier weaves, and (d) dutch weaves (Wu et. al, 2005)

1.4. Present Study

The goal of this work is to use the woven screens as a control element on suppressing the vortex shedding and vortex-induced vibrations downstream of a circular cylinder. The woven screens, constructed from chrome-nickel wires are called as permeable plates and they are attached on the midspan of circular cylinder. Despite being used for a wide range of engineering and industrial applications, in this study the screens are being used as a flow control element to control the unsteady flow around bluff bodies which might be an alternative flow control method within the context of vortex shedding suppression.

2. PRELIMINARY WORK

2.1. Flow around Bluff Bodies and Vortex- Induced Vibrations

The physics of the flow around bluff bodies was introduced in the first section of this study and it is important to review the works have been performed in order to analyze the details of the vortical structure around bluff bodies. Bluff body wake flows have direct engineering significance. The alternate shedding of vortices in the near wake, in the classical vortex street configuration, leads to large fluctuating pressure forces in a direction transverse to the flow and may cause structural vibrations, acoustic noise, or resonance, which in some cases can trigger failure. Recent interest in the far wake concerns the characteristic “signature” that remains after a body passes through a fluid (Williamson, 1996).

Williamson (1996) presented a comprehensive review on vortex dynamics in the cylinder wake. He introduced the complexity of bluff body wakes which involve the interactions of three shear layers in the same problem, namely a boundary layer, a separating free shear layer, and a wake. He addressed the study of Roshko (1993), by implying “the problem of bluff body flow remains almost entirely in the empirical, descriptive realm of knowledge”, although our knowledge of this flow is extensive. In fact, the recent surge of activity on wakes over the past decade from experiment, direct numerical simulation, and analysis has yielded a wealth of new understandings. In the case of the circular cylinder wake alone, there have been literally hundreds of papers, in part due to its engineering significance, and in part due to the tempting simplicity in setting up such an arrangement in an experimental or computational laboratory. Rather than attempt here to describe the vortex dynamics in both unseparated (for example, splitter-plate flows) and separated wakes for all the different types of 3-D and nominally 2-D body shapes, an overview is presented of the vortex dynamics phenomena in the wake of a circular cylinder, over a wide range of Reynolds numbers. New techniques in experiment, using laser-induced fluorescence and PIV (Particle-Image-Velocimetry), are vigorously being applied to wakes, but interestingly, several of the new discoveries have come from

careful use of classical methods. There is no question that strides forward in understanding of the wake problem are being made possible by ongoing three-dimensional direct numerical simulations, as well as by the surprisingly successful use of analytical modeling in these flows, and by secondary stability analyses.

Matsumoto (1999) described the various types of vortex generation and the related response characteristics of bluff bodies. He introduced the symmetrical vortex shedding, which is enhanced by the longitudinally fluctuating flow for 2-D rectangular cylinders with a 0.5 side ratio, and one-shear layer related vortices, which are generated on the side surfaces of flat 2-D rectangular cylinders and many bridge girder box sections by the stimulation of body motion or applied sound. Furthermore, he also discussed a peculiar 3-D vortex, the “axial vortex”, which is formed in near wake of inclined cables and then over restricted velocity ranges.

Williamson and Govardhan (2004) summarized the fundamental results and discoveries concerning vortex-induced vibration (VIV), that have been made over the last two decades, many of which are related to the push to explore very low mass and damping, and to new computational and experimental techniques that were hitherto not available. They brought together new concepts and phenomena generic to VIV systems, and pay special attention to the vortex dynamics and energy transfer that give rise to modes of vibration, the importance of mass and damping, the concept of a critical mass, the relationship between force and vorticity, and the concept of “effective elasticity,” among other points. They also presented new vortex wake modes, generally in the framework of a map of vortex modes compiled from forced vibration studies, some of which cause free vibration.

Ozgoren et al. (2011) performed an experimental investigation of flow structures downstream of a circular cylinder and sphere immersed in a free-stream flow for $Re=5000$ and 10000 using qualitative and quantitative flow visualization techniques. They stated that the concentration of small scale vortices is more dominant in the wake of the sphere than that of the cylinder.

Bearman (2011) presented a selective review of recent research on vortex-induced vibrations of isolated circular cylinders and the flow and vibration of circular cylinders in a tandem arrangement. In his study, the influence of Reynolds

number on the response of isolated cylinders was presented and recent developments using forced vibration were discussed. He concluded that the response of a cylinder free to respond in the in-line and transverse directions is contrasted with that of a cylinder responding in only one direction. He discussed the interference between two circular cylinders and prominence given to the case of cylinders in a tandem arrangement.

Other reviews including vortex shedding and induced vibrations are reported by Parkinson (1974), Sarpkaya (1979), Bearman (1984), Sarpkaya (2004), Gabbai and Benaroya (2005).

2.2. Flow Control Techniques

Every et al. (1982) stated in their review that the flow can be controlled by means of an alteration of the structural profile in order to affect the separation of the boundary layer to prevent the occurrence of vortex shedding along the length of the structure or alter the formation of the wake vortices and their interactions. Within this context, there are two types of flow control techniques to suppress or eliminate the vortex shedding and vortex induced vibrations, namely active and passive control methods in which energy is given to the fluid for the active methods while no power input is required for the passive ones. There are numerous investigations in the literature based on active and passive flow control. A good review by Choi et al. (2008) presents control methods for flow over a bluff body such as a circular cylinder, a 2D bluff body with a blunt trailing edge, and a sphere. They introduced recent major achievements in bluff body based flow controls such as 3D forcing, active feedback control, control based on local and global instability, and control with a synthetic jet. They also classified the controls as boundary-layer controls and direct-wake modifications and discuss important features associated with these controls.

2.2.1. Active Flow Control

Griffin and Votaw (1972) concluded that, when the cylinder vibrates transversely to the oncoming flow, the vortex frequency adjusts itself to the forcing frequency. An increase in the relative amplitude, A/D , where A is the vibration amplitude and D is the cylinder diameter, induces an increase in the drag and a reduction in the longitudinal distance of the vortices until they disappear completely at some threshold amplitude.

Smith and Stansby (1988) considered the locking-on of vortex shedding due to cross-stream vibration of the circular cylinder in uniform and shear flows. They showed that, in both flows, the vortex shedding frequency locked onto the cylinder frequency and onto sub-multiples of the cylinder frequency.

Baz and Kim (1993) made attempts to control the vibration of a cylinder in a cross-flow. In their study, surface-bonded piezo-ceramic patches were activated by a closed-loop control system to exert a force on the cylinder under resonance, that is, the vortex shedding frequency coincided with the natural frequency of the cylinder. The control increased the flow structure system damping and subsequently reduced the vortex-induced vibration.

The active control of vortex shedding using the internal acoustic excitation was first introduced by Hsiao and Shyu (1991). They investigated the influence of acoustic excitation on the aerodynamic performance of a cylinder, where the acoustic excitation was introduced internally to the flow over a circular cylinder through a slit with a loud speaker.

Ffowcs et al. (1989) were able to demonstrate, with active control of vortex shedding behind a cylinder with a diameter of 0.6 cm at a speed of 1 m/s and corresponding Reynolds number of 400.

Roussopoulos (1993) concluded that the feedback control was possible only at a Reynolds number 20% higher than at the critical Reynolds number. He claimed that Ffowcs Williams' active control of vortex shedding of around transitional Reynolds number at about 400 was really an active probe when affected, and prevented a second sensor from detecting a vortex shedding. He attempted to show

this interference of probes was actually claimed to be control of vortex shedding as claimed by them. He visualized the flow in a water channel and tried to show that there was no control of vortex shedding.

Tao et al. (1996) provided visualization photos between Reynolds numbers 48.5 and 51. This range is 18-24% above the onset of Reynolds number at the shedding frequency of about 1.05-1.1 Hz. At $x/d=10$ and 30 locations a complete elimination of vortex shedding is achieved, and flow-streak lines are reduced to almost a straight line in the cylinder wake.

Thereafter and Keles (1996) demonstrated experimental details of the transition to turbulence in the wake of a circular cylinder at $Re_D=170$. In those findings by Keles (1996) the data in the wake of cylinder first time, provides a clear picture of the feedback transition control. The distribution of the amplitude of velocity fluctuation's control was achieved for transitional at about $Re_D=170$.

The effect of externally imposed sound on the flow over a single cylinder has been reported by Blevins (1985), who concluded that sound levels greater than 130 dB resulted in a reduction in the small variation of the vortex-shedding frequency.

Hsiao and Shyu (1991) investigated the effect of acoustic excitation introduced internally to a circular cylinder through a slit to improve the aerodynamic performance of the cylinder in a uniform flow. They emitted the acoustic disturbances from a speaker system through the slit to the flow over the cylinder with sound pressure levels as high as 130 dB. The dimension of the slit was 1mm wide and 80mm long, while the cylinder diameter was 60mm with the length 200 mm, so that the aspect ratio of the cylinder was as small as 3.3. The drag was evaluated from wake velocity measurements and was found to be reduced by up to 40% at the optimum condition of the forcing frequency and slit location at a Reynolds number 2×10^4 . However, the measurement of pressure distributions did not indicate any drag reduction, which suggests that further research should be conducted to confirm the drag reduction by the acoustic excitation.

Fujisawa and Takeda (2003) investigated the mean fluid forces on a circular cylinder under acoustic excitation control by measuring the time-averaged pressure distributions over the cylinder. The result indicates that the mean drag is reduced up

to 30% at Reynolds number 9000. They also measured the distributions of mean and periodic velocities around the cylinder using a standard PIV system and showed that the near-wake velocity profile was considerably modified by the acoustic excitation control.

Lee and Lee (2006) investigated the flow characteristics of the wake behind a rotationally oscillating circular cylinder for and achieve good results. The cylinder wake was found to show five flow regimes according to the forcing conditions, with each regime having distinct vortex shedding characteristics and wake structure. Forcing the cylinder to undergo rotational oscillatory motion was found to be effective for controlling the wake flow. The maximum drag reduction was 32.82% in the low-frequency transition regime at $F_R = 0.8$ and 43.7% in the lock-on regime at $\theta_A = 60^\circ$ respectively. Thus, provided the forcing parameters are chosen appropriately, the rotational oscillation of a cylinder can be a promising method for controlling the drag acting on the cylinder. Importantly for practical applications, this approach is effective even at large Reynolds numbers.

Muddada and Patnaik (2009) numerically investigated the flow over a circular cylinder with the aid of a control algorithm using multiple-feedback sensors and actuators. They found that the rate of steady state injection depends on the proportionality coefficient, i.e. higher the rate of injection, the faster it will reach steady state and viceversa. They also demonstrated that energy expenditure can be further optimized through nimble control by enabling periodic switch on and off capability.

Jukes and Choi (2009) experimentally investigated the possibility of controlling the global flow field in the near wake of a circular cylinder at $Re = 6500$ using high-speed Particle Image Velocimetry. Surface plasma actuators were mounted at strategic locations around the cylinder (both fore and aft of the separation point) and used for flow control by producing a body force close to the wall. They found that the plasma can significantly alter the vortex shedding in the wake of the cylinder, with effectiveness depending upon the actuator location and forcing frequency. They also observed that most dramatic effects were observed when the plasma was located very close to the natural laminar separation point.

Chen and Aubry (2005) applied two different control techniques as distributed and localized Lorentz force. With the use of a symmetric and static Lorentz force over the entire surface of the cylinder, the vortex shedding behind the cylinder weakens and eventually disappears completely when the Lorentz force is sufficiently large. The localized Lorentz force along the rear surface of the cylinder was also used to control the vortex shedding behind the cylinder. Their numerical results showed that the efficiency of the localized Lorentz force in controlling the flow is to that of the Lorentz force distributed over the whole surface. When the interaction parameter (N) is about $N \geq 3$, vortex shedding completely disappears, and the total drag coefficient decreases as the Lorentz force increases due to the predominant role played by the pressure on the total drag coefficient.

Same authors apply a similar experiment with a close-loop control mechanism and the results presented correspond to the well-documented resonance response and are in good agreement with previous research results. The closed-loop control algorithm targeting zero drag coefficient, $C_d = 0$, which was developed through the introduction of Lorentz forces for the controlling of vortex shedding from a circular cylinder, was applied successfully to suppress the cylinder vibration in cross-flow direction both at $Re = 100$ and 200 .

Kieft and Rindt (2007) presented an investigation on the vortex formation and shedding process behind a heated cylinder which is exposed to a cold cross flow. The Reynolds number is chosen to be 75 while the Grashof number is varied between 0 and 5000 (resulting in a variation from forced to mixed convection). The numerical results of their study showed that the addition of heat disturbs the vortex formation process. The vortices shed from the upper half of the cylinder become stronger for increasing heat input. Therefore, the shedding process at the upper half of the cylinder becomes more effective compared with the process at the lower half. Consequently, the vortices shed from the upper half of the cylinder have a higher vorticity extreme and a higher temperature. The results show that the difference in effectiveness is mainly caused by a decreasing effect of strain rate during the formation of an upper vortex. This change in strain rate is caused by a change in flow pattern around the cylinder for increasing Grashof number. For higher heat input

more fluid flows underneath the cylinder, resulting in weaker shear layers at the upper part of the cylinder.

Konstantinidis and Balabani (2007) experimentally studied on the symmetric mode in the wake of a circular cylinder induced by periodic perturbations imposed on the in-flow velocity. They examined the wake field by PIV and LDV for Reynolds numbers about 1200 and for a range of perturbation frequencies between three and four times the natural shedding frequency of the unperturbed wake. In this range, they found that a strong competition between symmetric and antisymmetric vortex shedding occurs for the perturbation amplitudes employed. Their results show that symmetric formation of twin vortices occurs close to the cylinder synchronized with the oscillatory component of the flow. The symmetric mode rapidly breaks down and gives rise to an antisymmetric arrangement of vortex structures further downstream. The downstream wake may or may not be phase-locked to the imposed oscillation. The number of cycles for which the symmetric vortices persist in the near wake is a probabilistic function of the perturbation frequency and amplitude. From that study, they also showed that symmetric shedding is associated with positive energy transfer from the fluid to the cylinder due to the fluctuating drag.

Based on the study of a phenomenological model of FIV, Wu et al. (2007) experimentally investigated the high-frequency perturbation effects on the performance of suppressing flow-induced vibration (FIV). They found the possibility to suppress both structural vibration and wake vortex by introducing a perturbation input at a high-frequency range well-exceeding resonant frequency of structure.

Nosenchuck and Brown (1993) measured a reduction of skin friction of up to 90% by means of chessboard like arranged electrodes/magnets to induce a wall-normal Lorentz force in a flat plate boundary layer. The injection of an electrolyte generated a non-uniform electrical conductivity. Spanwise oscillating Lorentz-force was successfully used to reduce the skin friction in a turbulent boundary layer. A considerable scatter of results was found even on very similar arrangements. This could result from inadequate calculations of the Lorentz force in numerics or difficulties with established measurement techniques.

A nonlinear adjoint-based optimal control approach of cylinder wake by electromagnetic force is investigated numerically by Zhang et al. (2010). They constituted a cost functional representing the balance of the regulated quantities with different weights and interaction parameter N (Lorentz force), where the regulated quantities related with flow and force are taken as targets of regulation and the Lorentz force, (as interaction parameter N), is taken as a control input. They observed that under the action of optimal $N(t)$, the flow separation is suppressed fully, so that the oscillations of drag and lift are suppressed and the total drag coefficient decreases dramatically. For the different regulations, the control effects have some differences due to the different values of optimal inputs corresponding to the different adjoint flow fields.

2.2.2. Passive Flow Control

The scope of this study is to control the unsteady vortical flow structure downstream of a circular cylinder by passive means using attached permeable plates. Therefore the literature of passive flow control methods are widely researched and presented in this section of thesis.

A selective review of various aerodynamic passive control measures which could be adopted to reduce vortex induced vibrations (most general type of flow-induced vibration) is presented by Kumar et al. (2008). They introduced recent patents on this topic and concluding with notes on the current and future developments expected in the passive control of VIV. As mentioned by this study, there are various types of passive control technique in literature which are presented in next sections.

2.2.2.1. Splitter Plate

For a Reynolds number of 1.45×10^4 , (Roshko, 1954.) was notably one of the first to report periodic vortex formation suppression using a circular cylinder connected with a splitter plate. He also showed that, by positioning this splitter plate

at different fixed distances in the fully developed cylinder-wake, the Strouhal number St diminishes as the distance between the cylinder and the plate increases up to a limit position ($G \gg 2.7D$). He also observed that a splitter plate of chord $5D$, attached to the cylinder, results in suppression of shedding. A plate of chord $1D$ does not inhibit formation of vortices but changes the shedding frequency slightly.

Roshko (1955) observed flow changes at $Re_D = 1.45 \times 10^4$ when splitter plates were inserted behind a cylinder. When the plate was attached on the cylinder base and the length of the splitter plate (l) was larger than $5D$, the vortex shedding disappeared, and thus drag was significantly reduced. On the other hand, he also observed flow changes as the plate of $l=d$ moved downstream. When the splitter plate was located downstream less than $4d$, the Strouhal number decreased and the base pressure increased as the splitter plate moved downstream. However, when the splitter plate was located farther downstream, the flow field did not show much difference from the natural vortex shedding.

Bearman (1965) investigated the flow behind a two-dimensional model with a blunt trailing edge and fitted with splitter plates at Reynolds numbers between 140000 and 256000. He measured base pressure and shedding frequency and showed that the existence of a splitter plate increases the base pressure significantly (thereby reducing the pressure drag), even for very short plate length. These studies suggest that the effect of the splitter plate and base bleed is to delay the upper and lower separated shear layers interacting with each other and hence they increase the formation length (position at which the vortices are fully formed or, in a time-average sense, the length of the mean recirculation region). As a result, vortex shedding disappears for a connected splitter plate length L_{sp} equal to $2D$.

Gerrard (1966) studied a mechanism of vortex generation behind a cylinder. He conducted experiments at $Re_D = 2 \times 10^4$ to study the vortex size, frequency, and effect of the splitter plate attached on the cylinder, effects of free-stream turbulence. From that study, he showed that the Strouhal number decreased when the length of the splitter plate was smaller than d , but it increased for $D < L < 2D$. He also showed that the swirling formation zone is under a strong elongation in the presence of a SP

and that two modes of swirling formation could appear according to the length of the separating element and its distance from the model.

Unal and Rockwell (1987) divided the wake region into two separate regions: the pre-vortex formation region and the post-vortex formation region. They found that the pressure on the splitter plate was suddenly changed when the plate moved from the pre-vortex formation regime to the post-vortex formation regime.

Vortex shedding from bluff bodies with splitter plates was experimentally investigated by Nakamura (1996). It was shown that vortex shedding from bluff bodies with extended splitter plates is characterized by the impinging shear layer instability, where the Strouhal number in terms of splitter plate length increases stepwise with increasing splitter plate length.

Recently, Kwon and Choi (1995) investigated the effect of the splitter plate on laminar vortex shedding behind a circular cylinder at Reynolds numbers of 80-160. They showed that for $Re_D \geq 120$ the shedding frequency changed similarly as just described, whereas for $Re_D \leq 100$ the shedding frequency monotonically decreased as the plate length increased. They attributed the increased shedding frequency at $L > D$ in the case of $Re_D \geq 120$ to a generation of the secondary vortex at the tip of the splitter plate. This secondary vortex near the tip of the splitter plate may also be a strong source of the flow-induced noise.

Cimbala and Garg (1991) investigated the effect of the attached splitter plate on the flow characteristics behind a fixed cylinder and a freely rotating cylinder, respectively. For the case of a freely rotating cylinder, the flow field with the plate of $L > 2D$ was similar to that of the fixed cylinder with an attached plate. However, when $L < 2D$, the flow field was similar to that without the splitter plate. They also showed for the case of a fixed cylinder that the Strouhal number increased for $D < L < 2D$ and it decreased as the length of the plate further increased.

An experimental investigation was carried out by Mansingh and Oosthuizen, (1990) to study the effect of a splitter plate on the two-dimensional wake of a bluff body with fixed separation points. This experimental study has shown that the characteristics of the wake downstream of a bluff body with fixed separation points can be considerably altered by placing a splitter plate on the wake centerline

downstream of the bluff body. The effect of splitter plates on the wake flow characteristics of a rectangular cylinder with a splitter plate shows a relatively small change in Reynolds number.

Miau et al. (1993) carried out measurements of instantaneous pressure fluctuations on a trapezoidal cross-sectional cylinder. Their results indicate that the low frequency variations embedded in the vortex-shedding process can be successfully suppressed by insertion of a splitter plate whose length is twice the maximum width of the trapezoidal cylinder.

Unal and Rockwell (1988) carried out a similar study with a plate that is detached from the cylinder. They found that by placing the plate appropriately, compared to a cylinder with no control, it is possible to obtain more than double the vortex formation length. For low Re the vortex shedding is completely eliminated while for higher Re the large scale shedding is suppressed but the shear layer instability remains.

Ozono (1999) studied the effect of a plate of length $1D$ kept at various locations, including those away from the wake centerline, downstream of the cylinder for $6.7 \times 10^3 < Re_D < 2.5 \times 10^4$. Beyond a certain critical gap between the plate and the cylinder, for a certain range, the Strouhal number exceeds the natural one. This is because of a flow mode in which the shear layer separated from the cylinder is forced to interact with the splitter plate.

Roshko (1993) controlled the flow around a circular cylinder by attaching a splitter plate behind the cylinder and investigated the suppression of vortex shedding. When the gap between the cylinder and splitter plate was large, the effect of the splitter plate became strong. The parameter that had the greatest effect on the vortex shedding was the gap width rather than the length of splitter plate.

Zhou et al. (2005) conducted a numerical study on suppression of fluid forces on a square cylinder in cross-flow controlled by a flat plate. They found that the existence of the control plate changes the flow features in front of the square cylinder totally. The drag acting on the square cylinder and the total drag acting on the 2 bodies (the square cylinder together with the control plate) are normally reduced by the control plate. The maximum reduction of the drag acting on the square cylinder

and the optimum position of the control plate are given for a particular control plate height. They also revealed that the amplitude of the fluctuating lift on the square cylinder is well suppressed if the control plate is used. The fluctuating lift can be completely suppressed if the control plate with a certain height is placed at an appropriate position.

Akıllı et al. (2005) investigated the effect of splitter plate on the suppression of vortex shedding in shallow water using the Particle Image Velocimetry (PIV) technique. Splitter plates with a length of 50 mm, which is equal to the diameter of the cylinder (D), were used during the experiments. Gap distance between the base of the cylinder and the leading edge of the splitter plate ranged from 0 to $2D$. Velocity vector field and corresponding vorticity contours and streamline topology and Reynolds stress concentrations were used to explain the characteristics of the flow. The splitter plates having different thicknesses presented the same influence on the flow characteristics. The splitter plate has a substantial effect on suppression of the vortex shedding for the gap ratio between 0 and $1.75D$. In this gap ratios, shear layers extend up to the trailing edge of the splitter plate preventing the entrainment of the free- stream flow into the base region. For $G/D=1.75$, the normalized Reynolds stress has a peak value of 0.0158 which is approximately 8.5 times smaller than the concentrations occurring for a bare cylinder. After passing the splitter plate, the flow starts to oscillate with smaller frequency compared to the natural frequency of the bare cylinder. When the splitter plate was located at $2D$ location, no effect of the splitter plate was observed.

Hwang et al. (2007) numerically studied the drag reduction on a circular cylinder using dual detached splitter plates. They placed two splitter plates with the same length as the cylinder diameter (d) along the horizontal centerline; one is upstream of the cylinder and the other is in the near-wake region, respectively. They found that drag varies with the two gap ratios; it has the minimum value at a certain set of gap ratios for each Reynolds number. The upstream splitter plate reduces the stagnation pressure by friction, while the downstream one increases the base pressure by suppressing vortex shedding. This combined effect causes a significant drag reduction on the cylinder.

Akıllı et al. (2008) investigated experimentally passive control of vortex shedding behind a circular cylinder by splitter plates of various lengths attached on the cylinder base in shallow water flow ($Re=6300$). At this study the length of the splitter plate was varied from $L/D=0.2$ to $L/D=2.4$ in order to see the effect of the splitter plate length on the flow characteristics. Instantaneous and time-averaged flow data clearly indicate that the length of the splitter plate has a substantial effect on the flow characteristics. They found the flow characteristics in the wake region of the circular cylinder sharply change up to the splitter plate length of $L/D=1.0$. and above this plate length, small changes occur in the flow characteristics.

Turki (2008) numerically investigated the passive control of flow by introducing a splitter plate behind a square cylinder inside a horizontal channel at Reynolds number range of $110 \leq Re \leq 200$ and for a fixed blockage ratio of $\beta = h/H = 1/4$ and a fixed position of the square cylinder. In examining the instantaneous flow fields, he found that the critical length of the splitter plate L_c , characterizing the disappearance of vortex shedding, can be correlated by a linear relationship: $L_c = 0.0373 Re - 3.4373$. He also analyzed the effect of the splitter plate length and its location in the wake region on the Strouhal number, and the drag and lift coefficients. He came on the conclusion that, by varying the splitter plate length, the Strouhal number shows two different behaviors for the Reynolds number investigated. And finally he observed that in the variation of the Sr and the C_d with the splitter plate location G and for $Re = 150$ and 200 , a sudden increase in those quantities was noticed at the optimal location of $G= 3.27$.

Shukla et al. (2009) studied the problem of a hinged-splitter plate in the wake of a circular cylinder where the splitter plate can rotate about the hinge at the base of the cylinder due to the unsteady fluid forces acting on it, and hence the communication between the two sides of the wake is not totally disrupted as in the rigid splitter plate case. They investigated that this problem in the limit where the stiffness and internal damping associated with the hinge are negligible, and the mass ratio of the splitter plate is small. It is concluded that the splitter plate oscillations increase with Reynolds numbers at low values of Re , and are found to reach a saturation amplitude level at higher Re , $Re > 40000$.

Assi et al. (2009) achieved suppression of cross-flow and in-line VIV of a circular cylinder, with resulting drag coefficients less than that for a fixed plain cylinder using two-dimensional control plates. He demonstrated how vortex-induced vibration can be practically eliminated by using free-to-rotate, two-dimensional control plates and further conclusion is that these devices achieve VIV suppression with drag reduction. He found the largest drag reduction to have a drag coefficient equal to about 60% of that for a plain, fixed cylinder over the Reynolds number range of the experiments, up to 30 000.

Assi et al. (2010) studied on a pair of circular cylinders to investigate the effectiveness of pivoting parallel plates as wake-induced vibration suppressors. Measurements of amplitude of vibration and average drag are presented for a circular cylinder, free to respond in the cross-flow direction, with mass ratio of 2 and a damping level of 0.7% of critical damping. Reduced velocities were up to nearly 30, with associated Reynolds numbers up to 2.3×10^4 and the results presented are for a centre-to-centre separation of cylinders of 4 diameters. They showed how vortex-induced vibration and wake-induced vibration of the downstream cylinder of a tandem pair can be practically eliminated by using free to rotate parallel plates. They also presented the results for a single splitter plate and helical strakes for comparison, however they were found not to be effective in suppressing wake-induced vibration.

Stappenbelt (2010) investigated the proposed utility of splitter-plate wake stabilization as a passive control mechanism for vortex-induced vibration (VIV) mitigation for low aspect ratio cylinders. His experiments of Stationary cylinder show the effectiveness of splitter plates in reducing lift and drag coefficients, C_D and C_L for a cylinder in uniform flow. On the other hand, rigid attached splitter plates are capable of entirely eliminating vortex shedding in fixed cylinder investigations.

Gozmen et al. (2013) conducted an experimental work on the control of vortical flow of circular cylinder using a splitter plate having different lengths and heights. They found that as the ratio of h_p/h_w increases, the effect of the splitter plate on the suppression of the vortex shedding increases. Flow characteristics and the examination of spectra indicated that Karman vortex shedding is attenuated pronouncedly for the cases of $L/D \geq 1$ and $h_p/h_w \geq 0.75$. The transverse Reynolds

normal stress is more effective on the attenuation of turbulent kinetic energy than the streamwise Reynolds normal stress, $\langle u'u'/U^2 \rangle$. They also found that the value of peak transverse Reynolds normal stress, $\langle u'u'/U^2 \rangle$ is reduced to 90% of that of the bare cylinder at most.

2.2.2.2. Geometrical Modifications and Introducing Add-on Devices on the Body

Bearman and Tombazis (1993) and Tombazis and Bearman (1997) investigated the three-dimensional features of the wake behind a blunt-based model with a wavy trailing edge at a Reynolds number of 40000. For a two-dimensional body at high Reynolds numbers, it was noted that vortex dislocations appeared in the wake apparently randomly in time and spanwise position. However, they observed that the introduction of a spanwise waviness at the trailing edge fixed the positions of these vortex dislocations along the span of the body. They found that by increasing the wave steepness, defined as the ratio of peak-to-peak wave height divided by the wavelength, the base pressure was increased which resulted in a drag reduction. The maximum drag reduction of 34% at a Reynolds number of 40 000 occurred for a wave steepness of 0.14 (the non-dimensional wavelength, defined as the wavelength divided by the base height, was equal to 3.5). Based on these facts, they concluded that encouraging the formation of dislocations in the wake reduces the drag.

Most previous studies have investigated the flow structure over riblet surfaces placed on a flat plate. Walsh (1983) investigated drag reduction of longitudinal ribs of rectangular, triangular, and transversely curved shapes. He reported that the drag reduction was associated with a decrease of momentum thickness and turbulent velocity fluctuations and depended mainly on the riblet configuration such as the height h , spacing s , and shape. He suggested that a riblet with sharp peaks and large valley curvature is one of the optimum riblet shapes for drag reduction.

Cardell (1993) used permeable splitter plates spanning in the wake center plane. He defined the permeability as the pressure drop across the plates and the relationship between permeability and plate solidity was determined for a set of plates constructed from woven wire mesh, permitting unambiguous characterization

of the splitter plates by the solidity. The effects of different solidities on the flow in the near wake were investigated using smoke wire flow visualization, hot-wire anemometry and measurements of the mean pressure at the cylinder surface, and the results were related to cylinder flow without a splitter plate. He concluded that the introduction of low solidity splitter plates do not change the basic near wake structure, and that sufficiently high solidity uncouples the large-scale wake instability from the body, with primary vortex formation occurring downstream of the separation bubble due to instability of the wake profile. He pointed out that the presence of the splitter plates enhances shear layer development, and it is found that for all solidities the instability amplifies a broad frequency band without discrete spectral features.

Naudascher and Rockwell (1994) used surface modifications to alter the loading and structural vibration of a circular cylinder. They modified the surfaces of cylinders in numerous ways, for example by applying straight and helical wires, staggered wires, and collars and rings.

Bearman and Owen (1997) experimentally investigated the influence of spanwise waviness of the separation lines on the flow around common bluff forms. By performing the experiments in wind tunnel, they concluded that the drag of thin plates and rectangular cross-section bodies can be substantially reduced by the introduction of spanwise waves into the flow separation lines. They also indicated that drag reductions of up to at least 30% are achieved and for wave steepnesses in excess of between 0.06 and 0.09, vortex shedding is completely suppressed.

Owen et al. (2001) extended this concept to a circular cylinder by changing the straight axis of the cylinder into a sinusoidal axis along the spanwise direction and by spirally attaching hemispherical bumps on the cylinder surface producing an effect similar to that observed for the flow over a rectangular cylinder. Also the dependence on the angle of incidence could be removed by attaching bumps in a spiral pattern. Hemispherical bumps reduce vortex-induced oscillations (up to 47%) and drag (up to 25%) in cylindrical structures.

In their experimental study, Ahmed and Bays-Muchmore (1992) measured the surface-pressure distributions of a set of wavy cylinders with different axial

wavelengths based on the mean cylinder diameter, D . The Re in their experiment was 20 000 and the mean static pressure data was determined using linear interpolation. Their results showed that the separated flow structures near the geometric nodes were distinctly asymmetric for a large fraction of time, and the sectional drag coefficients at the geometric nodes are greater than at the geometric saddle.

Darekar and Sherwin (2001) performed numerical simulations to investigate the flow past a square cylinder with a wavy stagnation face at low Reynolds numbers and showed that the Karman vortex shedding is suppressed into a steady and symmetric structure by the spanwise waviness. They argued that vortex-shedding suppression is caused by the distortion of 2D shear layers due to geometric disturbances because the three-dimensionally redistributed shear layers become less susceptible to rolling up into a Karman vortex street. They also found that the optimal wavelength of spanwise waviness is close to the spanwise wavelength of mode A instability.

Lam et al. (2003) used a LDV system to measure the velocity profiles of the wake behind a sinusoidal cylinder in the range of $Re=3,000-9,000$. They concluded that the long vortex formation length of wavy cylinder should bring about drag reduction and suppression of Vortex induced vibration.

Bearman and Brankovic (2003) investigated the vortex-induced vibration of a plain circular cylinder and cylinders fitted with strakes and bumps. They carried out the experiments in water with mass ratios in a range from about 1.5 to 5.5 and combined mass and damping values between about 2×10^{-3} and 7×10^{-3} by taking the Reynolds number within the range of 10^3 to 10^4 . They indicated that although the wakes of fixed cylinders with strakes or bumps show little or no evidence of regular vortex shedding it was found to establish itself across a range of reduced velocity when the mass and damping parameters are sufficiently small. However, the amplitudes of vibration were found to be less than those for a plain cylinder. They also observed that the modes of shedding found for an oscillating plain cylinder are present for a responding cylinder fitted with vortex suppression devices.

Lam et al. (2004) experimentally investigated the effects of surface waviness of wavy (varicose) cylinders on mean drag and fluctuating lift reduction. They

presented the measurements on mean and fluctuating forces induced by a cross-flow over three wavy cylinders and the circumferential pressure distributions at various spanwise positions. They concluded that the mean drag coefficients of the wavy cylinders are less than that of a corresponding circular cylinder; a drag reduction of up to 20% was obtained. They also found that the root-mean-square fluctuating lift coefficients of the wavy cylinders are much lower than those of a circular cylinder. However, the spectral analysis of both hot-wire and load cell signals show that the Strouhal numbers of the wavy cylinders are essentially the same as a corresponding circular cylinder.

Bearman and Owen (2005) have found that providing smoothly curved protuberances on to a cylindrical structure (ellipsoidal shape) would significantly reduce drag and vortex-induced vibrations. They further suggested that, height of protuberances shall be in the range $0.25D$ to $0.5D$. It is to be noted that, the protuberances are radially distributed (in the range 30° to 90°) so as to provide effectiveness in a wide range of flow direction. The inventors suggest that, more pronounced protuberances (greater height) are required when used in denser liquids to become effective in controlling VIV. The main advantage of providing protuberances is that it is simple and relatively inexpensive. But, application of protuberances would lead to loss of structural geometry which may not be acceptable in some situations. Practical applications of this invention include legs or risers of offshore platforms, underwater cables in sea currents, chimneys in wind flow and wind turbine support tower.

Yoon (2005) found that the tab located near the separation point reduces drag on a circular cylinder and attenuates the Karman vortex shedding in the wake, and the optimal spanwise spacing between the adjacent tabs is similar to that in Darekar and Sherwin (2001).

Park et al. (2006) proposed a small-size tab, mounted on part of the upper and lower trailing edges of a bluff body, to effectively attenuate vortex shedding and reduce drag. In their parametric study, they varied the height (l_y) and width (l_z) of the tab and the spanwise spacing between the adjacent tabs (λ), and showed that drag is decreased (or the base pressure is increased) by attaching this simple device to the

trailing edge. The optimal configuration of tabs produced an approximately 33% increase in the base pressure. Owing to the tabs, the vortices shed from the upper and lower trailing edges lost their 2D nature, and vortex dislocation occurred. The Karman vortex shedding completely disappeared right behind the bluff body but occurred weakly at farther downstream locations. Because the main mechanism of drag reduction by the tab is to introduce the spanwise phase mismatch in the vortex-shedding process and thus break the nominally 2D nature of Karman vortex shedding, this passive device should work for other 2D bluff bodies such as the circular cylinder.

A large eddy simulation of cross-flow around a sinusoidal wavy cylinder at $Re=3000$ is performed by Zou and Lin (2008) by introducing the load cell measurement for the validation test. In the study, they presented the mean flow field and the near wake flow structures and compared with those for a circular cylinder at the same Reynolds number. They found that the mean drag coefficient for the wavy cylinder is smaller than that for a corresponding circular cylinder due to the formation of a longer wake vortex generated by the wavy cylinder. The fluctuating lift coefficient of the wavy cylinder is also greatly reduced. This kind of wavy surface leads to the formation of 3-D free shear layers which are more stable than purely 2-D free shear layers. From that study, they also showed that such free shear layers only roll up into mature vortices at further downstream position and significantly modify the near wake structures and the pressure distributions around the wavy cylinder.

Baek and Karniadis (2009) numerically investigated the hydrodynamic effect of a slit in circular cylinders with the goal of finding a geometric modification that minimizes vortex-induced vibrations (VIV) without any energy consumption. A slit parallel to the incoming flow is introduced and found to be very effective in suppressing VIV by either weakening or detuning vortex shedding. Through a series of simulations, they found the optimal size of the slit to suppress VIV at Reynolds number of 500; this size is smaller at a higher $Re = 1000$. They confirmed the effectiveness of the slit in suppressing VIV over a wide range of values of the reduced velocity. They also reported that the jet flow through a slit changes the

stability in the wake of cylinder by inducing two small pockets of absolute instability at the back of the cylinder followed by bands of convectively and absolutely unstable regions.

Turhal and Cuhadaroglu (2010) performed experiments to investigate some aerodynamic parameters of the flow around perforated-surface square, horizontal, and diagonal cylinders. The surface pressures of the perforated surface square cylinders were measured for the cases of injections through the top, rear, top-rear, and all surfaces separately at Re values of 10.000, 16.000, and 24.000. The C_D and St were determined based on pressure coefficient measurements and vortex shedding frequency measurements, respectively. Their experimental results showed that the injection through the various surfaces of a square cylinder has some effects on the aerodynamic characteristics of the square cylinder, depending on the position of the perforated surface. They concluded that C_D of a horizontal square cylinder can be reduced by injection through all the surfaces, and for a diagonal square cylinder; it also can be reduced by injection through the rear surfaces. They also revealed that the vortex shedding frequency of the flow around a horizontal square cylinder can also be decreased by high-momentum injection through the rear surface.

Xiao et al. (2011) presented a numerical study on the hydrodynamic performance of undulation NACA0012 foil in the near wake of D-section cylinder. They conducted the computations using unsteady incompressible Navier–Stokes equations with a moving adaptive mesh based on laminar flow. Investigations are focused on the effect of distance ratio between foil tip and centre of cylinder ($L/D2.0$) on the thrust/drag performance of foil and cylinder at various foil undulation frequency (St). They found that, foil thrust coefficient (C_t) increases considerably with the appearance of cylinder and an optimal distance exists at which C_t reaches maxima. According to their findings they concluded that insertion of an undulation foil can significantly lead to the drag reduction indicating that undulating foil could work efficiently as a passive vortex control device for cylinder drag reduction.

The numerical study of Xiao et al. (2012) examined the potential of cylinder wake vortex control using a downstream undulating foil. They showed that the

proper placement of foil, with its chord length equal to the upstream cylinder diameter, can result in a suppression of cylinder vortex shedding and an improvement in downstream foil propulsion. The detailed flow field analysis revealed that, such benefit is strongly linked to the interaction between the cylinder wake vortex and the reversed Karman vortex street, which is shedding continuously at the foil leading edge associated with foil undulation locomotion.

Within the context of geometric modifications, passive control of the flow by shrouded cylinders has been firstly studied by Price (1956). In his study, various porous shrouds were used to investigate their effects on suppression of the vortex-excited vibrations. He pointed out that the vibration suppression can be achieved by shrouded cylinders because of the delay in forming the normal Karman Street.

Every et al. (1982) reviewed the practical aspects of vortex excited structural vibrations in a marine environment with particular emphasis on methods of their suppression. In a part of their study, they focused on the effect of shrouds on the flow control, reported a comprehensive literature and stated that shrouds with the optimum dimension gave a 50% reduction in oscillations of a plain study on suppressing flow induced vibrations (Every and King, 1979).

Molin (1993) performed a numerical study about perforated shrouds used to reduce vortex- induced vibrations of cylinders. They compared their numerical results with the available experimental data and demonstrated that the loading on the shroud is reasonably well predicted and the load on the inner cylinder is found to be negative. However, he concluded that their proposed theory cannot be said to predict reliable values of the drag of shrouded cylinders since it fails to reproduce correctly the flow in the downstream part of the annular region, in-between the shroud and cylinder. A similar logic is being used in recent years called porous media which includes the idea of generating a permeable medium around a bluff body.

Zhao and Cheng (2010) established a numerical model for simulating the flow in and out of a porous media and concluded that lift reduction can be achieved by properly choosing the porous material.

Boorsma (2008) studied the effect of perforated fairings in order to investigate and optimize their effect on the noise control of landing gears. He

clarified the effects of porosity, β and location of perforation and found that perforated fairings applied to the simplified landing gear models reduce the low frequency noise introduced by the solid fairings.

Huang (2011) experimentally investigated the effects of triple- starting helical grooves on the drag of fixed circular cylinders and the vortex-induced vibration of elastically supported cylinders. A comparative approach which allows direct comparisons of the results was adopted where two cylinders of identical dimensions and physical properties with or without helical surface grooves were tested in exactly same experimental set-ups. In the elastically supported cylinder tests, the cylinders were attached to a vertically cantilevered supporting rod and towed in a towing tank. Both the in-line and cross-flow vibrations were permitted. He found that for the case investigated the helical grooves were effective in suppressing the vortex-induced cross-flow vibration amplitudes with the peak amplitude reduced by 64%. Drag reductions of up to 25% were also achieved in the sub-critical Reynolds number range tested in the study for the fixed cylinders.

Mazellier et al. (2011) studied the performances of an original passive control system based on a biomimetic approach by investigating the flow over a bluff body. Their control device consists of a couple of flaps made from the combination of a rigid plastic skeleton coated with a porous fabric mimicking the shaft and the vane of the bird's feathers, respectively. They found that by comparing both the uncontrolled and the controlled flow, a significant drag reduction ($\approx 22\%$ on average) has been obtained over a broad range of Reynolds numbers. This improvement is related to the increase of the base pressure in the controlled case. Their investigation of the mean flow revealed a noticeable modification of the flow topology at large scale in the vicinity of the controlled cylinder. They also stated that the study of the relative motion of both flaps highlights that their dynamics is sensitive to the Reynolds number. Furthermore, the analysis of the flow dynamics at large scale suggests a lock-in coupling between the flap motion and the vortex shedding.

A 2D numerical simulation of the flow around a circular cylinder is investigated by Fiabane et al. (2011) during the onset of unsteadiness within the range of Reynolds numbers between 50 and 400. They successfully approximated the

fluid force by a volume integral of a force density over a small flow domain surrounding the cylinder. They stated that the integration of the force density on both separated vortex layers and on the back-flow region gives two instantaneous contributions to the total force. For the mean drag they found that the back-flow contribution increases from almost 0% of the total drag at $Re = 50$ –20% of the total drag at $Re = 400$. They concluded that with regard to the force fluctuations, both regions contribute similarly to the lift oscillations, while only the back-flow region is responsible for the drag oscillations. They discussed the alternative comprehension of the fluid force origin and compared to that of the classical pressure/viscous formulation.

Kunze et al. (2011) presented experiments on separation control using flexible self-adaptive hairy-flaps. They investigated the wake-flow behind a circular cylinder without and with flexible hairy-flaps at the aft-part of the cylinder. Flow dynamics and hair motion were measured by particle image velocimetry and image processing in a range of Reynolds number $5000 < Re < 31000$. The experiments and POD analysis show that the hairy-flaps alter the natural vortex separation cycle in such a way that the vortices do not shed in a zigzag like arrangement as in the classical von Kármán vortex street but in line in a row with the cylinder wake axis. Thus, the wake-deficit is largely reduced. Furthermore, they have found that the flow fluctuations are considerably reduced about 42% in streamwise and 35% in transversal direction compared to the reference case without hairy-flaps, too.

Ozkan et al. (2012) carried out an experimental research directed for passively controlling flow around a circular cylinder vertically placed in uniform flow of water. The novelty of the study is the utilization of a screen (meshed controlling element), which has the streamlined shape, surrounding the cylinder. The velocity fields of near-wake downstream of the circular cylinder were experimentally determined by using the PIV technique. The turbulence statistics obtained from the velocity field measurements downstream of the cylinder were analyzed in detail. The results revealed that the existence of control element around the cylinder significantly suppressed the vortex shedding from the outer surface of the cylinder. As a result, the intensity of turbulence, turbulent kinetic energy, $\langle TKE \rangle$ magnitudes

and Reynolds shear stress, $\langle u'v'/U^2 \rangle$ were too low not only at the shear layers, but also they were diminished along both transverse and streamwise directions, compared to the bare cylinder case.

2.2.2.3. Use of a Small Control Cylinder

Bouak and Lemay (1998) made an application of passive control of the unsteady loads induced on a circular cylinder. For this purpose they developed an aerodynamic balance allowing the simultaneous measurement of fluctuating lift and drag forces. The balance, designed for a cylinder of 25.4 mm in diameter, was used in the subcritical regime ($40\,000 \geq Re \geq 15\,000$). They obtained 32% drag reduction by means of a small cylinder in tandem arrangement (stagger angle $\alpha=0^\circ$). Also they observed that at $\alpha=4.5^\circ$ a peak of mean lift coefficient and a 41% reduction of the rms lift fluctuations which leads to an instantaneous force vector which exhibits more steadiness both in angle and amplitude. Moreover, at this stagger angle, the energy of the lift fluctuations at the shedding frequency is reduced by 80% compared with the single cylinder case.

Strykowski and Sreenivasan (1990) have revealed that a small secondary cylinder placed in the near wake for vortex-shedding suppression may be associated with changes in absolute instability.

Tsutsui et al. (2002) investigated the control of flow around a circular cylinder in air-stream by a rod placed upstream of the circular cylinder. They stated that two flow patterns with and without vortex shedding from the rod occurred. The flow pattern changes depending on the rod diameter, its position and the Reynolds number. They founded the optimum conditions of the drag reduction as $d/D=0.25$; $L/D=1.75-2.0$, furthermore, they also concluded that vortices do not shed from the rod and the shear layer from the rod reattaches on the front face of the circular cylinder. The reduction of the total drag including the drag of the rod is 63% compared with that of a single cylinder.

Giannetti and Luchini (2003) and Hwang & Choi (2006) studied the effects of a small secondary control cylinder on the global and absolute instability in the wake,

respectively. They obtained regions in which the global or absolute instability is attenuated. These regions are the separated shear layer and the wake centerline.

Kuo et al. (2007) numerically studied the passive wake control behind a circular cylinder in uniform flow for Re_D ranging from 80 to 300. They used two small control cylinders, with diameter $d/D = 1/8$, placed at $x/D = 0.5$ and $y/D = \pm 6$ and the Reynolds number is varied from 80 to 300. Their mechanism, which can significantly reduce the fluctuating lift and the form drag on the main cylinder, is explored without completely suppressing the vortex street for $Re_D = 80\text{--}300$. They reported that, by this control, the vortex street behind the main cylinder is still maintained but the fluctuating lift and the form drag on the main cylinder decreases monotonously with increasing Reynolds number.

Passive flow control using two-dimensional hydrofoils to reduce vortex-induced vibrations (VIV) and drag on a cylinder of circular cross-section was studied by Galvao et al. (2008). They test the hypothesis that by using foils to bend the streamlines around the cylinder, and hence forcing the flow to approach potential flow-like patterns VIV and drag will be reduced. They showed that properly positioned foils can significantly alter the flow around a circular cylinder to eliminate VIV and reduce drag.

Kuo and Chen (2009) revealed the wake flow structures behind a circular cylinder manipulated by two small control cylinders and elucidate the primary mechanism that leads to a reduction of the lift and the pressure drag on the main cylinder. They found that when the control cylinders lie within $0.8 \leq X_C/D \leq 3.0$, the fluctuating lift on the main cylinder shows a 70–80% reduction without completely suppressing the vortex street.

Thiria et al. (2009) investigated the turbulent drag of a 2-D bluff body with a blunt trailing edge. Their study focused on the wake when the drag is reduced by 17.5% by means of a smaller cylinder located behind the bluff body. They showed that the bubble recirculation length and the pressure inside are increased by the presence of the control cylinder. They also found that the turbulent wake has less diffusion compared to that of the natural wake. The global mode envelope of the instability, measured with a hot wire anemometer, is found to be strongly inhibited

by the presence of the control cylinder. The maximum amplitudes of the envelope are shifted further downstream the recirculation bubble end and the Strouhal number is slightly increased by 4.3%. Phase averaging of the vorticity field showed that the circulation of the large-scale structures is not significantly affected.

2.3. Flow through Screens

The flow through screens is also an important issue in order to understand their effect on the flow control since they will be used throughout this study.

A comprehensive review by Laws and Livesey (1978) divides the flow through screens into 3 categories: (1) Investigations concerned principally with characterizing the flow properties of the screen, (2) investigations concerned with the effect of a screen on time-mean velocity distributions and (3) investigations concerned with the turbulence distribution downstream of gauze screens which is the combination of three distinct turbulence profiles. In their review the literature covering these three categories was discussed in details.

Harry et al. (1989) established a prediction model for the headloss through a screen that is placed in an open channel. They implied that for a given approach-flow condition, screen inclination, and screen characteristics (i.e., flow contraction and deflection caused by a screen); values of the headloss coefficient can be predicted by the model. They concluded that there is an optimal inclination angle to minimize the headloss. The predicted values of the headloss are found to be in good agreement with the measured values in their experiments.

Sahin (1988) investigated the mechanisms of screen flow and the associated flow resistance. He described the behavior of the flow emerging from perforated plates in relation to the measuring locations and geometry of the perforated plates. He concluded that in the case of using more than one screen as a flow control device, it is not advisable to install screens in close proximity to one another.

Armour and Cannon (2004) adopted a packed bed model to develop a general correlation applicable to the flow of Newtonian fluids through all types of woven metal screens. They used the main theoretical approaches to studying pressure drop

in packed systems by visualizing the screen as a collection of submerged objects with surface area to unit volume ratio a for laminar flow, and as a bundle of tubes of diameter D for turbulent flow. In the usual manner viscous and inertial energy losses are added to give an expression for the total pressure loss. They found that the rearrangement of the general equation to the form of a friction factor yields a unique definition of the Reynolds number for screens $N_{Re} = \rho u / \mu a^2 D$. In their study, the procedures were described for collection of pressure drop-velocity data for the flow of nitrogen and helium through plain square, full twill, fourdrinier, plain dutch, and twilled dutch weaves. The data were used to derive a viscous resistance coefficient $\alpha = 8.61$ and an inertial resistance coefficient $\beta = 0.52$. They concluded that the correlation successfully predicts pressure drop for a Reynolds number range of 0.1 to 1000, void fractions from 0.35 to 0.76, screen pore diameters from 5 to 550 μ , mesh sizes from 30 to 2400 wires/in, and surface area to unit volume ratios from 1,200 to 29,000 ft^{-1} .

O'Neill (2005) used the source panel method to investigate the singularity models of flow through and around screens. He identified the models that best represent the limited experimental data that is available and the need for more experimental data, in order to carry out a thorough assessment.

Wu et al. (2005) established an experimental setup to measure the pressure drop of flow through woven metal screens. Four woven metal screens with different porosities of the plain-square type were tested in their study with a porosity range of 0.834–0.919. Based on the measured pressure drops of the four woven metal screens, they developed an empirical equation of friction characteristic of plain-square-type woven metal screens. They found that in order to obtain good agreement between the fitted empirical equation and measured data, an empirical equation should be developed for each type of the woven metal screen. Their study developed five empirical equations respectively for the five types of metal screens, (plain square, plain dutch, fourdrinier, full twill, and twilled dutch types) of which experimental data were available in open literature and the present study.

Favier et al. (2009) have studied the passive control of flow separation using a fuzzy coating being inspired by the bird wings. To test and analyze their concept of

flow manipulation, they developed a numerical methodology and used to solve a fluid–structure interaction problem for $Re=200$. They found that aerodynamics performances quantified by the Strouhal number, the drag and the maximum lift in the laminar unsteady regime are modified by the presence of the coating. A set of parameters corresponding to a realistic coating (length of elements, porosity, rigidity) is found, yielding an average drag reduction of 15% and a decrease of lift fluctuations by about 40%, associated to a stabilization of the wake.

Teitel (2009) carried out a study to make a comparison between simulations of flow through a realistic woven screen and a porous slab that represented the realistic woven screen. He indicated that using the published relations for K and Y which are used in literature may lead to erroneous results for pressure drops through screens. In his study, an alternative way, basically similar to that used in previous studies, for calculating K and Y is presented; it led to good estimates of pressure drops through various porous media slabs that were used to simulate realistic screen geometries. He concluded that the Pressure drops through mono-filament woven screens can be determined by means of CFD simulations in which the realistic woven screen geometry is replaced with a porous medium. This approach reduces computation time considerably, since there is no need to use a very fine computational mesh to resolve the flow near the threads of the screen. Different turbulence models result in differing values of the pressure drop through a given screen; the differences may be as great as 50% when flow through a model of a realistic screen is simulated. The differences among the pressure drops yielded by the various turbulent-flow models were much larger when flow through realistic woven screens was simulated than when flow through a porous slab was simulated. They also concluded that for woven screens with mono-filament threads it is recommended to obtain K and Y by method I that is proposed in the present study. This method is essentially similar to the previously published method, but it treats each screen individually rather than seeking a global correlation that relates K and Y to a on the basis of tests of several screens with differing porosities. Attention should be given to exactly identifying the flow regime through the screen, and to verifying that the

Forchheimer equation is valid in that regime. Method I is of course not applicable to screens of irregular mesh or screens that are not made of mono-filament threads.

Teitel (2011) investigated the applicability of the Forchheimer equation to estimate pressure drops through woven screens. He validated the results of CFD simulations of flow through realistic models of woven screens by means of wind tunnel tests and by means of a published correlation for calculating pressure drops through such screens. Then he used CFD tool in conjunction with the wind tunnel tests and published correlation results to examine whether the coefficients in the Forchheimer equation are constant over a wide range of Reynolds numbers. He found that the values of the Forchheimer coefficients varied with Reynolds number, and that the changes in their values can be assumed to be relatively small and not very significant but only up to a Reynolds number, $Re_i \approx 130-160$. He concluded that presenting the Forchheimer coefficients as functions of only porosity may lead to erroneous predictions of pressure drops.

Middelstädt and Gerstmann (2013) numerically investigated the flow through metallic screens. A CAD model was created of two screens, the “Twilled Dutch 165x800” screen and the “Twilled Dutch 200x1400” screen, where the screen models were implemented in the commercial CFD Code ANSYS FLUENT. They performed the numerical calculations with varying screen Reynolds numbers in the range of 0,1 to 1000. They compared the results with experimental data and theories in the literature. The analysis of the numerical results showed further that at the screen Reynolds number 40, the flow separation takes place and the pressure drop increases rapidly. They stated that the flow becomes transient and the forming vortex need a long distance in the backflow to decay. The porosity was also found to be an important screen parameter, influencing the pressure drop and flow behavior. They calculated the occurring wall shear stress at the threads and analyzed for both screen types in dependence on the screen Reynolds number. Their numerical results showed that for low Reynolds numbers, in the laminar flow region, areas of high wall shear stress arise at chain threads. For high screen Reynolds numbers, in the turbulent flow region, these areas are blurred over the whole screen.

Bommisetty et al. (2013) carried out a numerical study to find the flow loss in screens. They used commercially available CFD code, ANSYS CFX 12.1 for numerical study by performing the simulations for both incompressible and compressible fluids. They concluded that the predicted discharge coefficient values of incompressible fluid matched with the available experimental results up to screen Reynolds number of 10. But for the turbulent region, predicted discharge coefficients are lesser compared to the experimental values. They also find out that the discharge coefficient has increased with increase in Reynolds number up to screen Reynolds number of 1000 and then constant discharge coefficient is obtained with further increase in Reynolds number. The constant discharge coefficient obtained from the simulation is 0.91 against the value of 1.4 in experiments. For compressible fluids, predicted discharge coefficients are in line with incompressible fluids till the density variation across the screen is about 20%. Considerable reduction (maximum of 50% reduction) is noticed in discharge coefficient with higher density change for compressible fluids. They also concluded that the density change of fluid in the screen is found to be stronger influencing parameter for flow loss in screens compared to the screen Reynolds number.

3. MATERIAL AND METHOD

3.1. Experimental Methods

Throughout this study, in order to analyze flow structure both flow visualization and Particle Image Velocimetry (PIV) techniques were used. Flow visualization experiments were performed to have a direct observation of the flow structure in the investigated flow field whereas the PIV experiments was used to measure the velocity vector field which is one of the best non- intrusive techniques within the velocity measurement systems. The velocity is determined by measuring the displacement of particles in a flow field that is illuminated by a laser sheet which will be explained in detail on further parts of this section.

Both PIV and dye visualization experiments were performed in a large-scale water channel (Figure 3.1) located in the Fluid Mechanics Laboratory at Çukurova University. The dimensions of the test section inside the water channel are as follows: a length of 8 m, width of 1 m, and a depth of 0.75 m. The test section is constructed of transparent Plexiglas with upstream and downstream fiberglass reservoirs. The water channel also has honeycomb screen arrangement, which is located at the entrance of contraction. These reservoirs and honeycomb screen arrangements are used to maintain the turbulence intensity in the test section less than 1%. The water flow speed was controlled by a centrifugal pump which can be adjusted by a frequency controller unit.

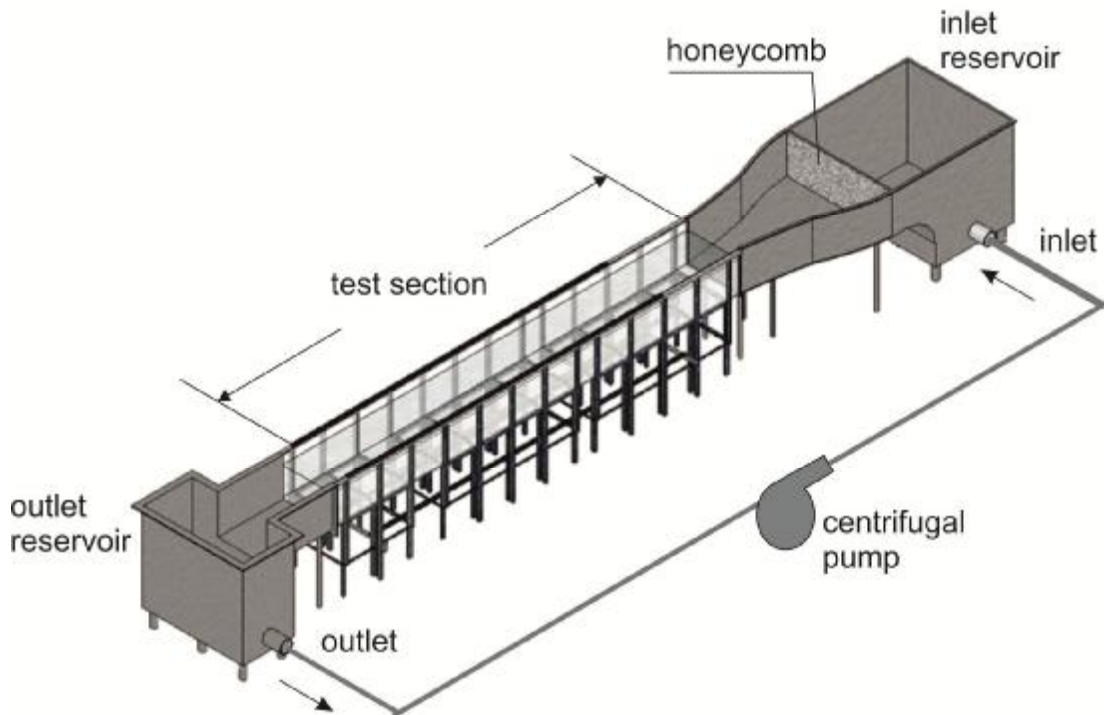


Figure 3.1. The water channel with its arrangements

3.1.1. Flow Visualization Technique

Flow visualization technique aims to visualize the flow structure qualitatively by spreading some amount of Rhodamine-B type dye that shines under the continuous laser light (Spectra Physics) in two-dimensional flow field. Dye comes from a container, which were located 1 m above the free surface of the water channel and goes through a plastic tube into the laser sheet available on the flow field. The evaluated flow visuals were recorded with a high-definition video camera (Sony HD-SR1) and instantaneous pictures were captured from the records by using the program of Sony, called Play Memories. The dye visualization technique offers no numerical analysis or data to use, but shows a brief demonstration of the flow that gives an idea about the flow field and the vortices. Furthermore it gives the chance of comparing the instantaneous quantitative measurements evaluated by PIV, with the qualitative ones. A brief demonstration of the dye visualization setup is shown in Figure 3.2 which shows the continuous laser unit, generated laser sheet and a picture taken from visualized flow field. The experiments were performed by closing all

light sources inside the laboratory to increase the brightness of dye which shines under the light of laser sheet.

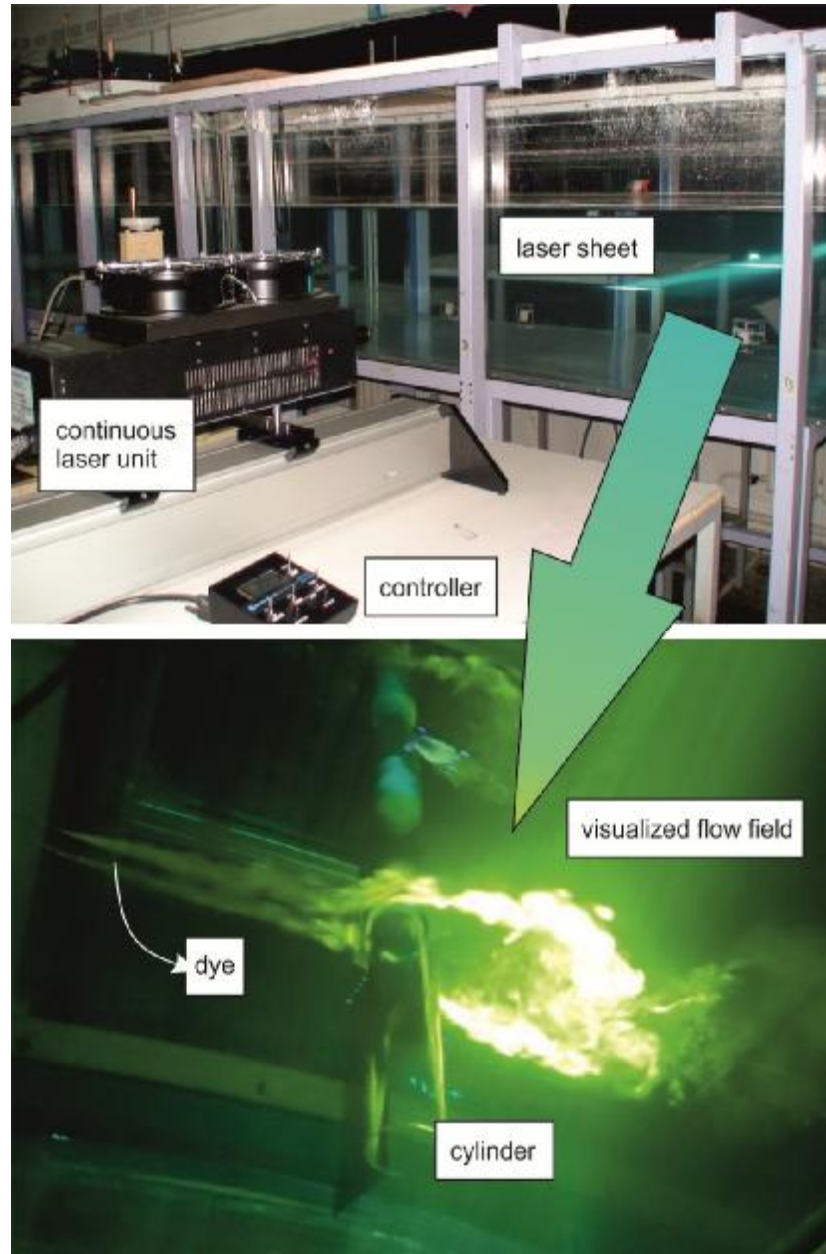


Figure 3.2. The apparatus for the flow visualization technique and a picture from visualized flow field

3.1.2. Particle Image Velocimetry (PIV) Technique

Particle image velocimetry (PIV) is a double-pulsed laser technique used to measure the instantaneous velocity distribution in a plane of flow by photographically determining the displacement of particles in the plane during a very short time interval. Unlike methods like hot-wire anemometry and LDV that measure velocity at a point, PIV provides velocity values simultaneously throughout an entire cross section, and thus it is a whole-field technique. PIV combines the accuracy of LDV with the capability of flow visualization and provides instantaneous flow field mapping. A PIV system can be viewed as a camera that can take a snapshot of velocity distribution at any desired plane in a flow. Ordinary flow visualization gives a qualitative picture of the details of flow. PIV also provides an accurate quantitative description of various flow quantities such as the velocity field, and thus the capability to analyze the flow numerically using the velocity data provided. Because of its whole-field capability, PIV is also used to validate computational fluid dynamics (CFD) codes.

The PIV technique has been used since the mid-1980s, and its use and capabilities have grown in recent years with improvements in frame grabber and charge-coupled device (CCD) camera technologies. The accuracy, flexibility, and versatility of PIV systems with their ability to capture whole-field images with submicrosecond exposure time have made them extremely valuable tools in the study of supersonic flows, explosions, flame propagation, bubble growth and collapse, turbulence, and unsteady flow (Çengel and Cimbala, 2007).

The main components of a PIV measurement system are presented schematically in Figure 3.3 as; a computer, a video camera synchronized with pulsed Nd: YAG laser unit by use of a synchronizer and a pulser. The figure presents an application of analyzing the flame stabilization.

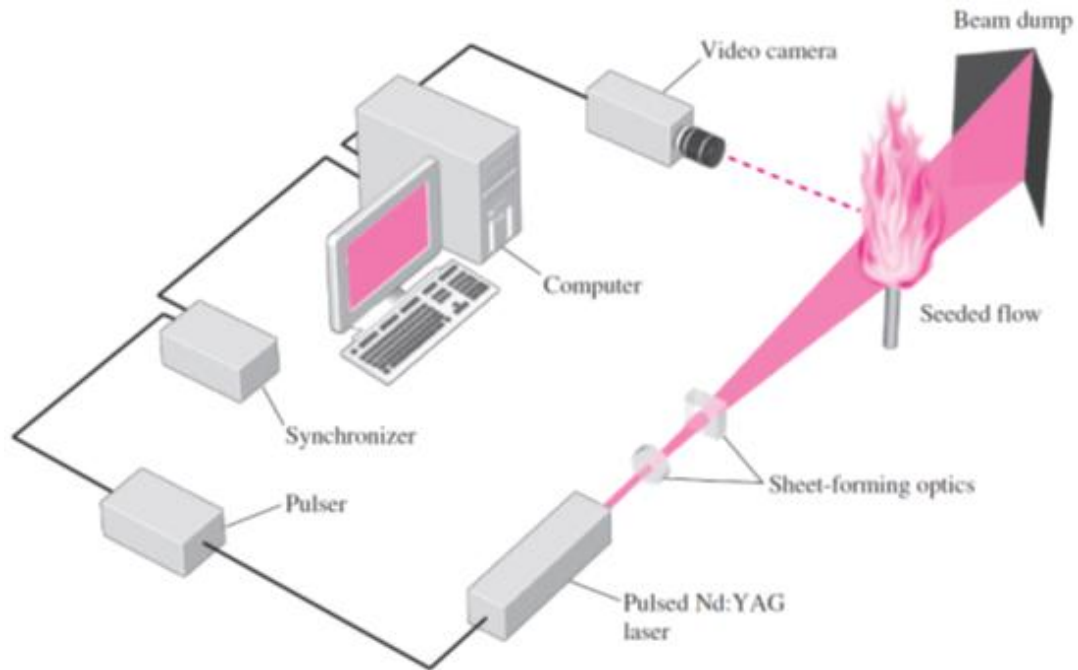


Fig. 3.3. A PIV system to study flame stabilization (www.tsi.com)

The theory of PIV was introduced by Adrian (1988) who presented the first experimental implementations. At the stage, due to hardware limitations, a single photographic frame was multiply exposed and analyzed using an auto-correlation technique. However, improved speed of photographic recording soon allowed images to be captured on separate frames for analysis by cross-correlation (Keane et al. 1992). The introduction of digital camera technology to PIV enabled the direct recording of particle images (Willert et al. 1991), at the expense of reduced resolution, resulting in the development of digital PIV (Westerweel, 1997). As well as these hardware advances, many new algorithms have been developed in the last decade, increasing the accuracy and speed of PIV analysis. One key advantage of the PIV technique over other conventional techniques such as LDV or hotwire anemometry is ability to obtain instantaneous flow-field information is important for probing the structure of turbulent flow fields (Westerweel et al. 1996; Meinhart and Adrian 1995).

3.1.2.1. Principles of PIV

The PIV technique for velocity measurement consists of two main steps: visualization and image processing. The first step is to seed the flow with suitable particles in order to trace the fluid motion. Then a pulse of laser light sheet illuminates a thin slice of the flow field at the desired plane, and the positions of particles in that plane are determined by detecting the light scattered by particles on a digital video or photographic camera positioned at right angles to the light sheet (Frame 1 in Figure 3.4). After a very short time period Δt (typically in μs), the particles are illuminated again by a second pulse of laser light sheet, and their new positions are recorded (Frame 2 in Figure 3.4). Using the information on these two superimposed camera images, the particle displacement, Δx are determined for all particles, and the magnitude of velocity of the particle in the plane of the laser light sheet is determined from $\Delta x/\Delta t$. The direction of motion of the particles is also determined from the two positions so that two components of velocity in the plane are calculated. The built-in algorithms of PIV systems determine the velocities at thousands of area elements called interrogation regions throughout the entire plane and display the velocity field on the computer monitor in any desired form (Çengel and Cimbala, 2007).

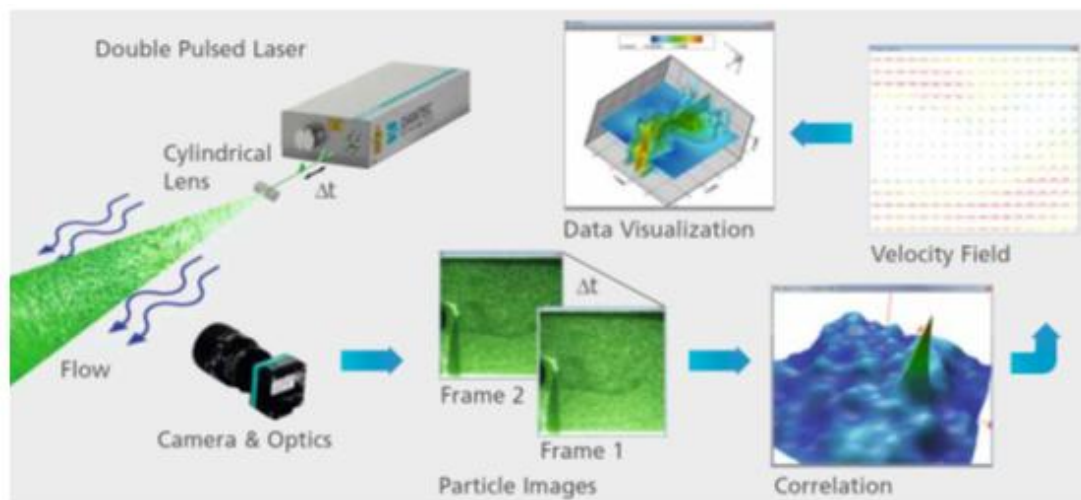


Figure 3.4. The evaluation of velocity vector field by PIV (www.dantecmt.com)

From the basic principles the PIV emerges the following main topics:

- Ø Seeding
- Ø Illumination
- Ø Cameras
- Ø Synchronization
- Ø Correlation
- Ø Validation and further analysis

3.1.2.1.(1). Seeding

The particles are added to the flow in PIV and known as seeding. Different types of seeding particles are used depending on the nature of the flow for PIV experiments. The silver coated hollow glass sphere particles (Figure 3.5) are most commonly used for PIV measurements of water flow conditions.

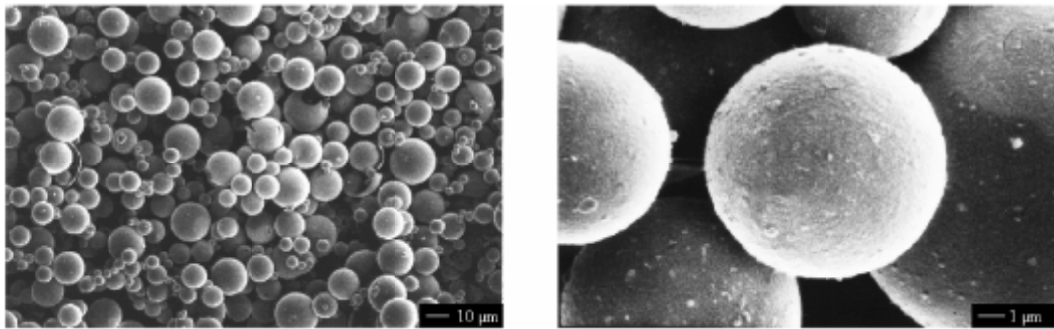


Figure 3.5. Micrographs of silver coated hollow glass spheres: $\times 500$ and $\times 5000$ (Raffel et al., 2007)

The type of seeding particle is chosen to follow the flow, and in order to detect their movement, an area of the flow field is illuminated by a light-sheet. The light-sheet, which is generated by a laser and a system of optical components, is not continuous/permanent, but pulsed to produce a stroboscopic effect, freezing the movement of the seeding particles.

An important parameter to be considered within the context of seeding is the Stokes number which provides a measure of the particles' ability to accurately track

the flow field, and the limiting Stokes number also depends on the flow geometry and the presence of significant accelerations or decelerations. Stokes number is defined as the ratio of the stopping distance of a particle to a characteristic dimension of the obstacle, or

$$St = \frac{t_p \cdot U_{\infty}}{d_{particle}}$$

where t_p is the relaxation time of the particle, U_{∞} is the fluid velocity of the flow well away from the obstacle and $d_{particle}$ is the diameter of the particle. The particle relaxation time is given by: $\tau_p = \frac{\rho_{particle} d_{particle}^2}{18 \rho_{fluid} \cdot \nu_{fluid}}$ where $\rho_{particle}$ is the particle density and $d_{particle}$ is particle diameter. For $St \gg 1$, particles will continue in a straight line as the fluid turns around the obstacle therefore impacting on the obstacle. For $St \ll 1$, particles will follow the fluid streamlines closely.

3.1.2.1.(2). Illumination

For the illumination, it is preferable to use a laser, since the laser beam is easy to form into a sheet by a cylindrical lens. A pulsed laser is to prefer, since one obtains a high light energy during a very short time interval (typically 5 ns for a YAG-laser), which means that the particle images will be practically frozen even for high velocities (> 100 m/s). The repetition rate of a YAG-laser is typically 10-30 Hz, which is too low except for very low velocities (< 1 cm/s). One therefore needs two lasers to get full freedom in terms of time separation between the pulses. Special PIV YAG-lasers are available that combine two laser cavities with a common beam outlet.

In the PIV technique, there are three principal methods of producing a light-sheet; the first one is the two pulsing beams from a double-cavity Q-switched Nd:YAG laser which are combined and formed into pulsing light-sheets using optics, the second one as continuous wave beam from an argon-ion laser which is chopped using a shutter and the resulting pulsing beam is expanded into a light-sheet using

optics, and finally the third method is a continuous wave beam from an argon-ion laser which is scanned by a rotating polygon mirror to form a pseudo-light sheet (with a shutter chopping the beam to ensure the PIV camera is only exposed to an integer number of sweeps of the beam).

The main selection criterion is the illumination energy density. Mie scattering theory can be used to predict the amount of light needed to obtain an image of an isolated seeding particle. However, multiple scattering phenomena mean the actual energy budget cannot be accurately predicted and so empirical assessments are required. In many cases, pulsed Nd:YAG lasers are required to provide an illumination source of sufficient energy to obtain PIV images. Nd:YAG PIV lasers are now being more commonly applied to the study of water flows since it offers the possibility of having the same energy budget no matter the value of the time between pulses. Furthermore, together with a cross-correlation camera, it is possible to optimise seeding before considering other data acquisition and processing parameters.

3.1.2.1.(3). Cameras (Image capturing)

In the developed series of PIV instrumentation, PIV cameras are one of the four classification groups since the recording system utilized to capture PIV data is an important element in the PIV measurement process. CCD based recording systems were chosen since on-line image maps are a pre-requisite to near real-time vector map processing. In PIV technique, the purpose of the camera is to capture the initial and final positions of seeding particles in the flow field and from these particle positions the displacement vector can be processed. The optical recording arrangement is shown in Figure 3.6. The camera is placed at right angles to the light sheet. Seeding particles scatter light from the first pulse of the light-sheet and this scattered light is detected by the camera ideally as a bright signal on a dark background. Thus, the camera image map has sampled the initial positions of the seeding particles. The light-sheet is switched off and the seeding particles are transported by the movement of the flow field and, in cross-correlation the camera

frame is advanced. The second pulse of the light-sheet is fired and the seeding particles scatter light which is imaged onto the camera detector so the final positions of the seeding particles are sampled.

The term CCD stands for charge-coupled device. A CCD camera comprises an array of detectors called pixels. Each pixel is a MOS capacitor, being charged by converting incident photons of light into electrons, like in a photodiode. The cells are isolated from each other by potential wells, created by the doping of the silicon chip and by applying voltages to a grid of transparent metallic electrodes deposited on the CCD surface. Light falling on a pixel is thus converted into an electronic charge. The charge falling on the individual pixels is transformed to a voltage during read-out of the CCD chip and the value of the voltage is seen as a grey scale distribution on the PIV image map. Ideally, images should have a high charge i.e. appear white and the background noise level of the CCD chip should be low i.e. appearing dark (Flow Manager Software and Introduction to PIV Instrumentation).

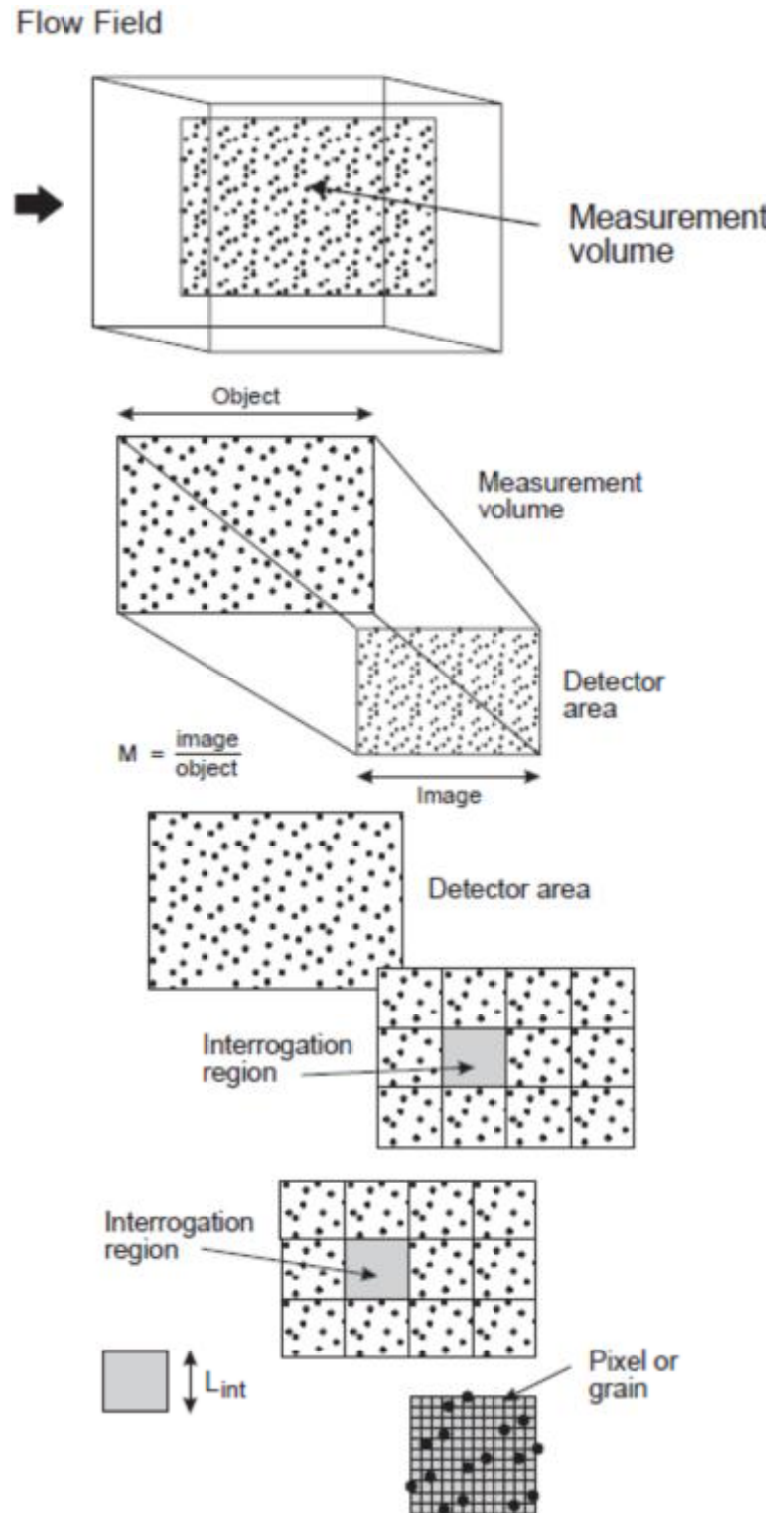


Figure 3.6. Role of cameras in the PIV measurement process

3.1.2.1.(4). Synchronization

Since repeatable and accurate timing signals are necessary for the collection of reliable particle image velocimetry results, the timing and synchronization requirements can be complex. Dual-pulsed lasers can require up to four signals each to control the pair of synchronized pulses necessary for PIV (Figure 3.7).

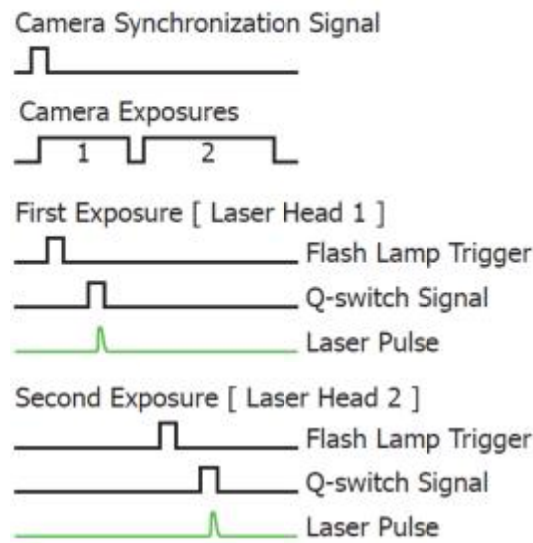


Figure 3.7. Timing signals required for full control of a PIV system (Technical notes of Particle Image Velocimetry, Berkeley Nucleonics Corporation)

While internal timing can be used to reduce the number of signals required, for complete control of each laser head, a flash lamp trigger to charge the laser and a Q-switch signal to trigger laser pulse emission are needed. By tuning these four separate signals, both the time separation between images and the relative beam intensity are adjustable. A fifth synchronized signal coordinates the camera exposure with the individual laser pulses. If the timing signals activating the laser and camera are not synchronized, the velocity field may be irrecoverable. Even small errors in the timing will lead to errors in the measured velocity field due to its direct reliance on the time separation between pulses. To maximize the ease of setup and reliability, using a single device to manage the timing pulses required can ensure precise and accurate particle image velocimetry results.

3.1.2.1.(5). Correlation

Since the introduction of the first PIV image evaluation methods, alternative analysis algorithms have been developed as well as error correction and post-processing procedures designed to improve speed and accuracy of the PIV method. However, the classical PIV analysis method is still the most frequently used and forms the basis of many other algorithms (Figure 3.8).

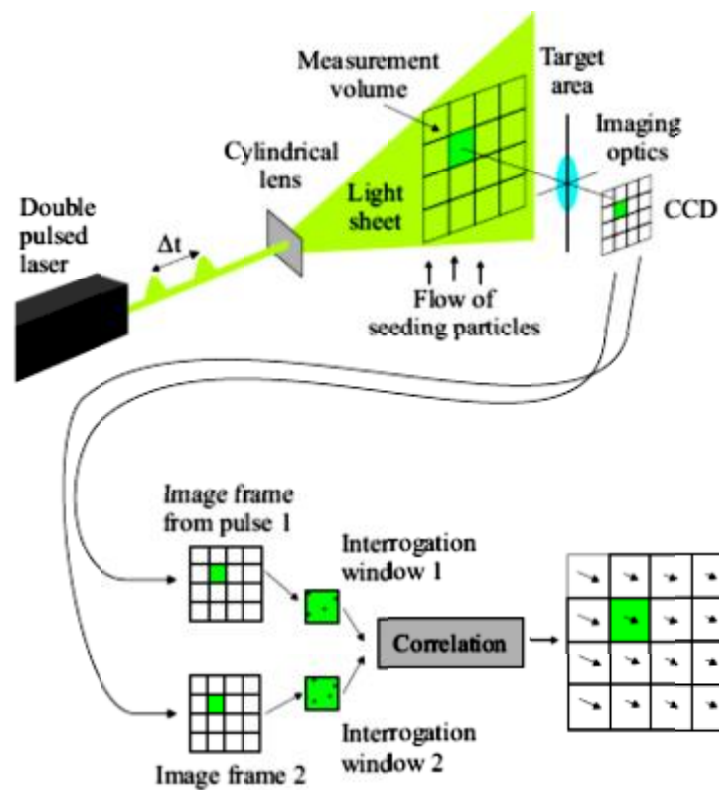


Figure 3.8. PIV overview (Schiwietz, T., Westermann, R., 2004)

The heart of the PIV analysis is the correlation of regions of the input images (known as interrogation areas) with each other to determine the displacement vector of the flow in that part. Knowing the time interval between the image captures enables a velocity vector to be calculated from the displacement vector. The correlation technique can be used for a single frame multiple exposed (auto-correlation) or multiple frames singly exposed (cross-correlation). To speed up the convolution process, correlation of each pair of interrogation areas is carried out in

Fourier space. After interrogating the images in this way and generating the vector map, post-processing is carried out to validate the data and to improve the vector map resolution and accuracy. Using this vector map, vorticity and Reynolds stress contours and streamline topology can be obtained.

Basically, the architecture of the CCD array influences whether the particular camera can be used for auto- or cross-correlation image capture. Historically, autocorrelation has been the most commonly used method of image capture in Particle Image Velocimetry initially because photographic film was used as an image capturing medium. Thereafter, CCD camera frames could not be advanced quickly enough to record images with a short time interval between them. Now, cameras are available which can have a very short time between double frames so cross-correlation image capture is now possible. Thus, cross-correlation is technically feasible as well as being theoretically more robust and inherently containing velocity direction information (Flow Manager Software and Introduction to PIV Instrumentation).

3.1.2.1.(5).(a). Cross- Correlation Process

In PIV two sequential image maps are sub sampled. The resolution of the sub sampling is defined as an interrogation area and the interrogation areas form a regular array. Within these interrogation area samples, an average spatial shift of seeding particles may be observed from one sample to its counterpart in the second camera image, provided a flow is present in the illuminated plane. This spatial shift may be described quite simply with a linear digital signal processing model as shown in Figure 3.9.

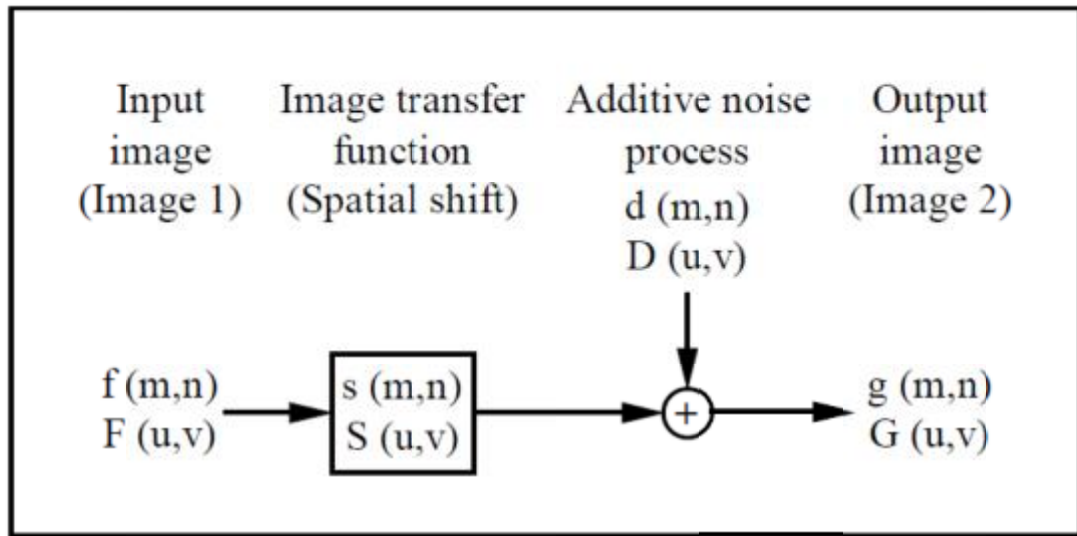


Figure 3.9. Image displacement function

The function $f(m, n)$ describes the light intensity within the interrogation area recorded at time t , and the function $g(m, n)$ describes the light intensity recorded at time $t + \Delta t$. The latter can be considered the output of an image transfer function $s(m, n)$, taking $f(m, n)$ as input, and with the noise function $d(m, n)$ added. Capitalized functions are the Fourier transforms of the corresponding lower case functions, and (u, v) are coordinates in the spatial frequency domain. The spatial shifting function $s(m, n)$ is directly related to the flow and the time between the two recordings, while the noise function $d(m, n)$ is a result of seeding particles moving into or out of the interrogation area in the period between the two recordings. The major task in PIV is the estimation of the spatial shifting function $s(m, n)$ on the basis of known (measured) values of $f(m, n)$ and $g(m, n)$, but the presence of noise complicates matters. The method chosen to estimate the displacement function $s(m, n)$ is the statistical technique of spatial cross-correlation. The discrete cross-correlation function $\phi_{fg}(m, n)$ of the sampled regions $f(m, n)$ and $g(m, n)$ is given by the expected value:

$$f_{fg}(m, n) = E[f(m, n), g(m, n)]$$

$$f_{fg}(m, n) = \sum_{k=-\infty}^{\infty} \sum_{l=-\infty}^{\infty} f(k, l) \cdot g(k + m, l + n)$$

In the literature, cross-correlation is often normalized to obtain values between 0 and 1, but in this context only relative correlation levels within the investigated interrogation area are of interest. A high cross-correlation value is observed, where many particles match up with their corresponding spatially shifted partners, and small cross-correlation peaks may be observed when individual particles match up with other particles. The former is known as true correlations, while the latter is called random correlations. Seeding particles entering or leaving the interrogation area between the recording of the first and the second image, will not contribute to the true correlation since either the initial or final particle position is missing. They do however contribute to the random correlations and as such decrease the signal-to-noise ratio. In PIV this phenomenon is often referred to as “loss-of-pairs” or “signal drop-out”. Nevertheless, the highest correlation peak can safely be considered to represent the best match between the functions $f(m, n)$ and $g(m, n)$ when the number of matching particle pairs is sufficiently large. The position of the peak in the correlation plane corresponds directly to the average particle displacement within the interrogation area investigated. In practice, to efficiently compute the correlation plane, Fourier transform processing is used in PIV. The justification of using FFT processing is as follows. A camera image may be considered a two-dimensional signal field analogous to a one-dimensional time series. Many one-dimensional signal processing techniques can readily be extended to two dimensions, and in this case Fast Fourier Transforms (FFT's) are used to speed up the cross-correlation process. Rather than performing a sum over all elements of the sampled region, the operation can be reduced to a complex conjugate multiplication of each corresponding pair of Fourier coefficients. The resulting new sets of coefficients are then inversely transformed to obtain the cross-correlation function ϕ_{fg} . The process is illustrated in Figure 3.10.

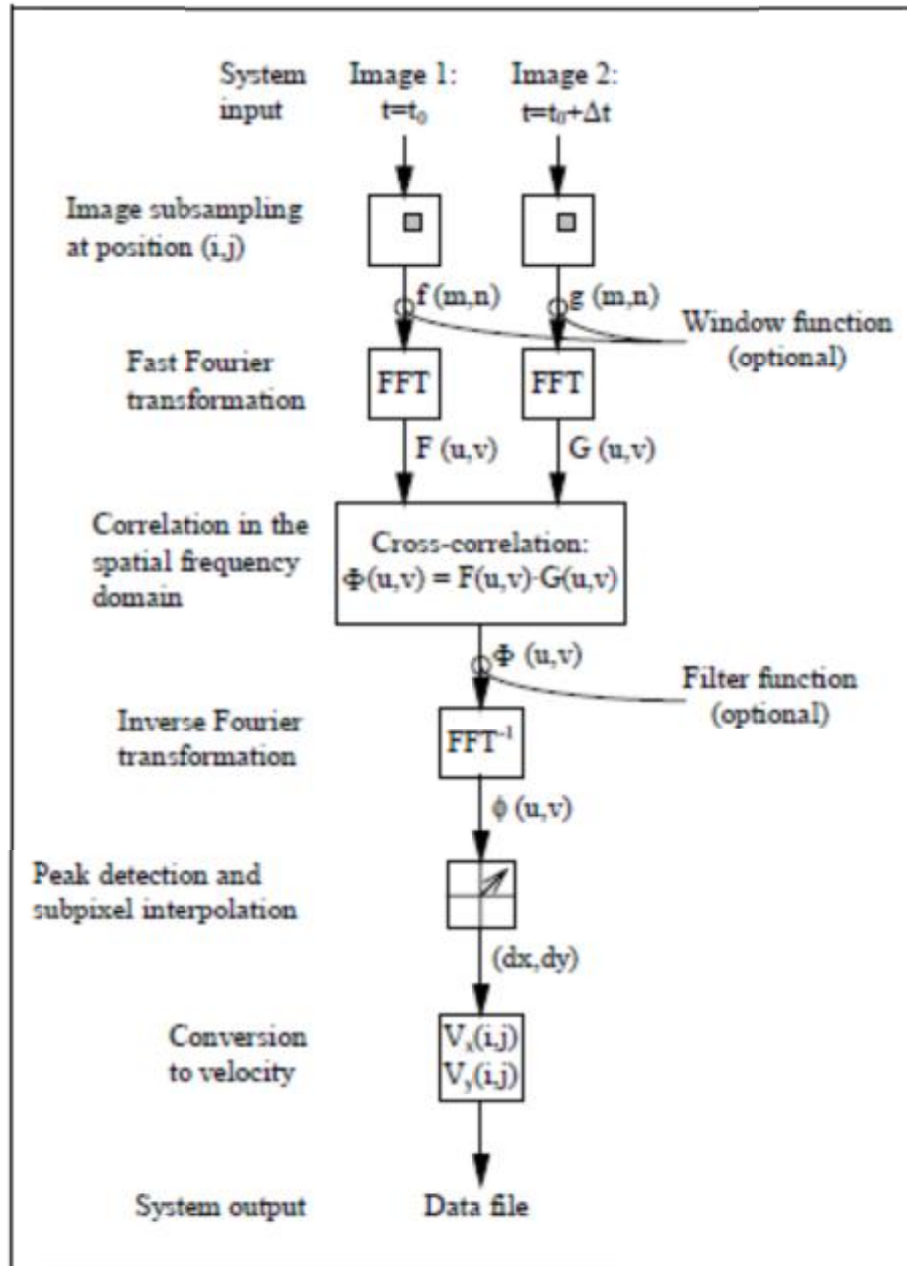


Figure 3.10. Numerical processing flowchart of PIV

Readers familiar with digital signal processing may know of the difference between a cross-correlation obtained via the summing procedure and the Fourier transform technique used here. The latter assumes the sampled regions to be periodic in space, and generates a circular form of the cross-correlation which does not fall off to zero at the edges of the interrogation area as with the linear technique. The immediate consequences of this is an artificial increase in the background noise

especially along the edges of the interrogation area, and the window and filter functions shown in the above figure are ways of dealing with this. These window and filter functions will be described in more detail later.

The Nyquist sampling criterion associated with the discrete Fourier transforms limits the maximum recoverable spatial displacement in any direction to half the window size in that direction. In reality even this displacement is often too large for the technique to work properly, since the signal-to-noise ratio in the cross-correlation decreases with increasing spatial shift. Given a window side of length N , a maximum particle displacement of $N/4$ is recommended to ensure a reasonable signal-to-noise ratio.

Another important parameter influencing the signal-to-noise ratio is the number of seeding particles within each interrogation area. With a large number of seeding particles, there will be many true correlations ensuring a high signal-to-noise ratio, and in this case an average displacement exceeding $N/4$ may be accepted. With a relatively small number of particles within the interrogation area, the signal-to-noise ratio will decrease, and generally a minimum of 5 particles/region is recommended to achieve reasonable results using the cross-correlation technique (Figure 3.11).

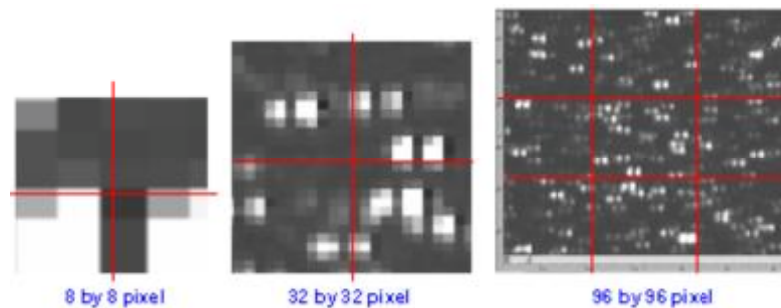


Figure 3.11. Successful vectors and satisfactory PIV signals require 5-10 image pair per interrogation area

Large amount of closely spaced vectors, squeezed out of one single standard size digital CCD image, is mind-bending. But it may not produce results to the full potential of the PIV technique in resolving your flow. Many researchers well documented work has clarified the details in obtaining successful PIV results. Taking

these details into account, one can obtain a number of benefits over standard PIV cross correlation techniques. Using Adaptive correlation technique, one will find an increased flexibility in capturing the flow characteristics with emphasis on the parameters, which are exactly of interest. This may be more successful vectors, covering a larger area while capturing the flow details, or an increased dynamic range and accuracy.

The fundamental principle of Adaptive correlation is an iterative procedure: From an initial guessed offset value, an offset is introduced from the first window (the interrogation area in the image frame from laser pulse one) to the second window. The obtained vector is validated and is used as a new estimate for the window offset. A new run is made, but this time with a smaller window (interrogation area).

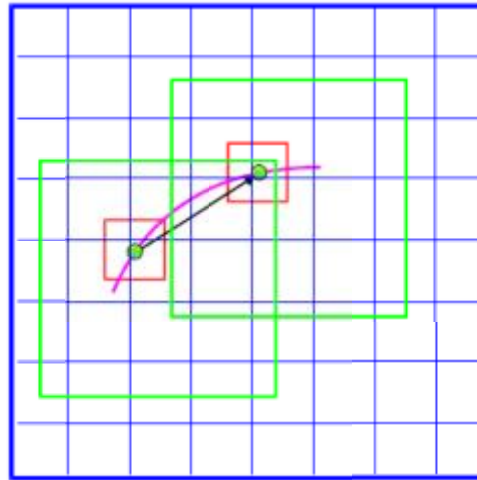


Figure 3.12. The interrogation areas used in adaptive correlation

The main benefit derived from using the shifted window is capturing the particle images that left the interrogation area during the time between the two light pulses. Loss of these particle images is known as the "in-plane dropout", which reduces signal strength and, as a result, the number of successful vectors that can be obtained.

In standard PIV cross-correlation, velocity gradients within an interrogation area produce less successful vectors because the signal peak in the correlation is

lowered. As a rule of thumb, the difference between the largest and smallest velocity within an interrogation area should be less than 5% of the mean velocity. Using adaptive correlation helps in two ways. First of all, the signal strength is raised due to the capture of the in-plane dropout. Secondly, a refinement of the interrogation area is possible because an adaptive window offset may be applied, again producing a successful signal.

As a conclusion, the Adaptive correlation results in a de-coupling of interrogation area and velocity measurement. The de-coupling between the size of the interrogation area and particle-displacement measurement obtained with Adaptive correlation also means that the term “spatial resolution” has a completely different meaning. The particle velocity is resolved with the dynamic range given by the system and the experimental circumstances. The spatial resolution of the particle trajectory is in principle given by the actual particle displacement (Figure 3.13) and by taking Nyquist sampling theorem into consideration, i.e., the resolution is $2 \cdot \Delta X$. The number of samples, which the velocity estimate is based upon, is related to the size of the interrogation area, or more precise, the number of particle images within the interrogation area.

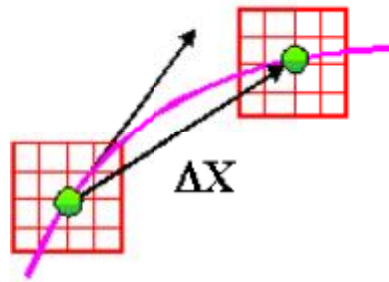


Figure 3.13. More parameters are of importance, using Adaptive correlation: interrogation area, particle displacement ΔX , actual velocity and actual particle trajectory

In standard PIV correlation, the consideration of spatial resolution is “automatically” included in the set-up, simply because successful vectors require the displacement to be less than 30% of the interrogation area (Flow Manager Software and Introduction to PIV Instrumentation).

The adaptive correlation was used in this study because of its advantages over standard cross-correlation method mentioned above.

3.1.2.1.(6). Validation and Further Analysis

Images were received from CCD camera that has resolution 1024×1024 at a rate of 15 frames per second. The time delay changes from 2 ms and 3 ms between frames depending on the flow conditions and camera location. Digital image was analyzed using DYNAMIC STUDIO software. The image was recorded on a CDD array. A frame grabber in the computer read the camera image from CCD camera and stored it as the digital image file format (TIFF) in the RAM. This digital image was processed and analyzed using the DYNAMIC STUDIO software. During each continuous run, a total of 1000 images were taken. In order to determine the velocity field, adaptive cross-correlation technique, with 32x32-interrogation window, was employed, with an overlap of 50%.

The resulting vector field obtained from DYNAMIC STUDIO software and the corresponding boundaries of objects were then viewed to determine incorrect vectors from interrogation, called outliers. The outliers can result if an incorrect particle correlation takes place (which mostly occurs near the boundaries or within shadow regions), when the particle images are too widely spaced for interrogation window side specified, or the power of laser sheet is poor.

Vector validation software called CLEANVEC was used to remove incorrect vectors. The software CLEANVEC contains four statistical filters designed for incorrect vectors removal:

- Absolute range filter
- RMS tolerance filter
- Magnitude difference filter
- Quality filter

Three of these four filters were used for the purposes of eliminating incorrect vectors. Here the quality filter requires a correlated data, which are supposed to be done by interrogation and this data, was not provided by this software.

In the absolute range filter, all streamwise and spanwise velocity components that lie outside of given range were removed. With this filtering method, one can identify a moving reference frame in order to save real vectors that are numerically specified and eliminated incorrect vectors on the main frame. Although very trivial, this filter might be very useful in removing the most tedious incorrect vectors, and hence improves the performance of the other filtering tools.

The RMS tolerance filter removes incorrect vectors those lay outside of given range. This filter must be applied to a velocity field in a reference frame moving with the mean velocity components in both directions, since it is involved with the fluctuating components only.

Finally, the magnitude difference tool removes unnecessary vectors based on the difference in magnitude between a vector and its neighborhood median. This is the local-median test defined by Westerweel (1994) which is the most effective among the available filters. However, it should be handled carefully, because it may lead to an excessive vector in a certain type of flows.

3.1.2.2. PIV parameters of current study

The experiments were performed and the measured data were processed using Dantec Dynamics PIV system and Dynamics Studio Software installed on a computer. The measurement field was illuminated by a thin and an intense laser light sheet by using a pair of double-pulsed Nd:YAG (yttrium aluminum garnet) laser units each having a maximum energy output of 120 mJ at 532 nm wavelength.

The image capturing was performed by an 8 bit cross-correlation charge-coupled device (CCD) camera having a resolution of 1200×1600 pixels, equipped with a Nikon AF Micro 60 *f* / 2.8*D* lens. In the image processing, 32×32 rectangular interrogation pixels were used and an overlap of 50% was employed. A total of 7326 (99×74) velocity vectors were obtained for an instantaneous velocity field at a rate of

15 frames/ s. The plan view of measuring planes (Figure 3.14) covers the downstream of the cylinder with the area of 219mm x 164mm for all configurations and the image magnification was 0.054 mm/pixel. The laser sheet was oriented parallel to the bottom surface of the water channel and the experiments were carried out at 25 mm away from the bottom surface where the water height was kept constant as $h_w=600\text{mm}$ throughout the experiments in order to have deep water conditions (Figure 3.15). All measurements were done on the plan view at mid-plane of the total water height and laser sheet was located parallel to the bottom of the water channel.

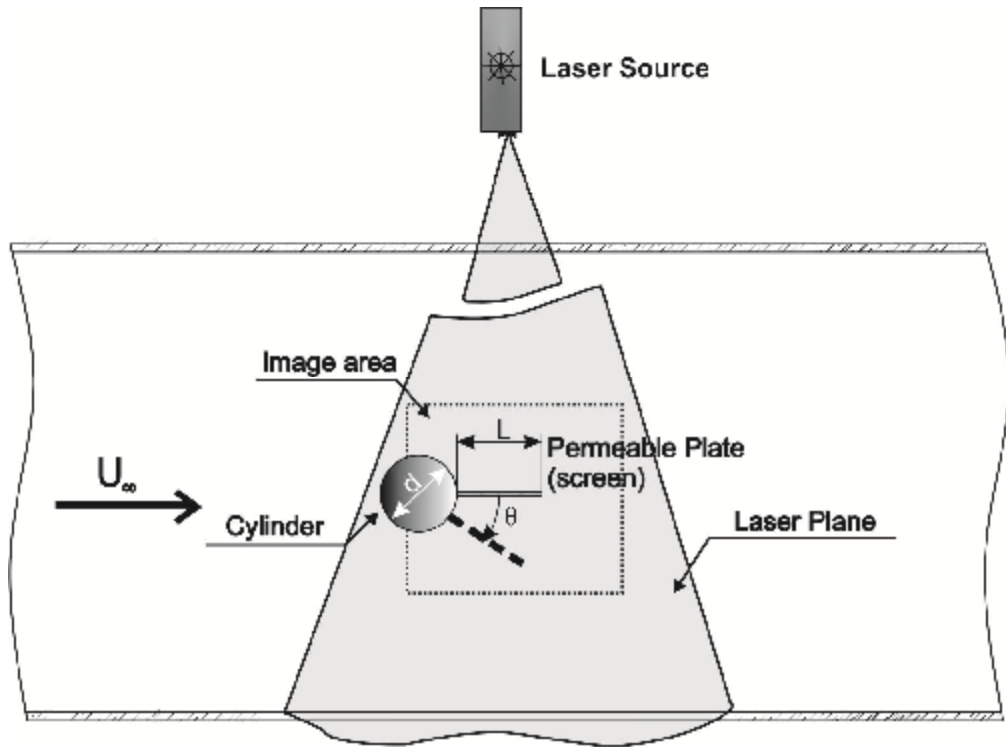


Figure 3.14. Plan view of the experimental set-up

The time interval between pulses was 1.75 ms for all experiments and the thickness of the laser sheet illuminating the measurement plane was approximately 2 mm. The time interval and the laser sheet thickness were selected such that the maximum amount of particles in the interrogation window was obtained. The number of particles in an interrogation area was in between 20 and 25. The uncertainty in velocity relative to depth averaged velocity is about 2% in the present

experiments. The water was seeded with 10 μm diameter hollow glass sphere particles which were shown in Figure 3.5. The instantaneous images were captured, recorded, and stored in order to obtain averaged-velocity vectors and other statistical properties of the flow field. Spurious velocity vectors (less than 3%) were removed using the local median-filter technique and replaced by using a bilinear least squares fit technique between surrounding vectors. The velocity vector field was also smoothed to avoid dramatic changes in the velocity field using the Gaussian smoothing technique. The vorticity value at each grid point was calculated from the circulation around the eight neighboring points.

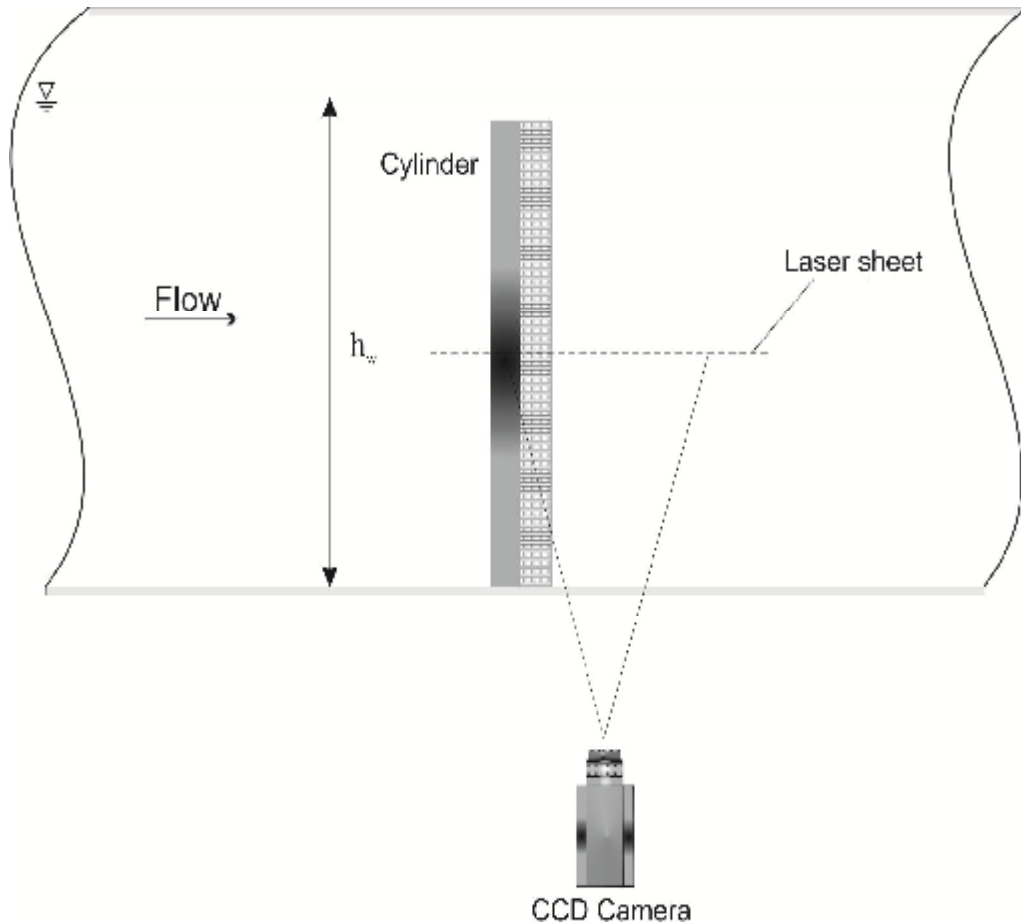


Figure 3.15. Side view of the experimental set-up

3.2.3. Experimental Model

Within the context of vortex shedding suppression and elimination of vortex induced vibrations, the control method used in this study is using attached permeable plate along the midspan of the circular cylinder (Figure 3.16). The permeable plates were obtained by cutting the woven screens which are made of chrome nickel wires. The woven screens were heated and attached onto the cylinders which are made of plexi glass material.

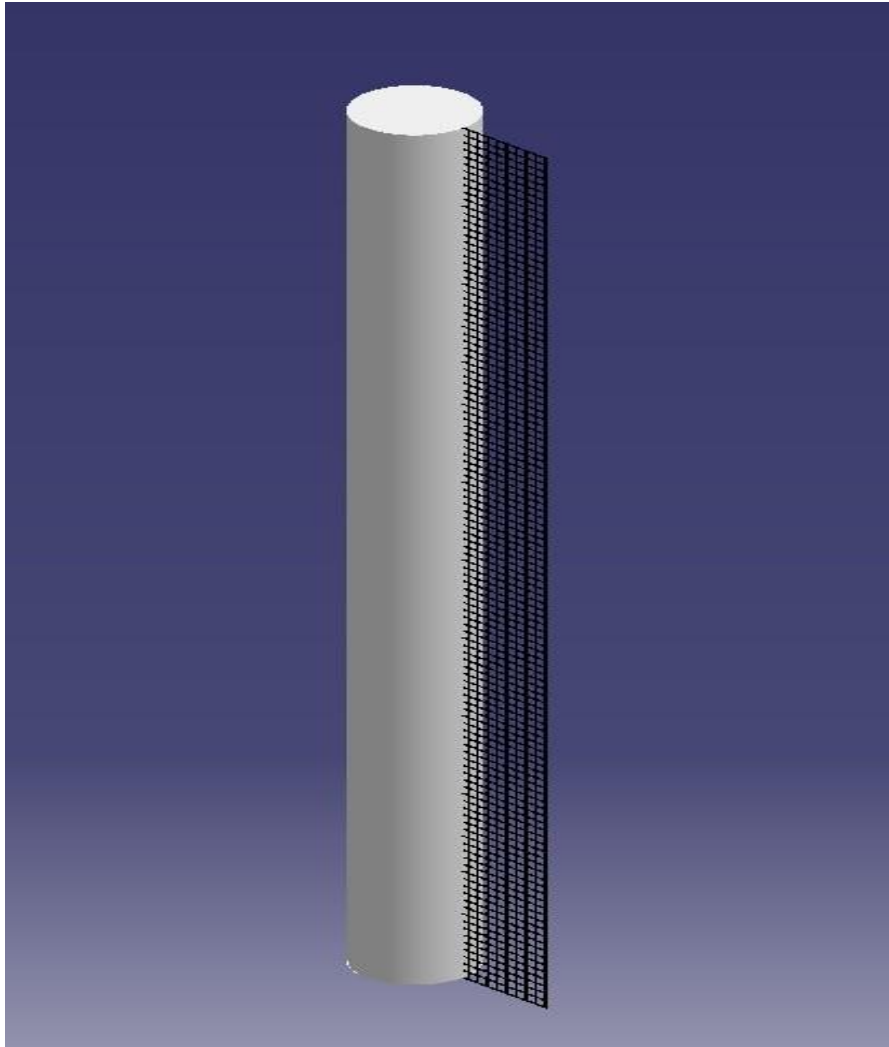


Figure 3.16. The control method; cylinder with an attached permeable plate

3.2.3.1. Investigation Parameters

The main parameters of this control method are the porosity of permeable plates, β which is defined as the ratio of the gap area on the screen to its whole surface area, the rotation angle with respect to the centerline of the cylinder, θ and length of the permeable plate, L/d . The calculation of the porosity and schematic presentation of the plate angle, as well as the plate length are presented in Figure 3.17.

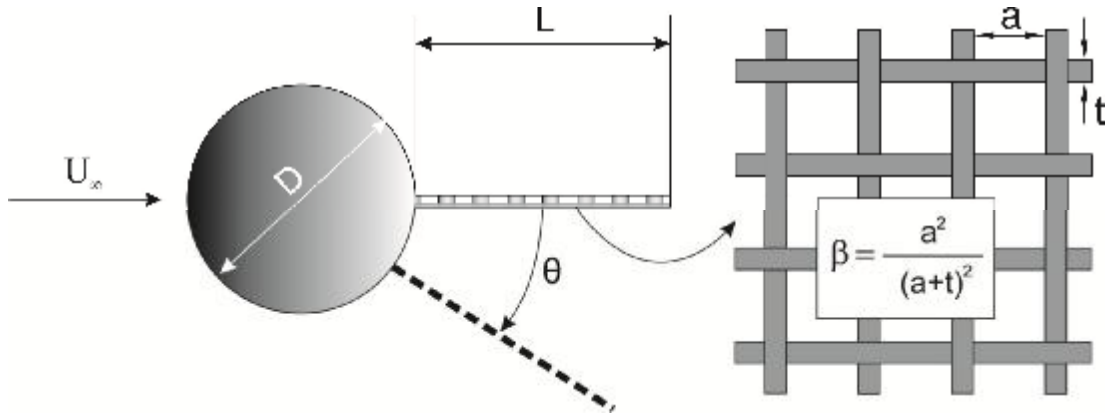


Figure 3.17. Schematic presentation of the plate angle, θ definition of the porosity, β and plate length, L

Keeping constant the cylinder diameter as $d=50$ mm and free stream velocity as $U_{\infty}=95.2$ mm/s (corresponding to a Reynolds number of $Re_d=5000$ based on the cylinder diameter), five different screens were used corresponding to five porosities as, $\beta=0.4, 0.5, 0.6, 0.7$ and 0.8 while the plate angle, θ was varied for 17 different angles ($\theta=0^\circ, 15^\circ, 30^\circ, 35^\circ, 40^\circ, 45^\circ, 60^\circ, 75^\circ, 90^\circ, 95^\circ, 100^\circ, 105^\circ, 120^\circ, 135^\circ, 150^\circ, 165^\circ, 180^\circ$) in order to understand the effect of porosity and plate angle on flow control. Furthermore the case for the solid plate ($\beta=0.0$) which is known as the splitter plate in literature, was also observed in order to compare the results with permeable cases. The constructed cylinder- plate arrangements are shown in Figure 3.18.

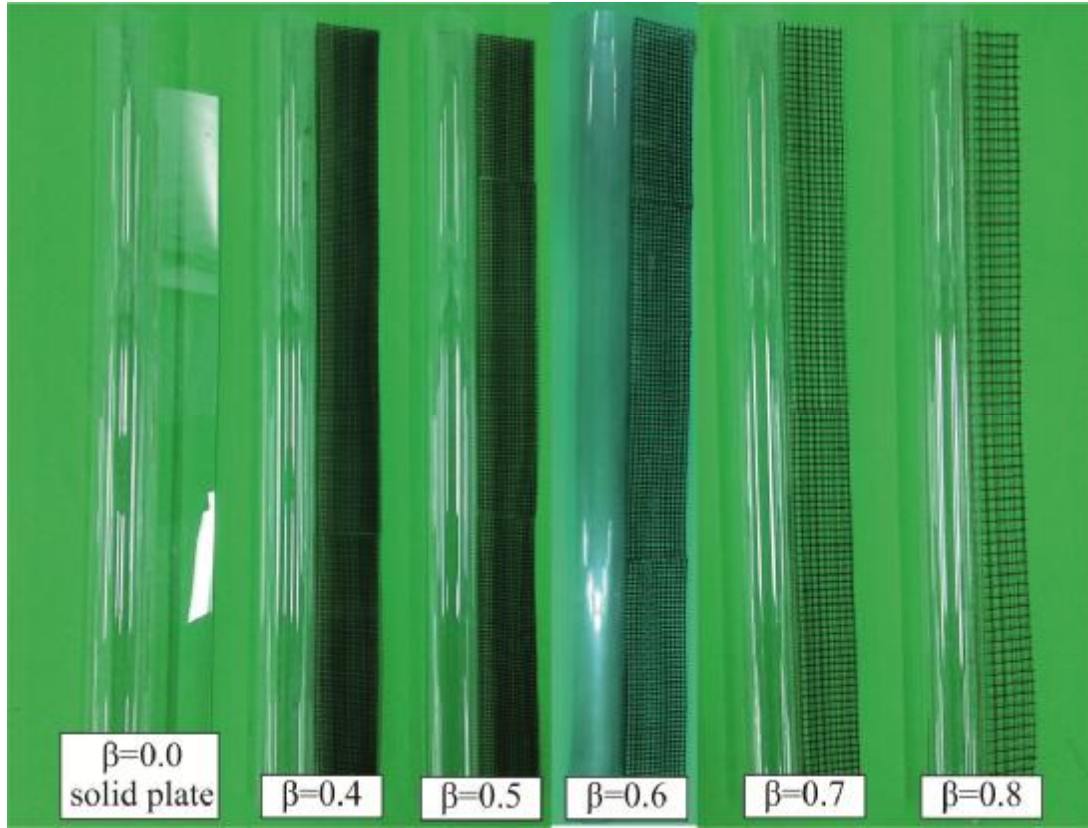


Figure 3.18. Manufactured test models for the solid plate and permeable plate cases

4. RESULTS AND DISCUSSIONS

4.1. The Effect of Porosity and Plate Angle on Flow Characteristics

4.1.1. Turbulent Kinetic Energy

As emphasized in previous sections, main objective of this research is to investigate the effect of permeable plates on flow control downstream of bluff bodies. A powerful indication of controlled flow in the case of interacted shear layers is the reduction in velocity fluctuations in the wake region of related body subjected to fluid flow. In this context, normalized Turbulent Kinetic Energy, TKE concentrations are presented in this section for all porosities and plate angles. The values of TKE have been made dimensionless by dividing them with the square of the depth-averaged free-stream velocity, U^2 for better understanding. In two-dimensional PIV, the values of $\langle u'^2 \rangle$ and $\langle v'^2 \rangle$ can be calculated in the usual manner (by subtracting the mean values of the velocity components from their instantaneous values and then by averaging the products) from the 1000 instantaneous data set obtained by two-dimensional PIV. Assuming isotropic flow, the third fluctuating velocity component can be estimated by a 2D approximation (Sheng et al. 2000), namely the third component has been supposed to be equal to the half of $(\langle u'^2 \rangle + \langle v'^2 \rangle)$, and then TKE can be calculated with the following expression

$$\text{TKE} = \frac{3}{4} (\langle u'^2 \rangle + \langle v'^2 \rangle)$$

Both minimum and incremental value of dimensionless TKE is 0.2 for all TKE contours throughout the thesis and for each figure, the results of natural cylinder case is given in order to compare the results easily.

Figure 4.1 shows the normalized TKE contours for natural cylinder and the cylinder with splitter plates of various angles. Concentrations of TKE contours

gradually decreases for $\theta=0^\circ$ (splitter plate case) as an indication of flow control downstream of the body which shows a good agreement with the literature. Slight changes on the flow structure exist for the case of $\theta=15^\circ$, however further increase in the plate angle results in larger wake widths with higher concentrations up to $\theta=45^\circ$. As expected, increasing the plate angle to larger values greatly enlarges the wake width since the splitter plate takes a dominant role on the formation of the structure of flow especially for $60^\circ \leq \theta \leq 150^\circ$. An interesting flow structure should be pointed out here, in which there exist two independent peaks for $0^\circ \leq \theta \leq 135^\circ$ compared to the natural cylinder case. These peaks are getting diverged by increasing the plate angle where the underlying physics of this flow structure must be the formation of shear layers, i.e. the shear layers are being formed by upper and lower shoulders of the cylinder in natural case, however it is obviously seen from Figure 4.1 that the splitter plate directly interferes to the lower shear layer of the cylinder. Hence, it can be concluded for $30^\circ \leq \theta \leq 105^\circ$ cases that the lower shear layer is generated by the plate with increasing angle; therefore the lower shear layer is formed away from the cylinder centerline compared to the natural cylinder case. This is why the peaks formed in the wake region diverge from each other.

For $120^\circ \leq \theta \leq 150^\circ$, the peaks get closer to each other since the splitter plate is not responsible for the formation of lower shear layer. At $\theta=150^\circ$, a single dominant peak exists similar to the natural cylinder case, however the concentration of the TKE contours are still comparably greater than that of the natural case. The case of $\theta=165^\circ$ shows an interesting situation where the TKE concentration dramatically decreases by 63.2% compared to the natural case which needs a detailed analysis for $150^\circ \leq \theta \leq 165^\circ$. As expected, $\theta=180^\circ$ is similar to that of natural cylinder case both in shape of the structure and intensity.

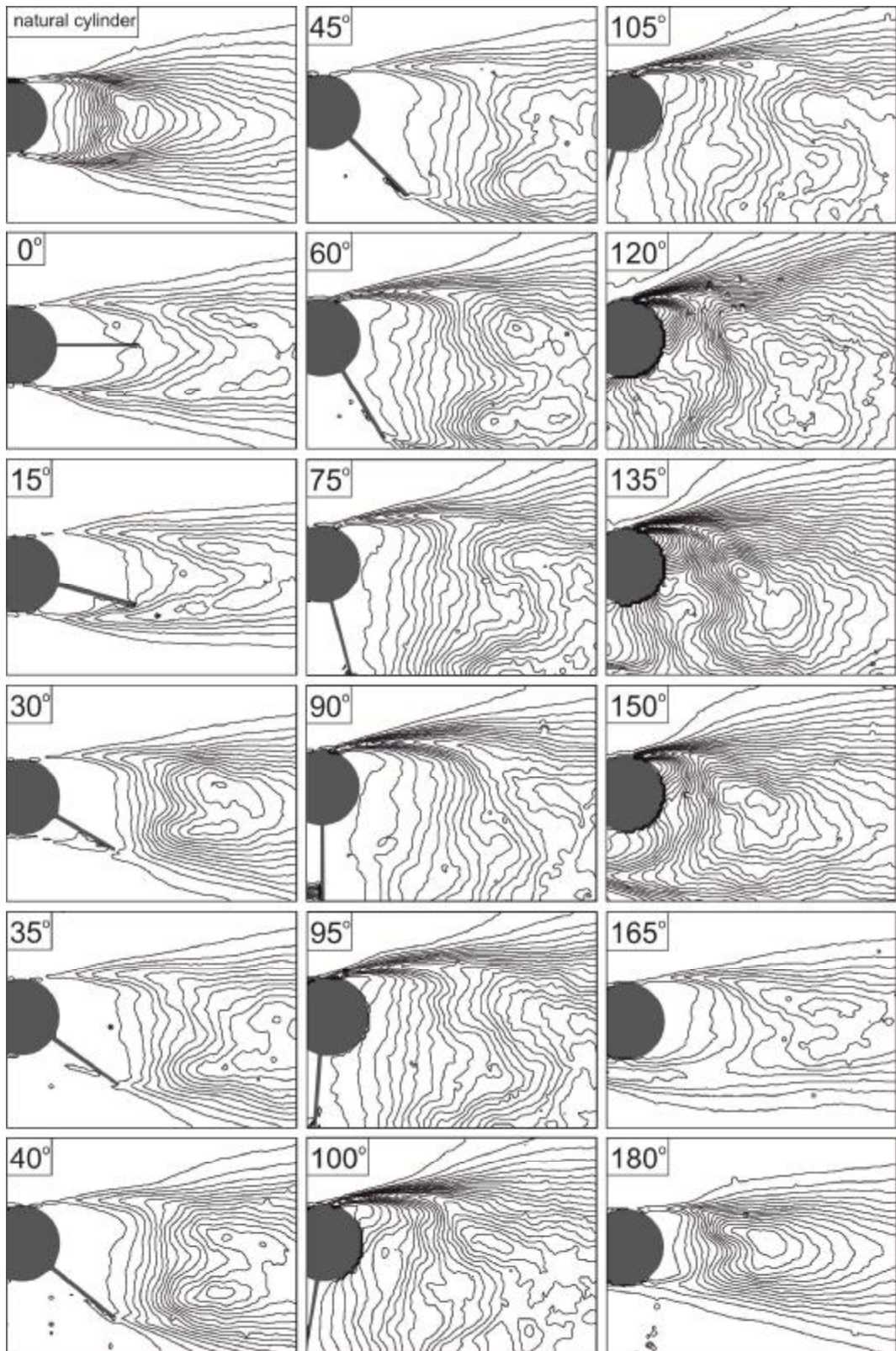


Figure 4.1. Time- averaged normalized Turbulent Kinetic Energy (TKE) distribution for the natural cylinder and all cases of $\beta=0.0$ (solid plate)

The use of a permeable plate, instead of a solid one brings a new insight to the passive control techniques related to the suppression of vortex shedding and its adverse effects such as Vortex Induced Vibrations (VIV). Apart from the solid plate, the permeable plate would give an ability to free stream flow to pass through the meshes which means the addition of momentum into the flow. This additional momentum might result in important effects on flow structure for any application it may be used.

Figure 4.2 presents the TKE contours for the permeable plate having $\beta=0.4$ porosity for all plate angles including the natural cylinder case for comparison purpose. First of all, it is clearly seen that the angled permeable plate cases have smaller magnitudes in TKE with lower concentrations compared to the natural cylinder case, except for $\theta=180^\circ$. The use of a permeable splitter plate ($\theta=0^\circ$ case) shows a different flow structure than that of for the solid splitter plate as explained in Figure 4.1, in which the formation of a single peak at farther regions of the wake occurs with respect to the natural cylinder case where the solid plate generates two distinct peaks at all. The case of $\theta=15^\circ$ adversely affects the TKE values where the concentrations increase compared to $\theta=0^\circ$ case. As mentioned before, on and after the plate angle of $\theta=30^\circ$, the lower shear layer is affected by the permeable plate. Unlikely to the solid plate, the permeable plates provide the opportunity of fluid flow through the screen holes as being a jet-like flow, resulting in momentum injection from the rear side of permeable plate. The existence of the permeable plate has two main objectives on the flow structure; first is to directly prevent a vortex roll-in from the lower side of the cylinder, and secondly to inject high amount of momentum through its holes which directly prevents the roll-in of the upper shear layer. Consequently the lower vortex is not formed and upper vortex is not rolled into the wake region and finally the vortex shedding is successfully prevented. According to this physical reality, the velocity fluctuations dramatically decrease downstream of the cylinder for $35^\circ \leq \theta \leq 150^\circ$. The most effective case on reduction of maximum TKE seems to be the case of $\theta=45^\circ$ where the value of TKE is measured as 0.85242 while it is measured to be 0.3285 for the natural cylinder case which means a 74% reduction as being an indication of flow control concerning the vortex shedding. The

lower shear layer is generated by the permeable plate at farther locations normal to the cylinder centreline; therefore it is supposed to have the elimination of vortex shedding and consequent Karman Vortex Street downstream of the cylinder which will be discussed in details on further sections. It can be concluded that the use of a permeable plate having a porosity value of $\beta=0.4$ reduces the velocity fluctuations in the wake region of the cylinder for plate angles of $35^\circ \leq \theta \leq 150^\circ$. However, each plate angle shows different effects on flow control, i.e. the plate angles of $95^\circ \leq \theta \leq 105^\circ$ come across to the separation point of the cylinder which is assumed to be occurred for laminar boundary layer flows ($Re \leq 3 \times 10^5$). This interval is also very effective on reducing the maximum values of TKE by directly affecting the separation points. Finally, the case of $\theta=180^\circ$ shows similar behaviour in both distribution and intensity on TKE concentrations with natural cylinder case.

The TKE concentrations of angled permeable plate having a porosity of $\beta=0.5$ are shown on Figure 4.3. One can easily see and conclude that the results of this porosity shows similarly with the results of $\beta=0.4$ case; the effective interval of plate angle is $30^\circ \leq \theta \leq 150^\circ$ where the density of TKE distributions are significantly reduced in the wake region. According to last two figures (Figure 4.2 and 4.3) it can be concluded that the porosities of $\beta=0.4$ and 0.5 affect the flow structure similarly by greatly reducing the velocity fluctuations in downstream direction.

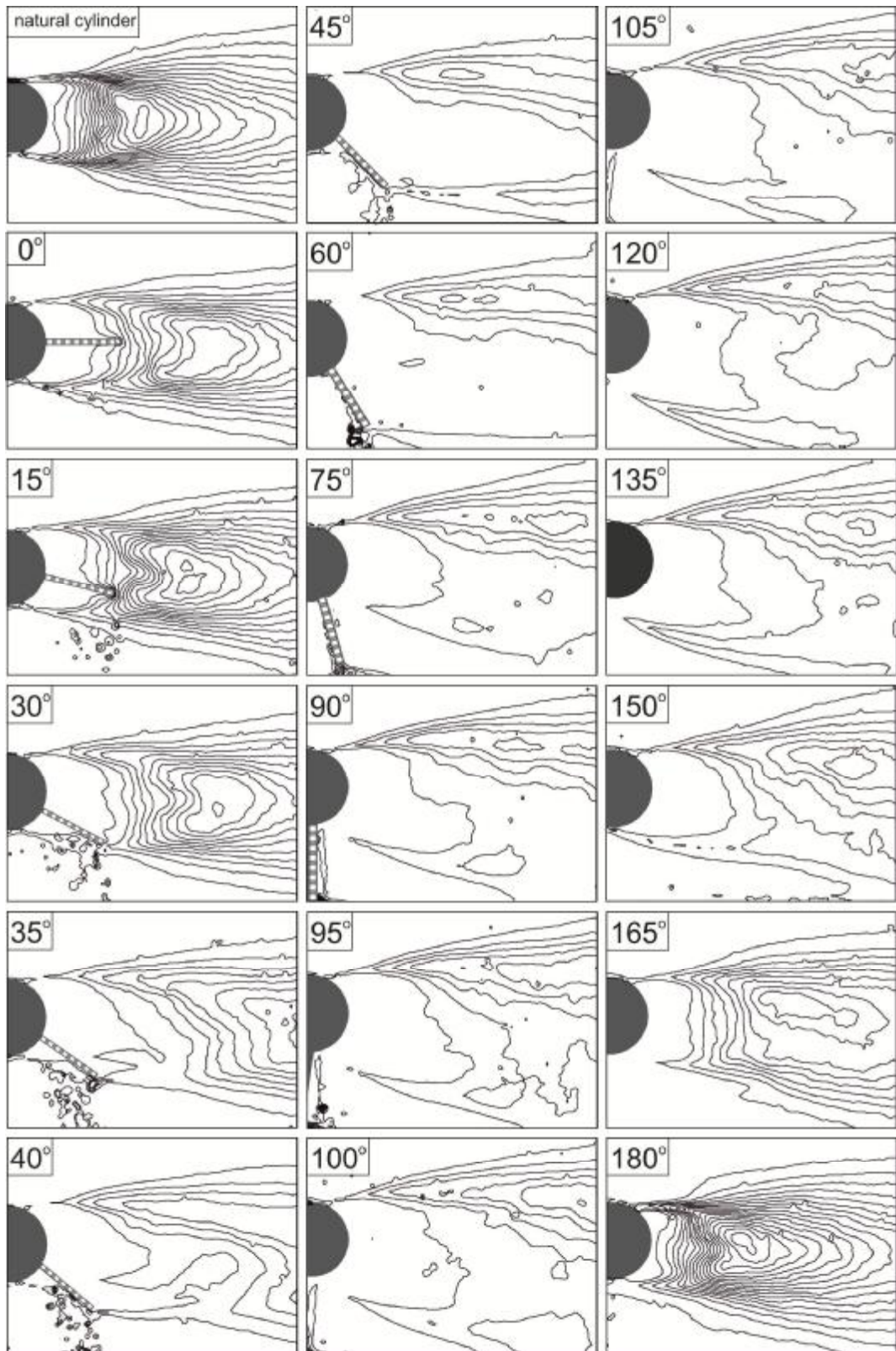


Figure 4.2. Time- averaged normalized Turbulent Kinetic Energy (TKE) distribution for the natural cylinder and all cases of $\beta=0.4$

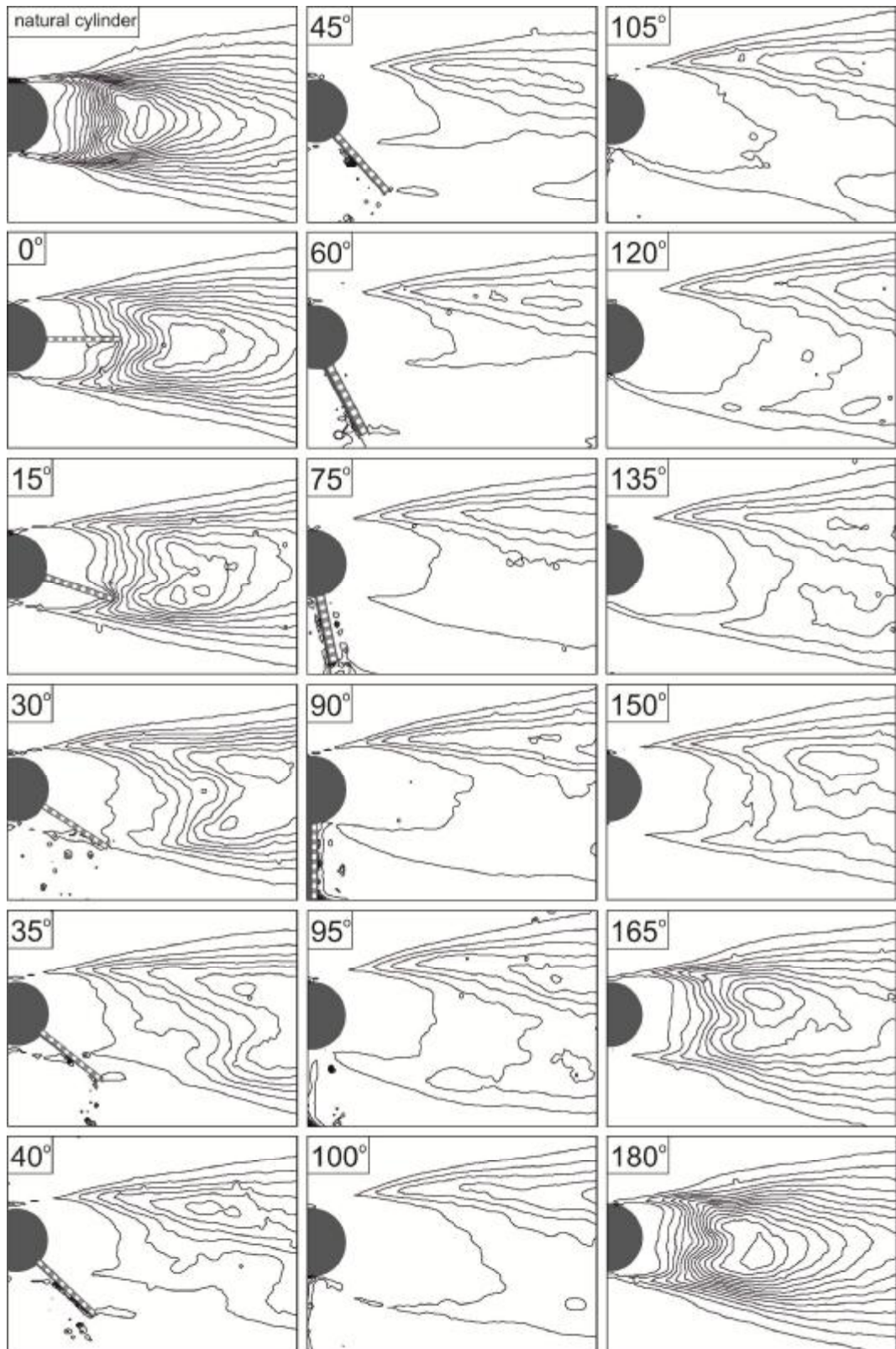


Figure 4.3. Time- averaged normalized Turbulent Kinetic Energy (TKE) distribution for the natural cylinder and all cases of $\beta=0.5$

Since the porosity, β is defined as the ratio of open area to the whole body surface (here the screen) area, an increase in its value means that the size of the holes on screen would increase, too. Hence more flow would pass through the larger sizes which mean more flow injection; however, this does not mean greater momentum injections into the wake at higher porosities. As mentioned earlier, the main mechanism of preventing the vortex shedding and reducing the velocity fluctuations is the elimination of lower vortex formation and prevention of the roll-in of upper shear layer into the wake region (by injection of momentum through the meshes). At this point, the question is; which porosity value would not be able to maintain this mechanism?

Increasing the porosity value to $\beta=0.6$ reduces the effect of permeable plate especially for $0^\circ \leq \theta \leq 30^\circ$ compared to lower porosities. The results are shown in Figure 4.4. The plate angles of $30^\circ \leq \theta \leq 40^\circ$ slightly reduce the TKE intensities but the most effective interval on TKE reduction is obviously seen as $45^\circ \leq \theta \leq 135^\circ$ cases. The latter angles nearly show a similar behavior in both shape of the flow structure and intensity of TKE as that of the natural cylinder case. The results of $\beta=0.6$ tell us the truth of any further increase in open area, e.g. the porosity, β would result in the reduction in the effectiveness of permeable plates on flow control since the physical mechanism of flow control would not survive any more. Nevertheless, $\beta=0.6$ is still very effective on reducing the concentrations of TKE and can be suggested to be used as a vortex shedding control device.

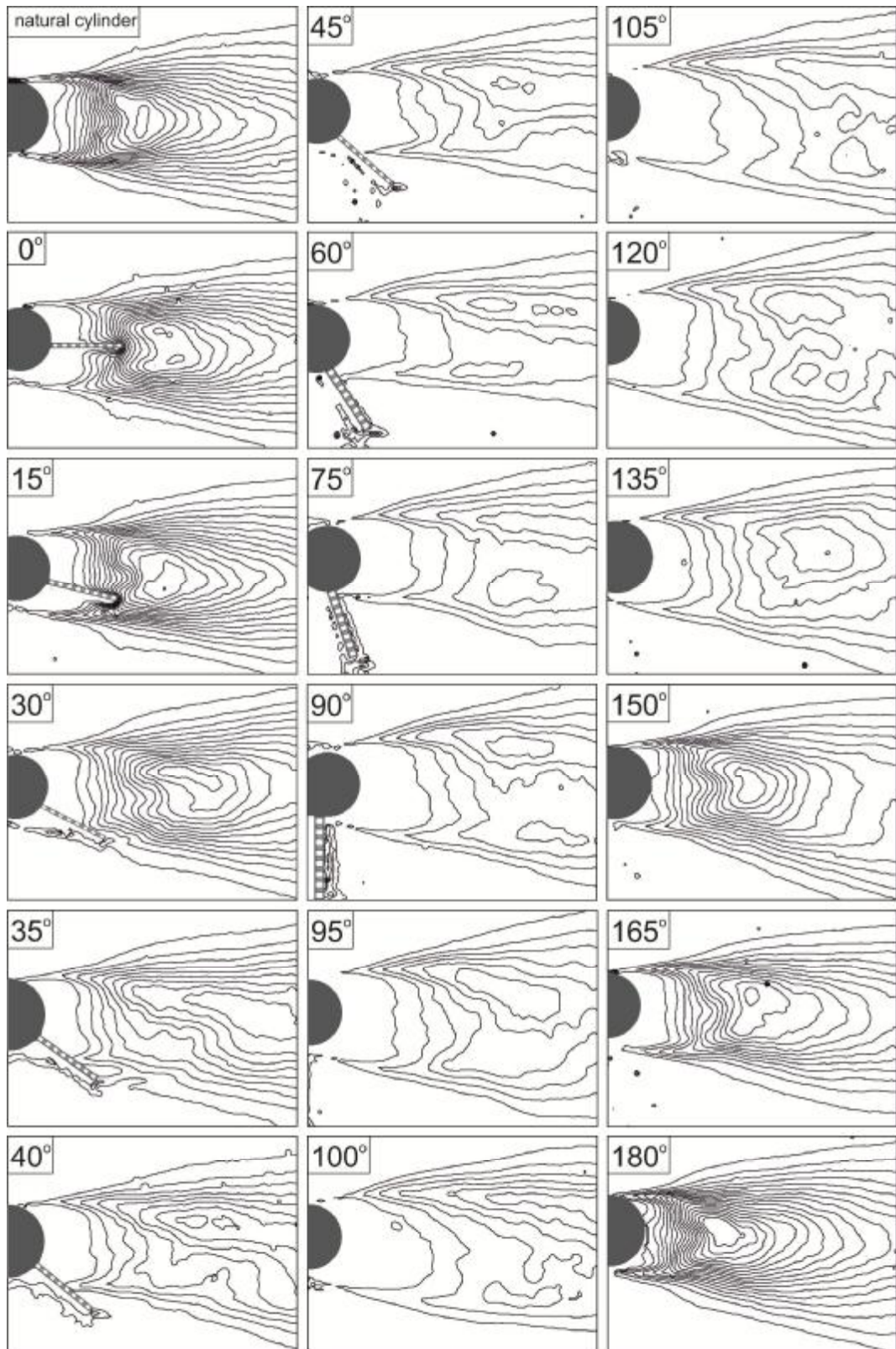


Figure 4.4. Time- averaged normalized Turbulent Kinetic Energy (TKE) distribution for the natural cylinder and all cases of $\beta=0.6$

Figure 4.5 shows the TKE distributions of $\beta=0.7$ case. First of all, concentrations of TKE seem to be increased compared to the lower porosities, even though the values are still smaller than that of the natural cylinder. The case of $\theta=0^\circ$ clearly shows the effect of increased porosity, i.e. the entrainment of the flow through the plate is easily seen by the TKE contours around the plate surface. However, the location of peak magnitude of TKE is still shifted in downstream direction. By giving angle to the plate, the location of peak magnitude starts to move towards the upper side and the concentrations gradually diminish downstream of the cylinder. The effective range of angle seems to be $35^\circ \leq \theta \leq 100^\circ$ for $\beta=0.7$ case and for $\theta \geq 105^\circ$, the magnitude of concentrations dramatically increases and gets closer to the value of natural cylinder case. It should be pointed out that the velocity fluctuations just downstream of the permeable plate significantly increase as an indication of high amount of fluid flow through the meshes. This condition seems to be valid for $60^\circ \leq \theta \leq 95^\circ$ according to the available flow field; however it is supposed to be the case for greater plate angles, too.

Further increase in porosity is shown in Figure 4.6 where the results of $\beta=0.8$ are presented. Since the open area gets its maximum value in this porosity, the fluid directly flows through the meshes, and the control mechanism does not work anymore even for the angled plate cases. The malfunction of the control mechanism means that the velocity fluctuations are grater with respect to the lower porosities, however the peak magnitudes are still lower than that of the natural cylinder results except for $\theta=35^\circ$ and 40° (an increase in intensity of TKE occurs for those plate angles compared with the natural cylinder case). Finally it can be concluded that the $\beta=0.8$ case cannot be suggested for effective flow control, however the angle range of $45^\circ \leq \theta \leq 95^\circ$ could be given as an active limits of this porosity for flow control since small amount of reduction in concentrations takes place.

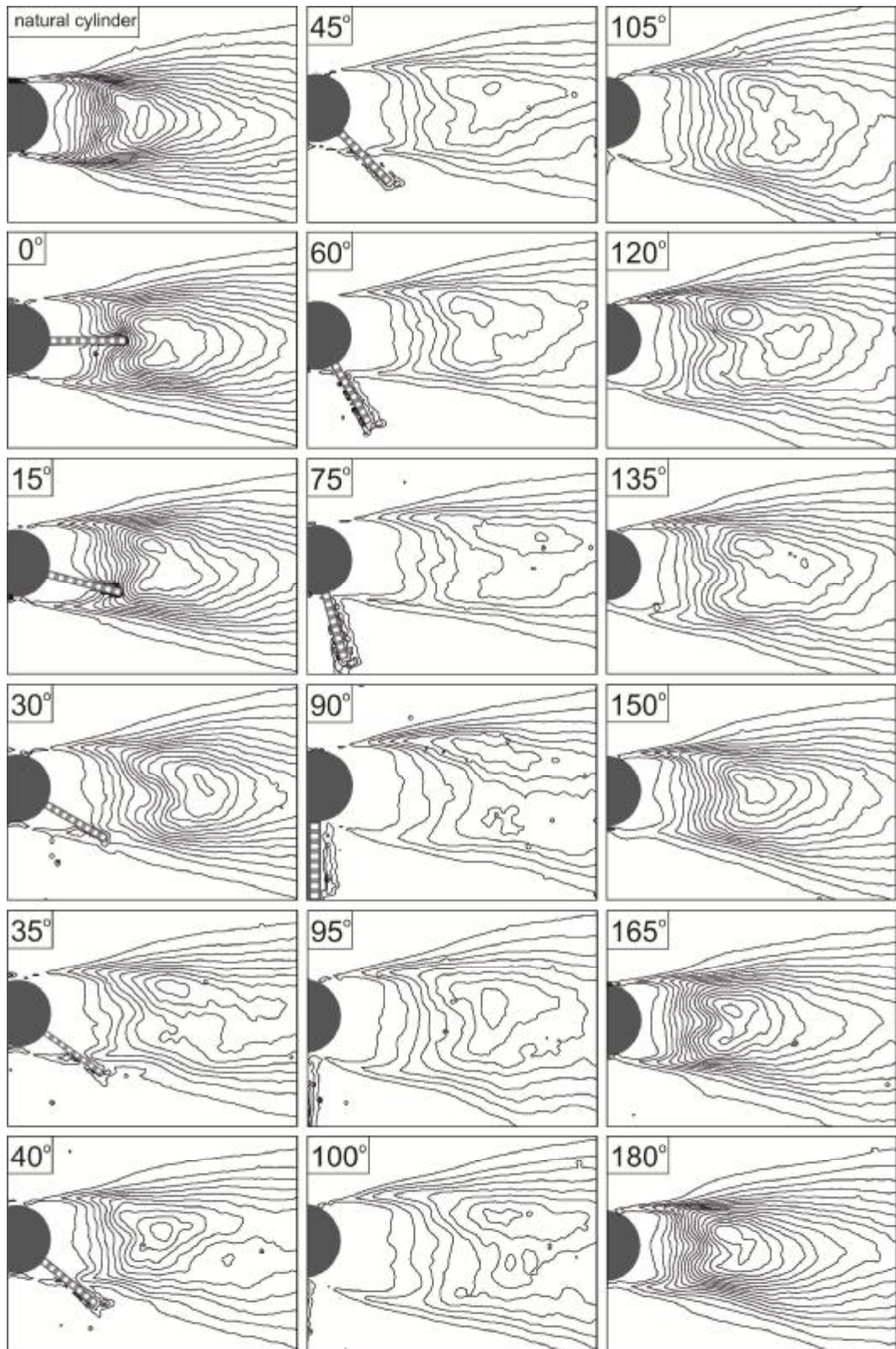


Figure 4.5. Time- averaged normalized Turbulent Kinetic Energy (TKE) distribution for the natural cylinder and all cases of $\beta=0.7$

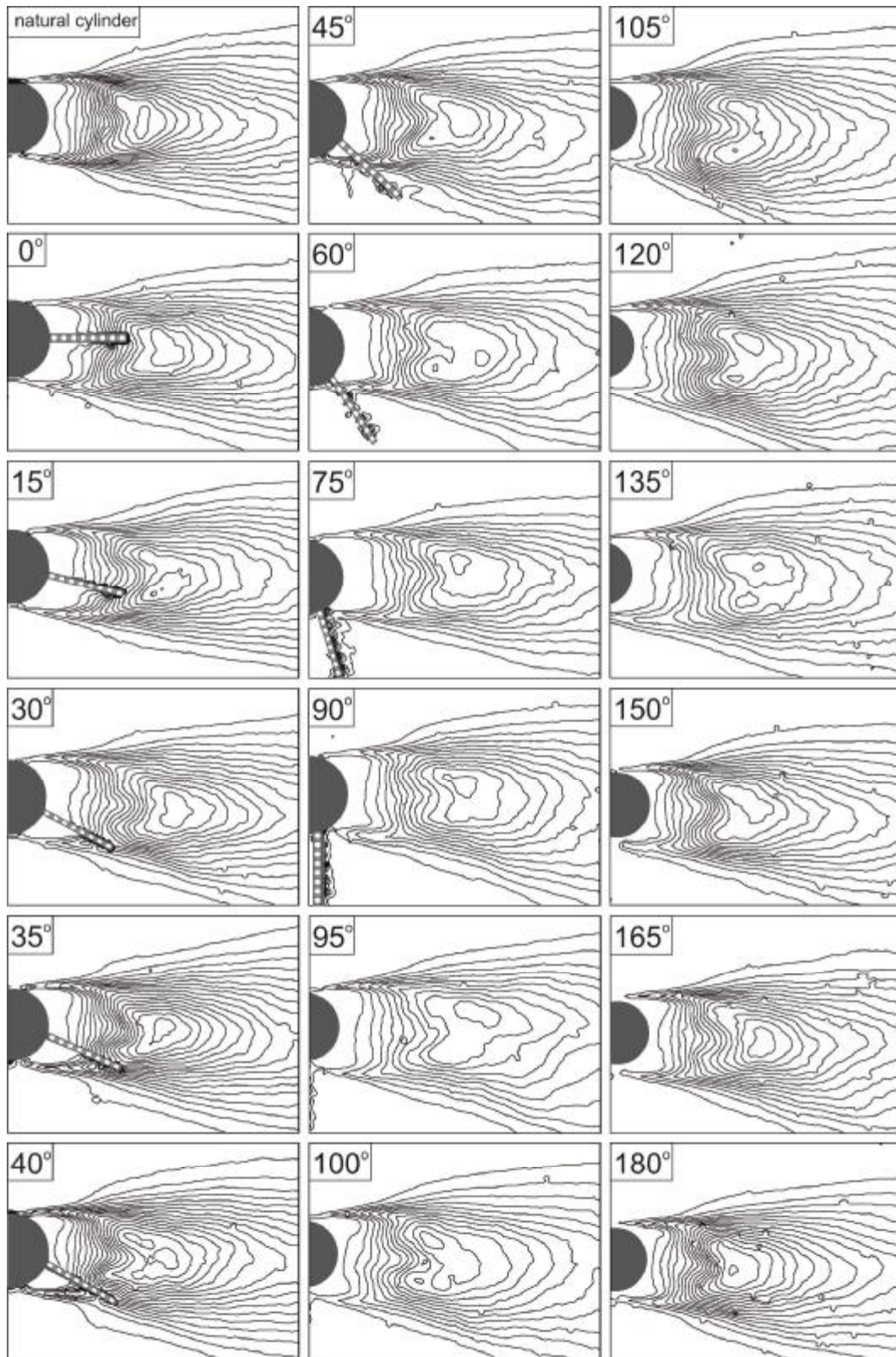


Figure 4.6. Time- averaged normalized Turbulent Kinetic Energy (TKE) distribution for the natural cylinder and all cases of $\beta=0.8$

Up to this point, the effect of porosity, β and plate angle, θ is tried to be shown by normalized TKE contours for each porosity values, independently. It is important to see the effects of parameters on peak magnitudes of TKE by two dimensional curves which are given in Figure 4.7. The maximum value of TKE generated downstream of the natural cylinder is measured to be 0.3235 which is shown by the red dot on y axis of the graph. The solid plate is significantly effective on reduction in the peak values for plate angles of $0^\circ \leq \theta \leq 60^\circ$ whereas the peak magnitude starts to increase for $\theta \geq 75^\circ$ and reaches its maximum value by 79% compared to the natural cylinder case. Nevertheless, a reduction of 55% could be achieved using a solid plate with plate angle of $\theta = 15^\circ$. Interestingly a sudden decrease in maximum value takes place when applying a plate angle of $\theta = 150^\circ$ and 165° which could be a new research interval to be studied in details. It is clearly seen by the graphical results that the peak magnitudes of TKE dramatically reduce by using the permeable plates especially for $0.4 \leq \beta \leq 0.6$ porosity range. In this range of porosity, TKE_{max} gradually decreases by increasing plate angle (except for $\theta = 15^\circ$) for plate angles of $0^\circ \leq \theta \leq 135^\circ$ and show a typical trend in reductions. After the plate angle of $\theta \geq 150^\circ$, the trend of maximum TKE curves begins to steering up to the values of natural cylinder case as clearly seen on the graph for $0.4 \leq \beta \leq 0.6$. The maximum percent reduction in peak magnitudes of TKE achieved in this interval is found to be 74% for $\beta = 0.4$ at $\theta = 45^\circ$. The porosity of $\beta = 0.7$ is also very effective on recusing the maximum values and shows a typical trend as evaluated for lower porosity values, however its effectiveness interval limited for $45^\circ \leq \theta \leq 100^\circ$. General behavior of $\beta = 0.8$ is being closer to the value of natural cylinder result of 0.3285, however a slight decrease in peak magnitude of TKE exists for $45^\circ \leq \theta \leq 95^\circ$ as demonstrated by TKE contours.

According to this graphical results, it is concluded that the porosity range of $0.4 \leq \beta \leq 0.6$ and plate angle range of $45^\circ \leq \theta \leq 135^\circ$ are the effective parameters on reduction in the velocity fluctuations which might be useful for elimination of the vortex shedding and consequent Karman Vortex Street, as well as Vortex Induced Vibrations.

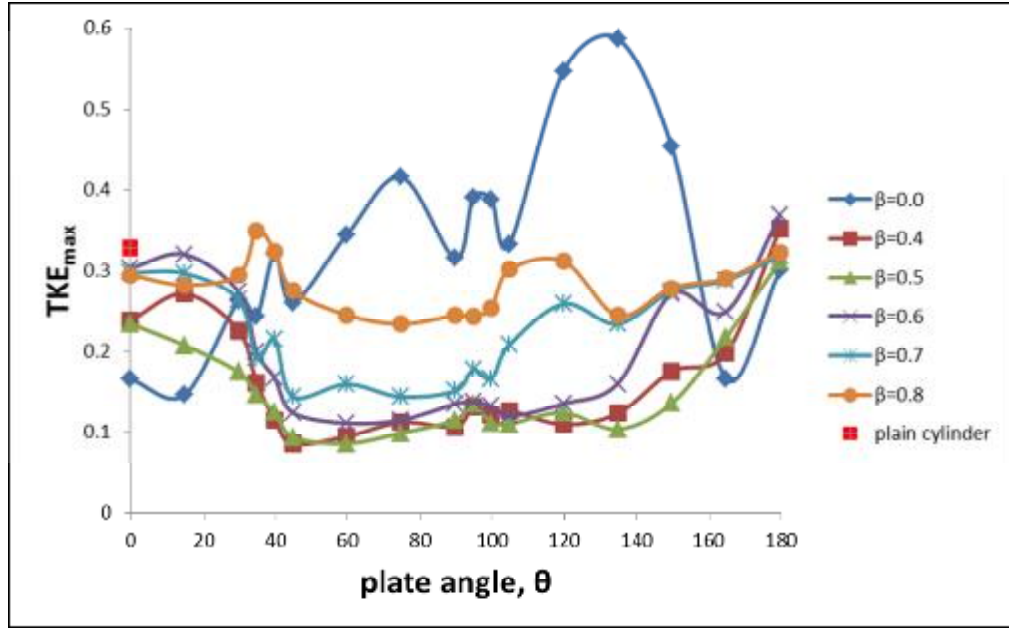


Figure 4.7. The effect of plate angle, θ on the peak values of turbulent kinetic energy, TKE_{max} for various porosities

4.1.2. Reynolds Shear Stress

Another impressive way to analyze the amount of velocity fluctuations in the near wake of the cylinder is the use of Reynolds Stress correlations. Within this context, Reynolds Shear Stresses (RSS) evaluated from PIV data are presented in this section which was calculated by the multiplication of velocity fluctuations in streamwise (u') and spanwise (v') directions. Taking the average of the multiplications gives the time- averaged Reynolds Shear Stresses, $\langle \overline{u'v'} \rangle$. The obtained values are divided by the depth averaged free stream velocity, U^2 in order to make better comparisons. In the physical point of view, the Reynolds stresses, $\langle \overline{u'v'} \rangle$ define the amount of momentum transferred from one layer of the fluid to another layer by the effect of velocity fluctuations in all directions. Thus, the greater amount of this momentum transfer means that the flow in that region is highly turbulent and complex, however the fluid layers are being alternately changed by each other by the time. Beyond these physical realities, it can be concluded that if the amount of RSS is high in the wake of a cylinder then the vortex formation and unsteady shedding of these vortices are inevitable downstream of the cylinder.

Therefore, reducing the RSS in the wake of a cylinder has great importance in suppression of vortex shedding and consequent effects; such as Karman Vortex Street and vortex induced vibrations (VIV). Following figures in this part of the study show the effect of porosity and plate angle on the Reynolds shear stress correlations for all cases, as well as the natural cylinder. Here, the solid and dashed lines represent the positive and negative concentrations, respectively. Throughout the figures, the minimum of positive and maximum of negative values of RSS contours are, +0.01 and -0.01, respectively while the increment is taken as 0.01.

The results of $\beta=0.0$ (solid plate) are shown on Figure 4.8 in which a pair of small scale clusters of RSS exists just behind the cylinder base. Large scale RSS cluster is formed in downstream direction presenting the velocity fluctuations caused by the existence of unsteady vortex shedding. The reduction of these RSS correlations means that the attenuation of vortex shedding is achieved, whereas the elimination of these large scale RSS correlations means the complete suppression of vortex shedding is occurred downstream of the cylinder. Within this context, as the cases of $\theta=0^\circ$ and 15° reduce the concentrations of RSS values compared with the natural cylinder case, it can be concluded that these plate angles are effective on suppressing the vortex shedding in the near wake of the cylinder. Further increase in the plate angle, i.e. the case of $\theta \geq 30^\circ$ increases the vortex formation length and the width of the wake region due to the increased area of body cross section. For $30^\circ \leq \theta \leq 150^\circ$, the concentrations are slightly increased which means the velocity fluctuations get higher values in the whole wake of the cylinder. Furthermore, since the solid plate blocks the necessary amount of fluid flow from the lower shoulder of the cylinder into the wake, as well as the corresponding generation of the lower shear layer, the negative RSS contours cover the wake region dominantly, especially for $60^\circ \leq \theta \leq 120^\circ$ cases. As evaluated and seen by the TKE results in previous section, the effect of $\theta=165^\circ$ plate angle substantially reduces the RSS concentrations by shifting the locations of positive and negative clusters. The case of $\theta=0^\circ$ forms a similar flow structure and RSS distribution as evaluated by the natural cylinder case. As a result, the cases of $\theta=0^\circ$ and 15° are effective on controlling the vortex development for the solid plate; $\beta=0.0$.

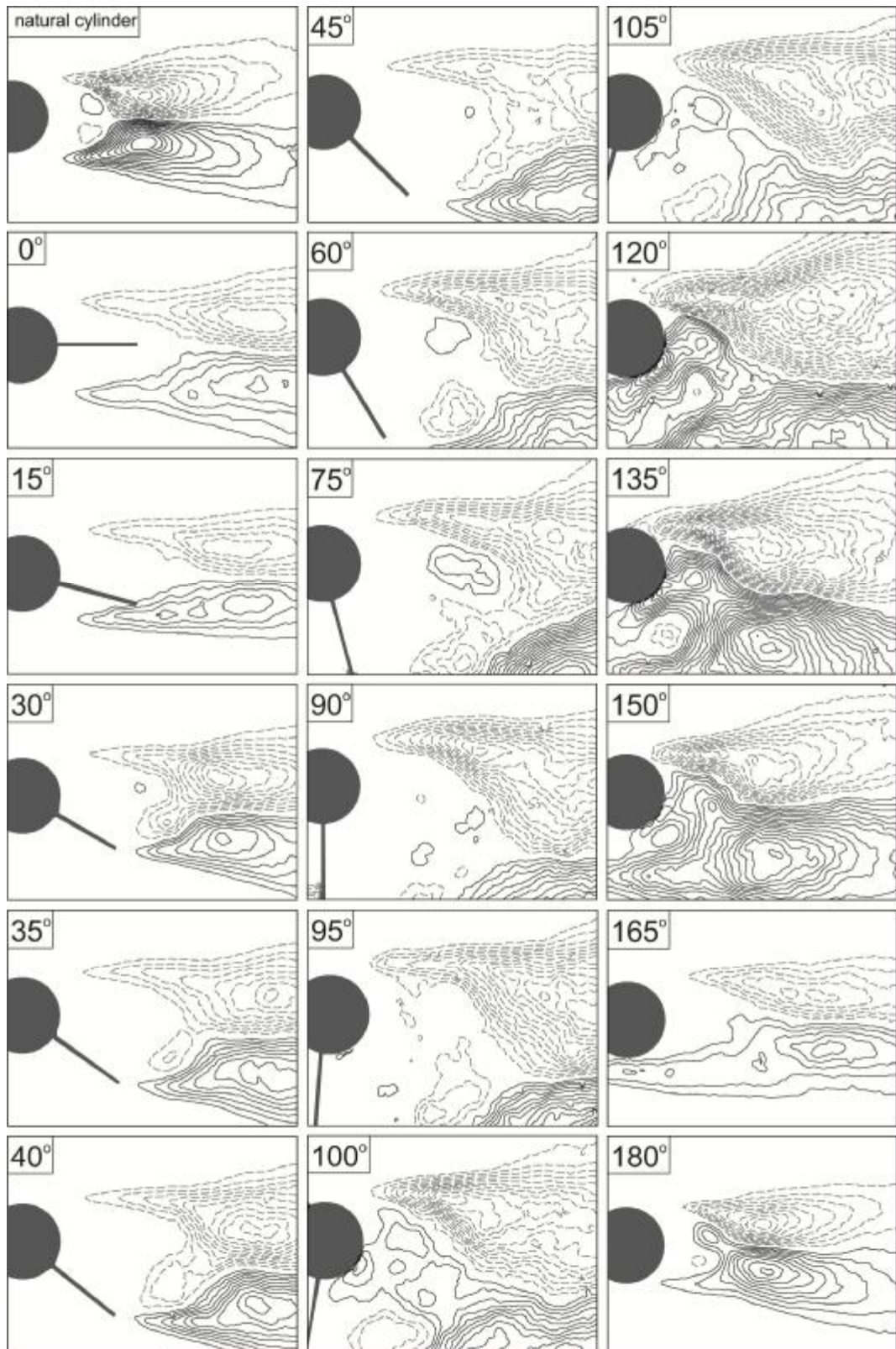


Figure 4.8. Time- averaged dimensionless Reynolds Shear Stress (RSS) correlations for the natural cylinder and all cases of $\beta=0.0$ (solid plate)

It is expected that the use of a permeable plate would significantly affect the RSS correlations especially for larger flow angles since it directly interacts with the shear layer, thus the amount of momentum transfer from the lower shear layer into the wake region is restricted (not blocked as evaluated for the solid plate ($\beta=0.0$) case). The effect of using a permeable plate having $\beta=0.4$ porosity is shown in Figure 4.9. It is clearly seen that the use of a permeable plate instead of a solid one is advantageous, especially for larger plate angles. For the plate angles of $0^\circ \leq \theta \leq 30^\circ$, the concentrations of RSS slightly reduce and the location of the vortex formation is gradually shifted in downstream direction. Up to this point, the formation of both upper and lower shear layers is obviously seen similar to the natural cylinder case. Increasing the plate angle directly affects the lower shear layer, hence the location of lower shear layer moves in downstream direction by losing its intensity for $35^\circ \leq \theta \leq 45^\circ$. At $\theta=60^\circ$ plate angle, the lower shear layer disappeared in the wake region by dramatically reducing the RSS intensity on upper shear layer. The effect of permeable plate on reduction of RSS correlations exists for the plate angles of $40^\circ \leq \theta \leq 150^\circ$ where this interval could be addressed of effective suppression of vortex shedding, i.e. elimination of the lower shear layer means that there is no vortex roll-up from that layer into the wake, thus no momentum transfer and vortex interaction, finally there exists the complete elimination of vortex shedding. Therefore it can be concluded that the use of a permeable plate of $\beta=0.4$ porosity for plate angles of $35^\circ \leq \theta \leq 150^\circ$ is an effective way of controlling the flow structure downstream of a cylinder. Furthermore the latter cases of $\theta=165^\circ$ and 180° shows that the contours of RSS increase as a result of ineffectiveness of the permeable plate on flow control.

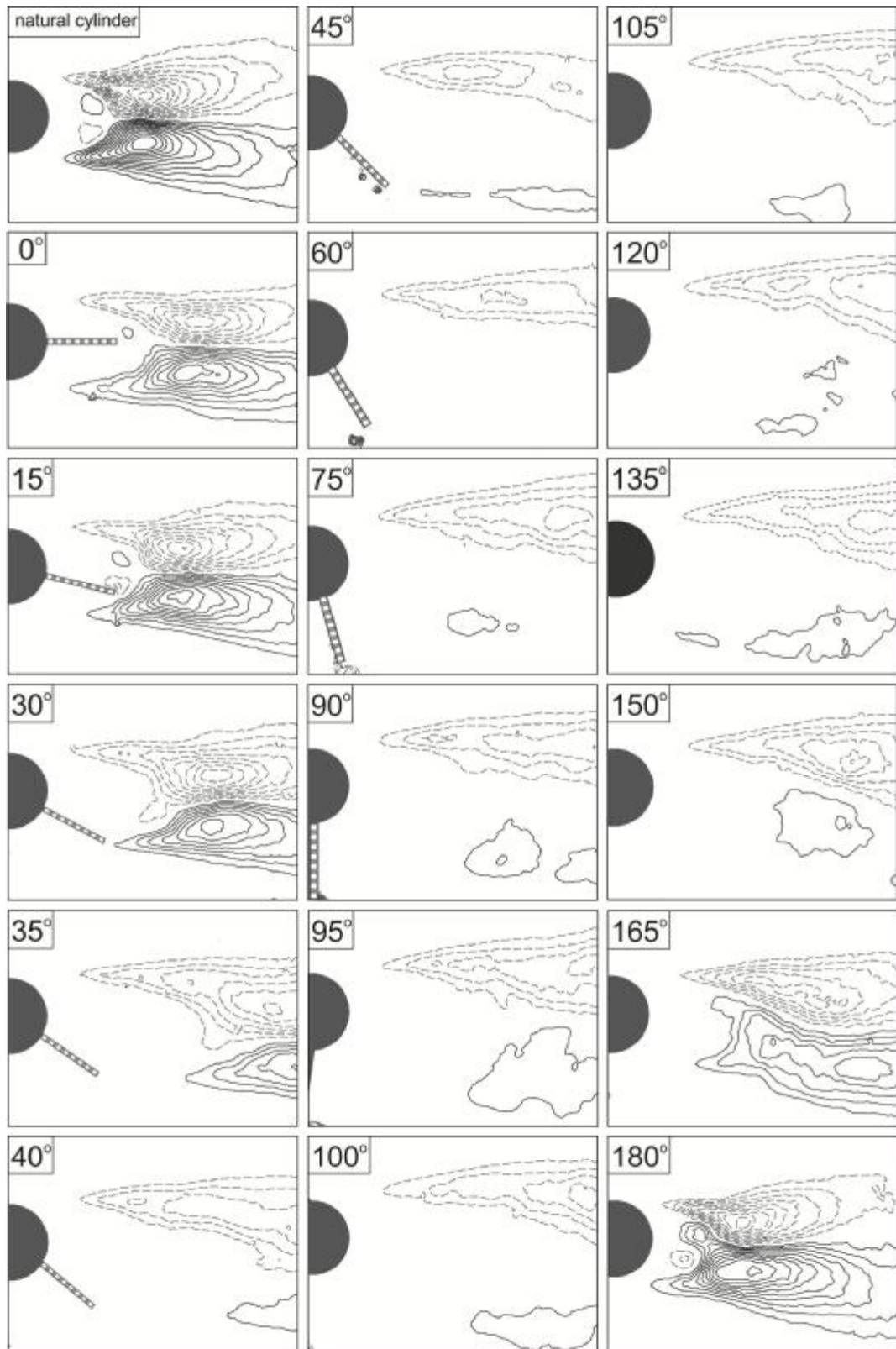


Figure 4.9. Time- averaged dimensionless Reynolds Shear Stress (RSS) correlations for the natural cylinder and all cases of $\beta=0.4$

The results of $\beta=0.5$ case is almost similar with the RSS concentrations of $\beta=0.4$ case as clearly seen by comparing Figures 4.9 and 4.10, where Figure 4.10 presents the RSS correlations for $\beta=0.5$ case for all plate angles as well as the natural cylinder. In each case, the most effective plate angles on reduction of RSS concentrations are $35^\circ \leq \theta \leq 150^\circ$. However, the complete elimination of the lower shear layer is achieved of $\theta=45^\circ$ and 60° plate angles. The same conclusion could be done for the $\beta=0.5$ case, i.e. the effective interval of plate angles; $35^\circ \leq \theta \leq 150^\circ$ would be suggested as an impressive way of effective control of vortex shedding.

Figure 4.11 presents the RSS correlations for all plate angles of $\beta=0.6$ case. As porosity increases, amount of momentum passing through the wire meshes increases, too. Therefore the flow structure and the intensity are expected to be significantly changed at a distinct value of the increasing porosity, which may be called as transition point of flow control. First look on the Figure 4.11 brings out the idea of transition process of porosity from being effective to ineffective on flow control. For the cases of $\theta=0^\circ$ and 15° , the contours of RSS are similar with those for the natural case even there exists a slight reduction in RSS values. Furthermore, for $\theta=0^\circ$ plate angle the small-scale clusters of RSS still exist by increasing in intensity unlikely with the lower porosities. The permeable plate of $\beta=0.6$ shows its effect on flow structure and velocity fluctuations after the plate angle of $\theta=30^\circ$ where the positive and negative clusters of RSS contours take an asymmetrical shape downstream of the cylinder. For $\theta=35^\circ$ and 40° plate angles, a new structure takes place in which there exits two different positive clusters of RSS contours on lower shear layer of the cylinder, namely A and B as shown on Figure 4.11. The cluster A is generated by the interaction of flow through the meshes and the upper shear layer where the cluster B is formed by the flow from trailing edge of the permeable plate. The cluster B disappears by further increase in plate angle, even the existence of both positive and negative RSS contours still valid in the measured flow field of view. The effective interval of plate angle for this porosity ratio seems to be $45^\circ \leq \theta \leq 135^\circ$ since the concentrations of RSS are smaller compared to the letter cases, however further detailed analysis needed. For the other plate angles, the values of RSS are still lower than that of the natural cylinder case.

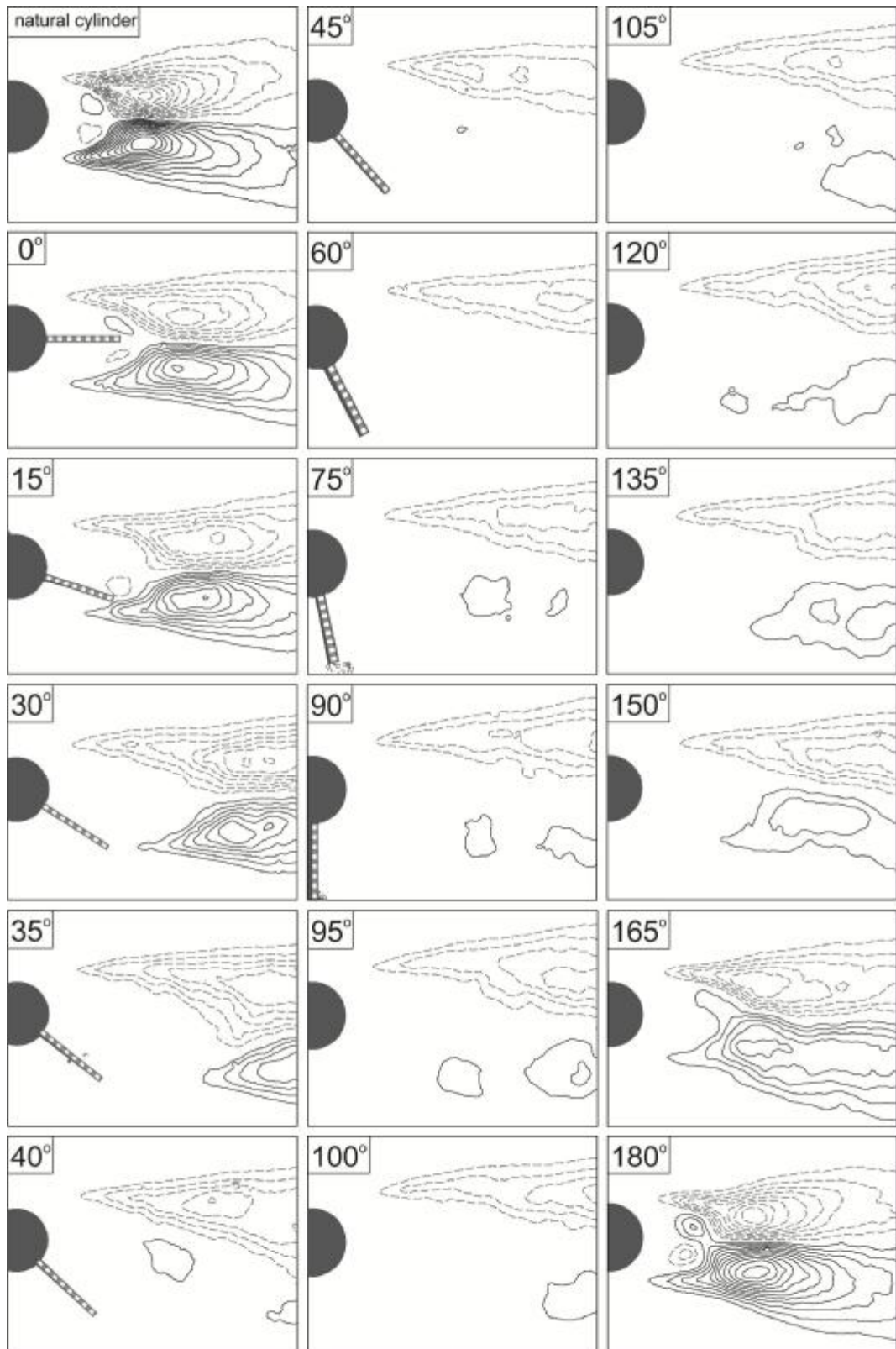


Figure 4.10. Time- averaged dimensionless Reynolds Shear Stress (RSS) correlations for the natural cylinder and all cases of $\beta=0.5$

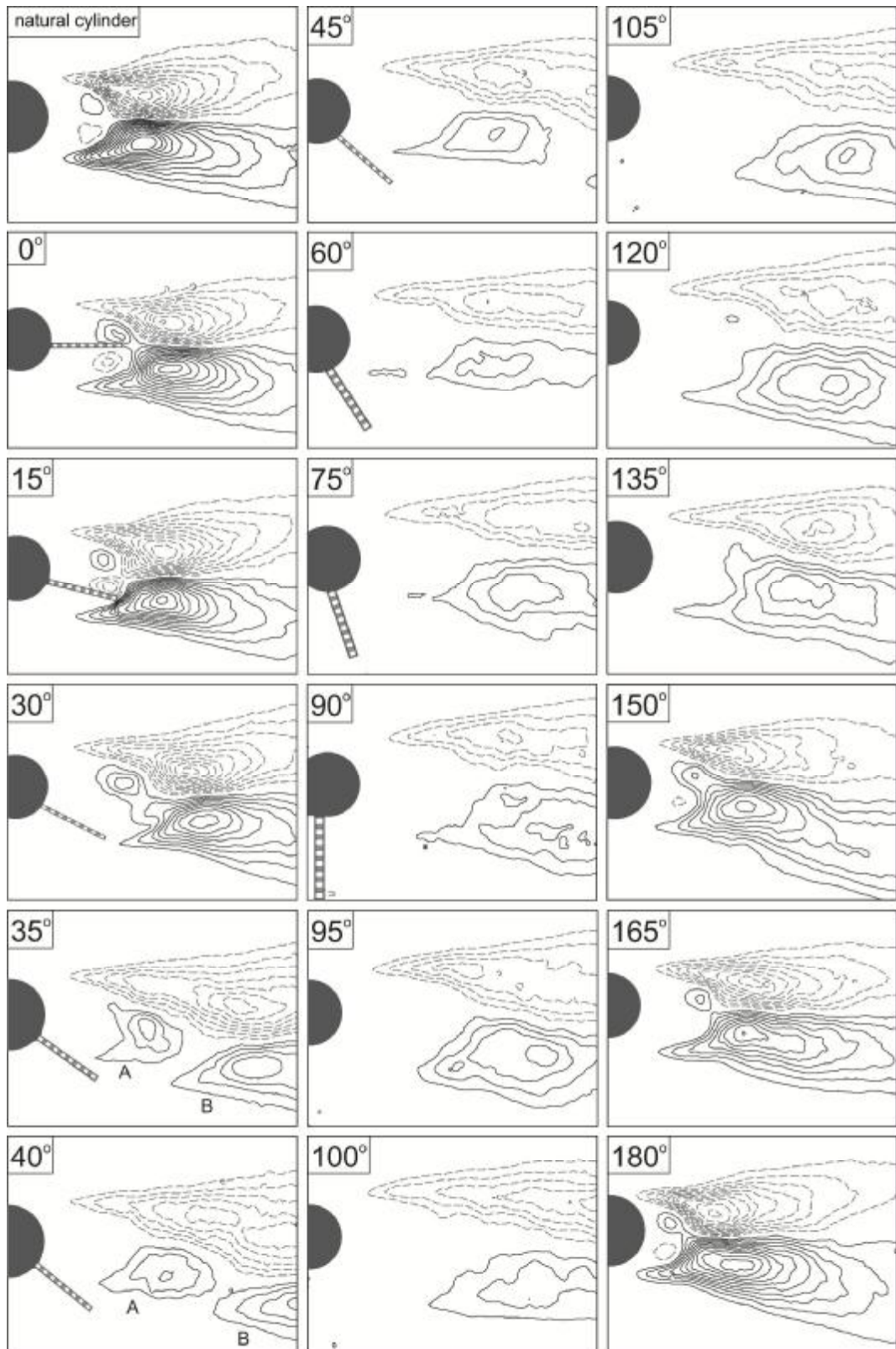


Figure 4.11. Time- averaged dimensionless Reynolds Shear Stress (RSS) correlations for the natural cylinder and all cases of $\beta=0.6$

Figure 4.12 shows the effect of $\beta=0.7$ porosity on the concentrations of RSS for all plate angles and the natural cylinder case. First of all, Figure 4.12 demonstrates that the previous porosity value, $\beta=0.6$ is the transition point of permeable plates where the shape and concentration of RSS values changes dramatically. It is obviously seen from the contours of RSS that the permeable plate of $\beta=0.7$ loses its effectiveness on reducing the velocity fluctuations in the near wake of the cylinder. Nevertheless, for $45^\circ \leq \theta \leq 105^\circ$ plate angles, the values of RSS are still significantly lower in concentrations when compared with the natural cylinder case. It should also be pointed out that two separate clusters of RSS contours, namely A and B as observed for $\beta=0.6$ case in Figure 4.11 are still distinguishable for the plate angles of $\theta=35^\circ$ and 40° , however they are closer to each other compared with the $\beta=0.6$ case.

It's expected that the permeable plate would completely lose its effectiveness by further increase in porosity. This is an unavoidable result since the fluid would pass directly through the meshes, instead of being injected. Since the injection does not exist anymore, only slight changes are waited to be taken into consideration for $\beta=0.8$ cases which are presented in Figure 4.13. It is obviously seen by the RSS contours that only slight reductions in the values of RSS are evaluated for plate angles of $60^\circ \leq \theta \leq 100^\circ$. The reasoning of this situation relies on the results of $15^\circ \leq \theta \leq 40^\circ$ cases in which there exist the small-scale RSS concentrations just behind the cylinder base similar with the natural case, as well as the large scale clusters on both sides of the plate (upstream- downstream) since the amount of mesh sizes are greatly increased and the fluid directly passes through the meshes by minimum intervention of wires on the lower shear layer. Therefore, this porosity ratio could not be suggested in effective control of the flow downstream of the cylinder.

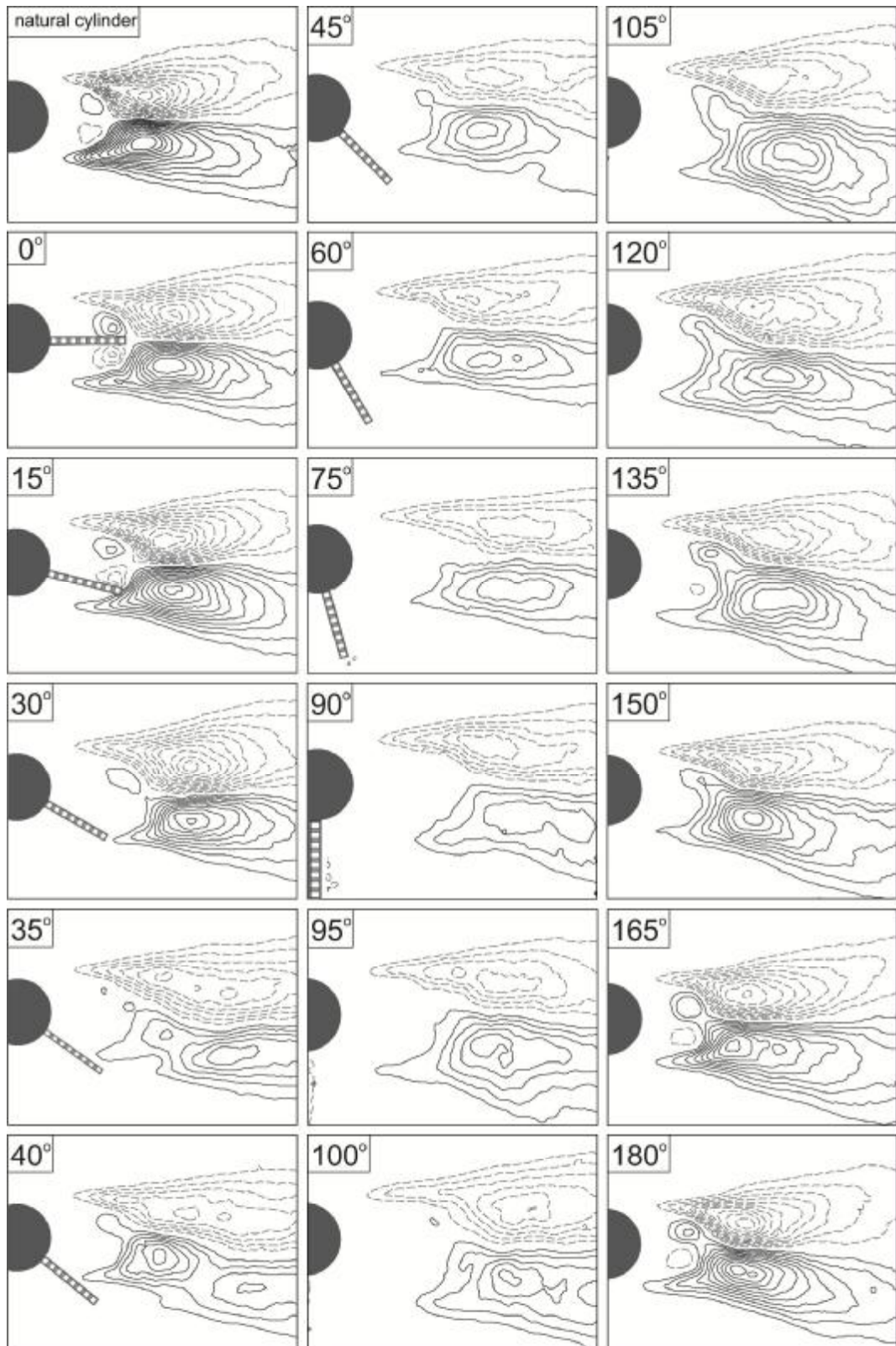


Figure 4.12. Time- averaged dimensionless Reynolds Shear Stress (RSS) correlations for the natural cylinder and all cases of $\beta=0.7$

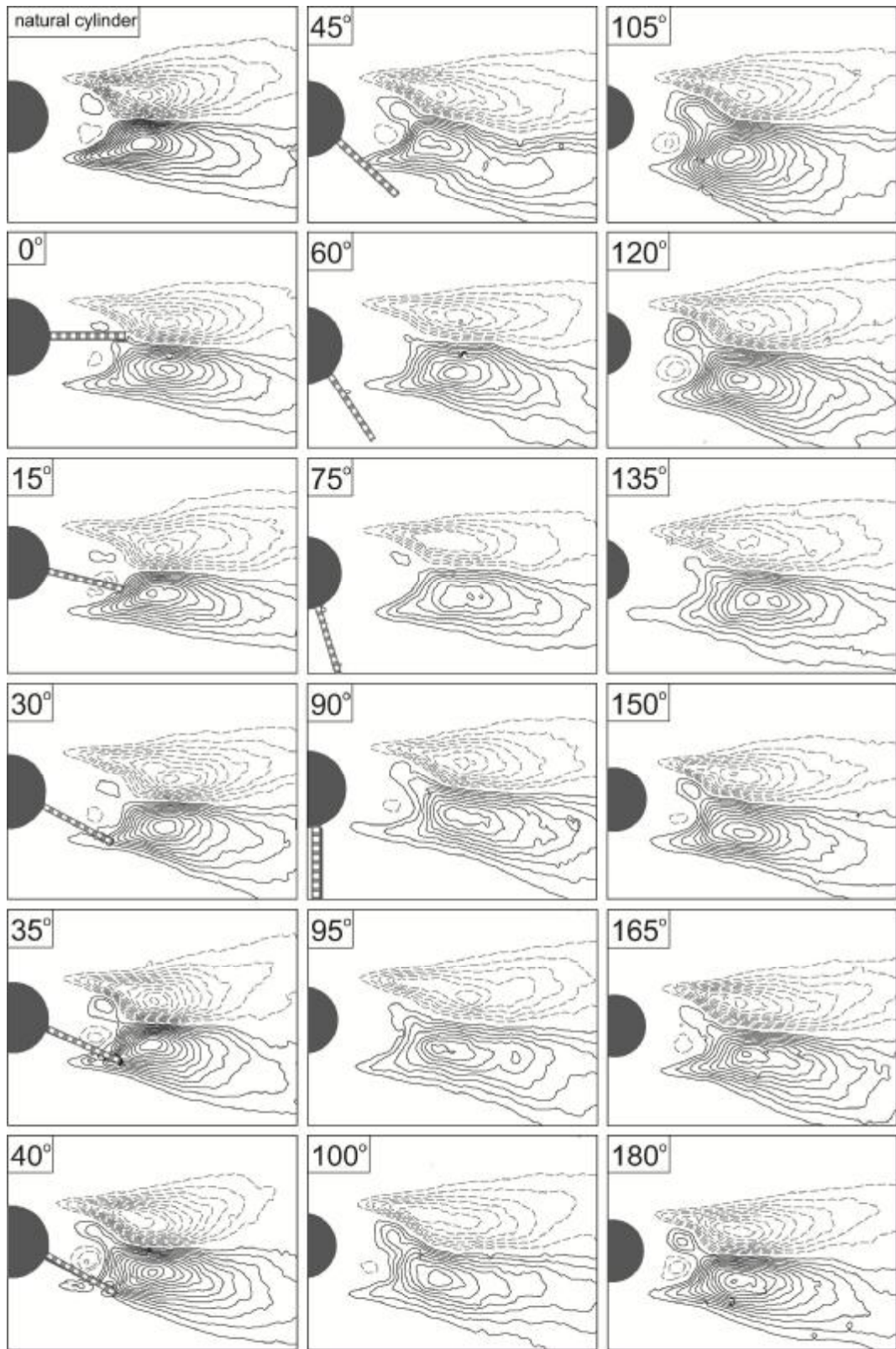


Figure 4.13. Time- averaged dimensionless Reynolds Shear Stress (RSS) correlations for the natural cylinder and all cases of $\beta=0.8$

4.2. The Effect of Plate Length on Flow Control

In order to understand the effect of plate length on flow control, the lengths of $L/d=1.0$ and $L/d=0.4$ are compared for the porosity value of $\beta=0.6$ using Reynolds Shear Stress correlations in Figures 4.11 and 4.14, respectively.

It is obviously seen by the Figure 4.14 that the shorter permeable plate results in higher RSS concentrations in the near wake compared to the cases of $L/d=1.0$, especially for lower plate angles of $0^\circ \leq \theta \leq 40^\circ$. For larger plate angles the concentrations move in downstream direction as clear for plate angle of $\theta=45^\circ$, however the intensities of RSS is still have high values.

For $\theta=60^\circ$, the concentrations dramatically decreases and lower shear layer is separated to two independent clusters which were obtained at $\theta=35^\circ$ for $L/d=1.0$. This delay in control effect of permeable plate is due to the ineffectiveness of plate length to interfere the lower shear layer, i.e. shorter the plate length interacts with the lower shear layer at larger plate angles. Increasing the plate angle further results in reduced magnitude clusters on the lower shear layer, however the RSS concentrations still have high intensity on the upper shear layer. This condition is valid for $90^\circ \leq \theta \leq 105^\circ$ where the fluctuations slightly increase by increasing plate angle and the permeable plate loses its effectiveness on reducing the RSS concentrations for $120^\circ \leq \theta \leq 180^\circ$.

It can be concluded that the permeable plate having a length of $L/d=0.4$ is effective on controlling the unsteady vortical structure in the wake of a cylinder for plate angles of $60^\circ \leq \theta \leq 75^\circ$ where the effective interval for $L/d=1.0$ was found to be $45^\circ \leq \theta \leq 135^\circ$. Since the effective interval of plate angle on flow control for $L/d=0.4$ is very limited, the plate length of $L/d=1.0$ was used throughout this study.

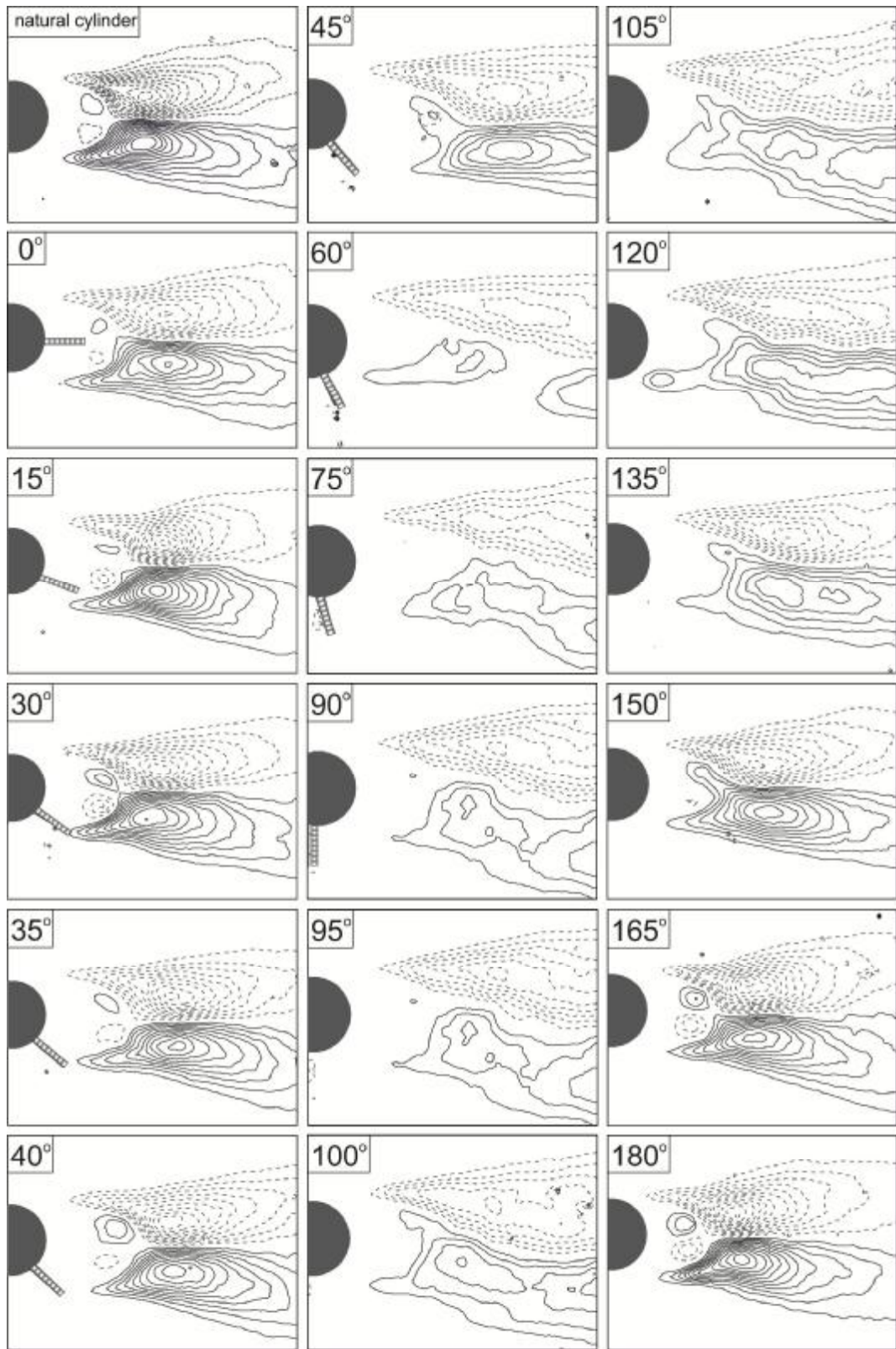


Figure 4.14. Time- averaged dimensionless Reynolds Shear Stress (RSS) correlations for the natural cylinder and all cases of $\beta=0.6$ for $L/d=0.4$

4.3. Further Analysis on Effective Ranges of Flow Control

Up to now, all porosity and plate angle cases were presented by using the turbulent kinetic energy and Reynolds shear stress values and effective range of flow control mechanisms on reducing the velocity fluctuations were defined. It was found that the solid plate is not so useful on reducing the velocity fluctuations except for some special cases ($\theta=0^\circ, 15^\circ, 165^\circ$). Using a permeable plate was found to be more effective on flow control and the most effective porosity values were determined to be $\beta=0.4$ and 0.5 for a wide range of plate angle; $35^\circ \leq \theta \leq 150^\circ$, however this effective interval gets narrower for higher porosity values, for instance it is $45^\circ \leq \theta \leq 135^\circ$ for $\beta=0.6$; $45^\circ \leq \theta \leq 135^\circ$ for $\beta=0.7$ and $60^\circ \leq \theta \leq 100^\circ$ for $\beta=0.8$. According to these findings, it would be more appropriate to analyze the results in details for the effective cases. On this section of the work, the porosities of $0.4 \leq \beta \leq 0.7$ and the plate angles of $35^\circ \leq \theta \leq 150^\circ$ are going to be analyzed in details by using the time- averaged streamline topologies, $\langle \psi \rangle$, vorticity contours, $\langle w \rangle$, streamwise velocity contours, $\langle u \rangle$ and velocity vector fields, $\langle V \rangle$.

Figure 4.15 presents the results obtained for the natural cylinder which are in good agreement with the previous works, i.e. a recirculation region appears in the near wake of the cylinder with formations of two well- defined foci, namely F_1 and F_2 which are almost symmetrical with respect to the centerline and generated at $0.8d$ away from the base of the cylinder. The fluid flow gets zero velocity at a position $1.2d$ away from the cylinder base which is called as saddle point designated by S_1 in the figure. The location of saddle point, S_1 also defines the vortex formation length, i.e. the vortex formation length is defined as the distance from the base of the cylinder to the saddle point on the centerline (Aydin et al., 2010). Therefore, it can be stated that the vortex formation length for the natural cylinder is evaluated as $1.2d$. The magnitudes of time-averaged vorticity contours and streamwise velocity contours are presented by legends shown at right side of the figure. The vorticity contours are symmetrical and get closer to each other in the wake region which is an indication of interaction of the vortices and consequent vortex shedding. It is seen by the streamwise velocity contours that the contours have negative values in the wake

region at a location approximately same with the locations of focal points. Besides, the magnitudes of velocity vectors get higher values just around this backflow region. A high amount of reduction in the streamwise flow takes place in the wake region of the cylinder, as a result of interaction of lower and upper shear layers due to the unsteady vortex shedding.

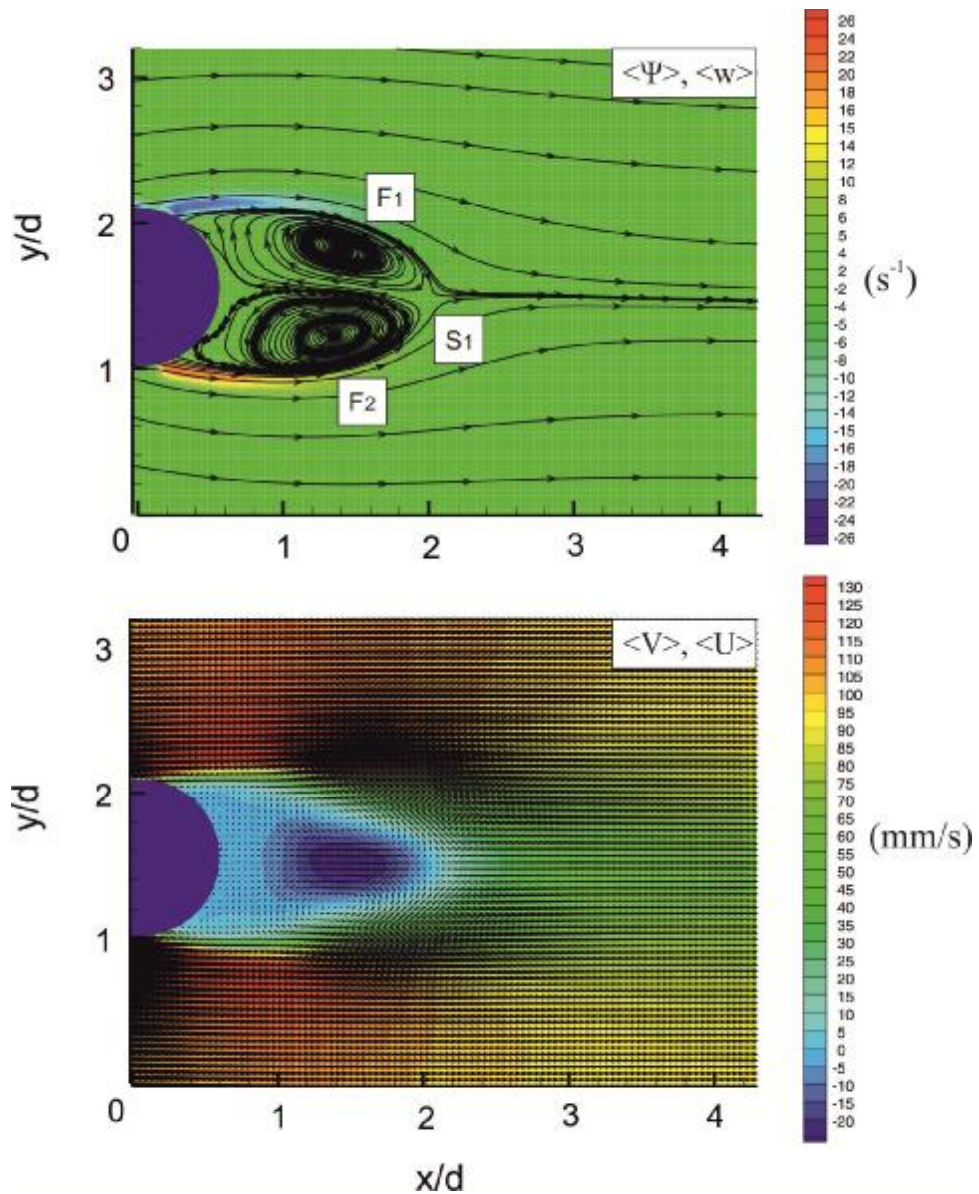


Figure 4. 15. The results of natural cylinder: Time averaged streamline topology, $\langle \Psi \rangle$ superposed with the vorticity contours, $\langle w \rangle$ and time- averaged velocity vector field, $\langle V \rangle$ superposed with streamwise velocity contours, $\langle u \rangle$

These are the well-known findings for the flow structure around a circular cylinder which will be used as references in order to understand the effect of permeable plate on flow control cases presented in future parts of the study.

Figure 4.16a presents the time averaged streamline topology, $\langle \Psi \rangle$ superposed with the vorticity contours, $\langle w \rangle$ for $\beta=0.4$ and 0.5 cases for the plate angles of $35^\circ \leq \theta \leq 90^\circ$. The main mechanism of controlling the flow is to prevent or attenuate the velocity fluctuations caused by interaction of shear layers in the near wake. Within this context the best way of observing the shear layer interaction is to analyze the vorticity contours. It is obviously seen in Figure 4.16a that the lower shear layer evolved from the lower shoulder of the cylinder encounters with the permeable plate while it is being elongated in the downstream direction in all cases. However, a new shear layer is generated from the trailing edge of the permeable plate since the sizes of the meshes are narrow for $\beta=0.4$ and 0.5 porosities. As a result of this situation, the recirculation region of the cylinder and corresponding location of the generated foci and saddle points are significantly affected compared to the natural cylinder. The focal points get an asymmetric shape and the vortex formation region elongates in downstream direction. Furthermore the width of the wake slightly increases as a result of increased cross-sectional area of the cylinder- permeable plate arrangement. In general trend, both porosity values show a similar behavior on the flow structure except some differences. It should be noted that in all cases both shear layers, especially the upper layer elongate in downstream direction compared to the natural cylinder case as depicted in Figure 4.15.

For plate angle of $\theta=35^\circ$ two porosity value the flow structures are pretty much the same, further increase in the plate angle to $\theta=40^\circ$ narrows the wake width and almost $2.1d$ difference between the focal points observed for $\beta=0.4$, however for $\beta=0.5$ the focal points get closer to each other compared to the $\theta=35^\circ$ plate angle.

Most reduction in the wake width exists for plate angle of $\theta=60^\circ$ which is the most effective case on controlling the vortex shedding. It should be noted at this point that two independent recirculation region exist for the plate angle of $\theta=60^\circ$ at $\beta=0.4$ porosity value, one is formed just behind the rear stagnation point of the cylinder with two focal points, and the other is at farther downstream regions

approximately at $4d$ away from the flow field. This is due to the small mesh sizes of the permeable plate where the trailing edge of the cylinder is solid enough to generate a shear layer and consequently a new wake in farther regions of the trailing edge. This situation might be valid for the larger plate angles which are probably on the out-of plane of the current flow field.

For both porosity ratios, it should also be implied that a small- magnitude negative vorticity layer is formed in addition to the positive shear layer on the trailing edge of the permeable plate, especially for plate angles of $40^\circ \leq \theta \leq 60^\circ$. This small- magnitude negative shear layer plays an important role on flow control as well, i.e. the amount of fluid entrainment from the positive shear layer into the wake region is limited or prevented by existence of this negative shear layer. However, the formation of a new recirculation region as specified by $\theta=60^\circ$ at $\beta=0.4$ case is due to the interaction of shear layers on trailing edge of the permeable plate. For plate angles of $\theta=75^\circ$ and 90° , the recirculation region grows, focal points get closer to each other in both directions and the vorticity layers consist their stability as evaluated for smaller plate angles.

The larger effective plate angles are presented in Figure 4.16b where the interval of for $95^\circ \leq \theta \leq 150^\circ$ is shown. In all cases, it is impossible to observe interaction of shear layers since both upper and lower shear layers grow away from each other for all cases. The plate angles of for $95^\circ \leq \theta \leq 105^\circ$ can be called as the special cases since they coincide with the occurrence of separation point on the lower shear layer of the cylinder. Therefore, the permeable plate directly affects the possible separation point for these plate angles. Here the length and width of the wake are increased and recirculation region covers almost the whole flow field, however it should also be pointed out that more than two focal points are generated for $\theta=100^\circ$ plate angle of $\beta=0.4$ and $\theta=100^\circ$ plate angle of $\beta=0.5$ which might be due to the intervention on separation point.

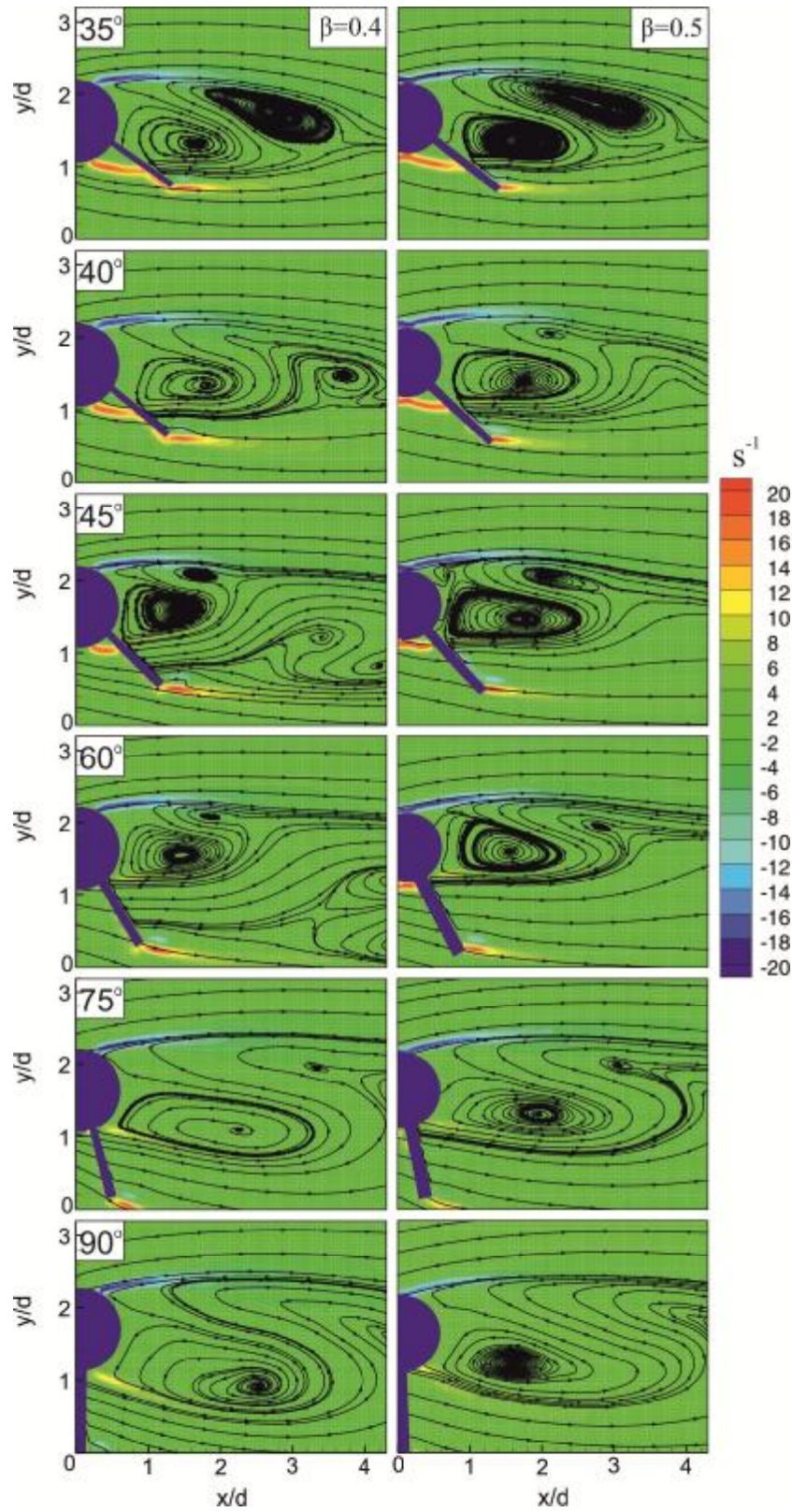


Figure 4. 16a. Time averaged streamline topology, $\langle \Psi \rangle$ superposed with the vorticity contours, $\langle \omega \rangle$ for $35^\circ \leq \theta \leq 90^\circ$ plate angles of $\beta = 0.4$ and 0.5

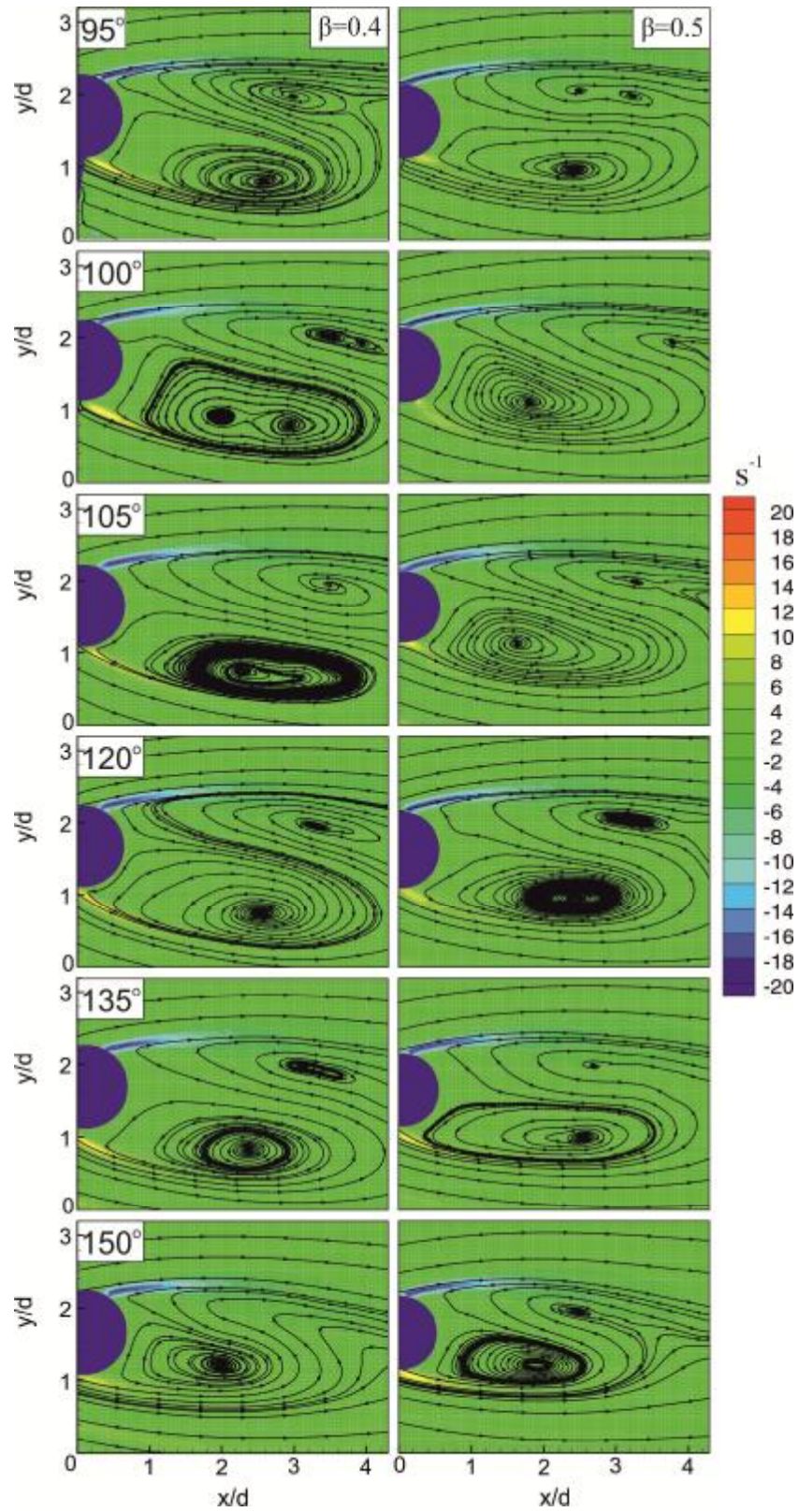


Figure 4. 16.b. Time averaged streamline topology, $\langle \Psi \rangle$ superposed with the vorticity contours, $\langle w \rangle$ for $95^\circ \leq \theta \leq 150^\circ$ plate angles of $\beta=0.4$ and 0.5

As explained before, increasing the porosity would result in larger mesh sizes, hence larger amount of fluid emanate from the meshes into the wake region, especially for smaller plate angles. Therefore, it is important to see the effect of greater porosities on flow control by means of shear layer interactions and streamlines. The time averaged streamline topology, $\langle \Psi \rangle$ superposed with the vorticity contours, $\langle w \rangle$ for $\beta=0.6$ and 0.7 cases for plate angles of $35^\circ \leq \theta \leq 90^\circ$ are shown in Figure 4.17a. Similar with the results of lower porosity values, it is obviously seen that the shear layers elongate in downstream direction where the upper shear layer is longer than the lower one for all plate angles of $\beta=0.6$. The recirculation region for $\beta=0.6$ slightly increases by increasing plate angle by generating two well- defined foci and a saddle point in all plate angles. Although the flow structures are similar with each other for both porosity values, increasing the porosity to $\beta=0.7$ dramatically increases the vorticity magnitudes as seen by the vorticity contour legend. Almost 50% increase in the vorticity magnitudes takes place due to the increased open area of the permeable plates, i.e. sizes of the meshes.

Similar with the lower porosities, a new shear layer is formed from the trailing edge of the plate for $\beta=0.6$ cases, however no counter shear (negative) is developed which was evaluated for $40^\circ \leq \theta \leq 60^\circ$ of $\beta=0.4$ and 0.5 porosities. This could be an indication to the transition of the permeable plate to less porous to high porous where the porosity of $\beta=0.6$ is the transition point. According to this finding, it is assumed that further increase in porosity would adversely affect the flow control as evaluated in TKE and RSS concentrations on previous sections.

The results of $\beta=0.7$ clearly shows that the existence of permeable plate is not effective on prevention of interaction of shear layers since the generated shear layers get closer to each other although their length is elongated. Furthermore, the lower shear layer which is formed by passing through the plate has higher vorticity magnitudes and the following interaction of the layers is unavoidable in the case of $\beta=0.7$.

Figure 4.17b provides the time averaged streamline topology, $\langle \Psi \rangle$ superposed with the vorticity contours, $\langle w \rangle$ for $\beta=0.6$ and 0.7 cases for plate angles of $95^\circ \leq \theta \leq 150^\circ$ to understand the effects of larger angles. It is obviously seen for the

$\beta=0.6$ that the length of the recirculation gradually increases up to $\theta=100^\circ$ which tells us the permeable plate is still effective on controlling the flow, however it gradually diminishes for $\theta \geq 105^\circ$ by decreasing the width of the wake. Furthermore the shear layers get closer to each other for $\theta=135^\circ$ and 150° which is an indication of interaction of the vortices and unsteady vortex shedding. This is the case also for $95^\circ \leq \theta \leq 150^\circ$, so as all cases in Figure 4.17b. Beginning from $\theta=100^\circ$ to higher plate angles, the vortex formation length decreases gradually with narrower wakes. It can be concluded that the increase in velocity fluctuations defined in previous sections is due to this shear layer interaction which comes from the ineffectiveness of permeable plate to prevent the interaction of shear layers because of increased open area, especially for $\beta=0.7$ and 0.8 .

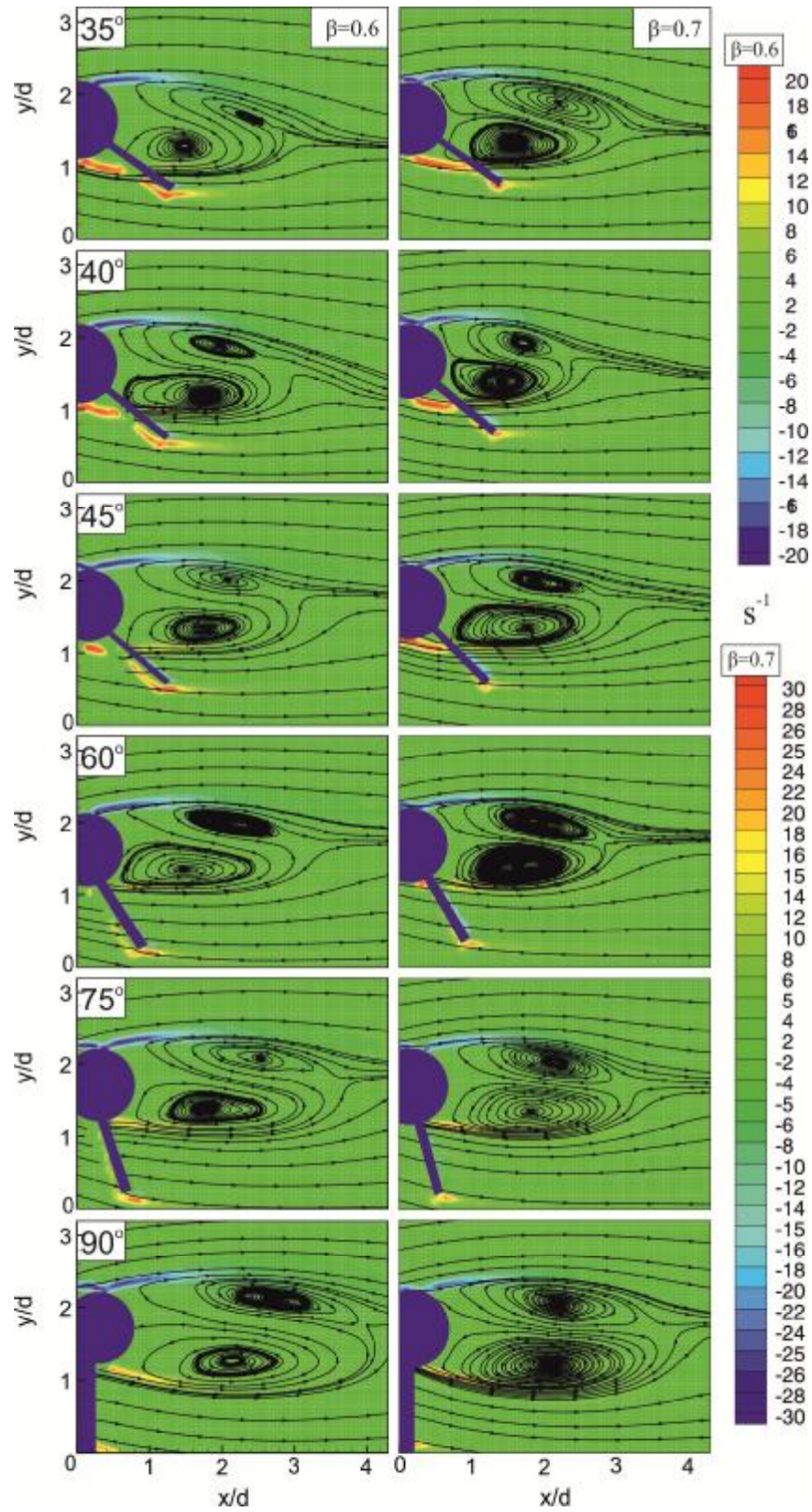


Figure 4. 17a. Time averaged streamline topology, $\langle \Psi \rangle$ superposed with the vorticity contours, $\langle \omega \rangle$ for $35^\circ \leq \theta \leq 90^\circ$ plate angles of $\beta=0.6$ and 0.7

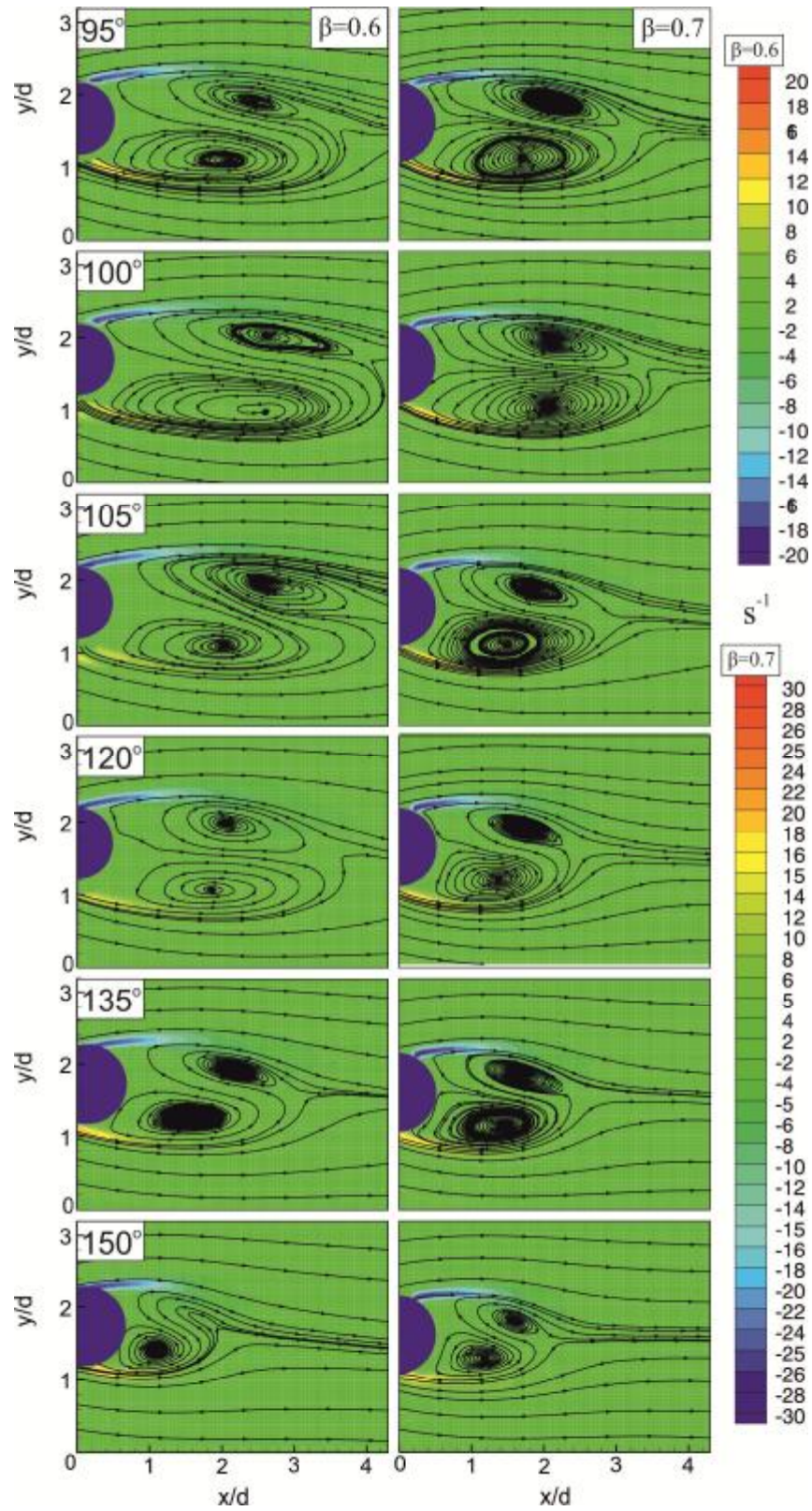


Figure 4. 17b. Time averaged streamline topology, $\langle \Psi \rangle$ superposed with the vorticity contours, $\langle w \rangle$ for $95^\circ \leq \theta \leq 150^\circ$ plate angles of $\beta=0.6$ and 0.7

It can be concluded from Figures 4.16 and 4.17 that the flow structure downstream of the cylinder is significantly affected by the permeable plate for all angles. The vortex formation length, evaluated by measuring the distance between saddle point and the cylinder base is an important factor in which an increase in the vortex formation length would be effective on vortex shedding control since the vortical structures generated on further regions of the wake which means the cylinder is less affected by the unsteady flow structure. Vortex formation lengths for the cases of $\beta=0.4$ and 0.5 cannot be calculated since there exists a complex flow structure for almost all plate angles including more than one focal and saddle points as can be seen on Figures 4.16a and 4.16b. Therefore, the vortex formation lengths are calculated for $\beta=0.6$ and 0.7 cases and presented as a function of plate angle, θ in Figure 4.18. First of all, it is clear from the figure that all cases are effective on increasing the vortex formation length compared to the natural cylinder which was calculated as $1.2d$. It is also seen from Figure 4.18 that the porosity value of $\beta=0.6$ is more effective on increasing the vortex formation length compared to the $\beta=0.7$ although the trends are similar. The most effective plate angle for $\beta=0.6$ is found to be $3.56d$ (nearly two times larger than that of the natural cylinder) for $\theta=100^\circ$ to an increase by which coincides with the separation point of the cylinder. For $\beta=0.7$, the values are similar with those of the $\beta=0.6$ case, however the vortex formation length dramatically decreases for $\theta \geq 90^\circ$. This is the case for $\beta=0.6$ for $\theta \geq 105^\circ$ and the values get closer to $1.2d$ which was evaluated for the natural cylinder.

It can be concluded by the results of vortex formation lengths that the porosity values of $\beta=0.6$ and 0.7 are effective on increasing the vortex formation lengths for $35^\circ \leq \theta \leq 135^\circ$ which is an indication of delay of unsteady vortex shedding in the cylinder wake. However, the vortex formation lengths could not be evaluated for the porosities of $\beta=0.4$ and 0.5 due to the complex flow structure which consists various focal and saddle points.

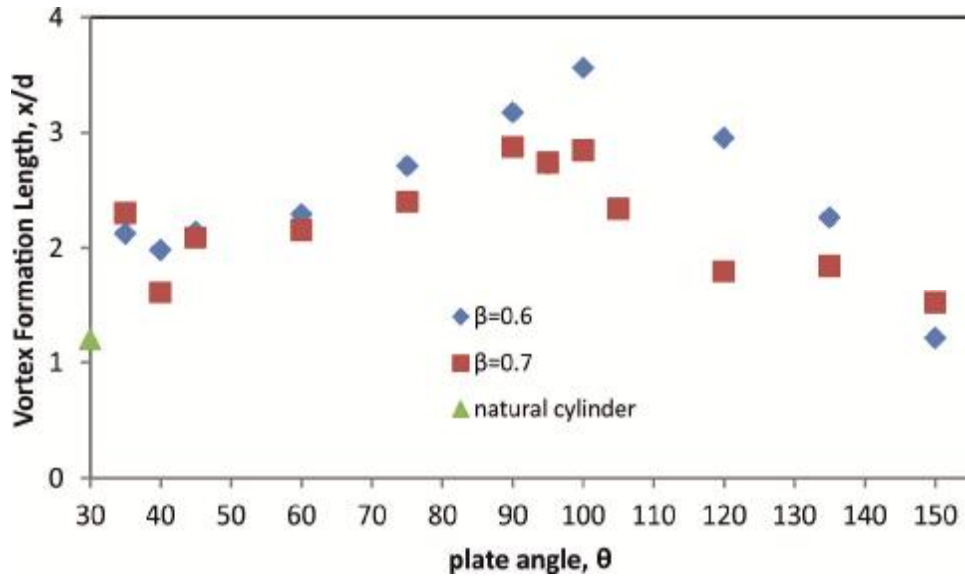


Figure 4.18. Dimensionless vortex formation lengths as a function of plate angles, θ for $\beta=0.6$ and 0.7

Time averaged velocity vector fields, $\langle V \rangle$ superposed with the streamwise velocity contours for $\beta=0.4$ and 0.5 cases are presented in Figures 4.19a and 4.19b, respectively. The amount of shrinking and elongation of the wake region are easily seen by these figures. For all cases, there exists a negative flow region corresponding to the reverse flow as indicated by streamline topologies with significant differences in both locations and magnitudes by varying porosity. The plate angle of $\theta=35^\circ$ for both porosities in Figure 4.19a shows a similar behavior where the negative velocity region covers the cylinder wake for the current field of view. An increase of the plate angle to $\theta=40^\circ$ results in a reduction in the location of negative velocity region while it is being shifted to the upper shear layer. This is due to the larger velocity gradient on the upper side as indicated by the distributions of velocity vectors. For $\beta=0.4$, at plate angles of $\theta=45^\circ$ and 60° , as different than all the other cases, two negative velocity regions are indicated, one is just behind the cylinder base (near to the upper shear layer), and the other is on farther downstream locations along the shear layer formed by the trailing edge. It can be said that the magnitude of negative velocity contours observed in further downstream regions are lowered due to the permeable plate in which the evaluation starts from the trailing edge of the plate. Furthermore, there exists positive velocity gradient in between these two low velocity regions. A

sudden increase in negative values take place while passing through the wire meshes with the range of $75^\circ \leq \theta \leq 90^\circ$ having $\beta=0.5$ porosity where a larger wake exists compared to the other plate angles.

Further increase in plate angle for both porosity increases both the length and width of the cylinder wake, especially for $\beta=0.4$ case as seen in Figure 4.19b. This is valid for both cases up to the plate angle of $\theta=120^\circ$, and for larger plate angles it can be concluded that a reduction in both the wake length and width takes place, finally the shape of the streamwise velocity distribution and the vector field shows a similar flow structure with the natural cylinder case. The length of the negative velocity region to the cylinder base is smaller for $\beta=0.5$ compared to the $\beta=0.4$ case for the range of $100^\circ \leq \theta \leq 150^\circ$. A great difference between the streamwise velocities between the upper and lower shear layer is clearly observed for $95^\circ \leq \theta \leq 150^\circ$ plate angles of both porosity values, however the values slightly increase on the lower shear layer for $\theta=150^\circ$ plate angle.

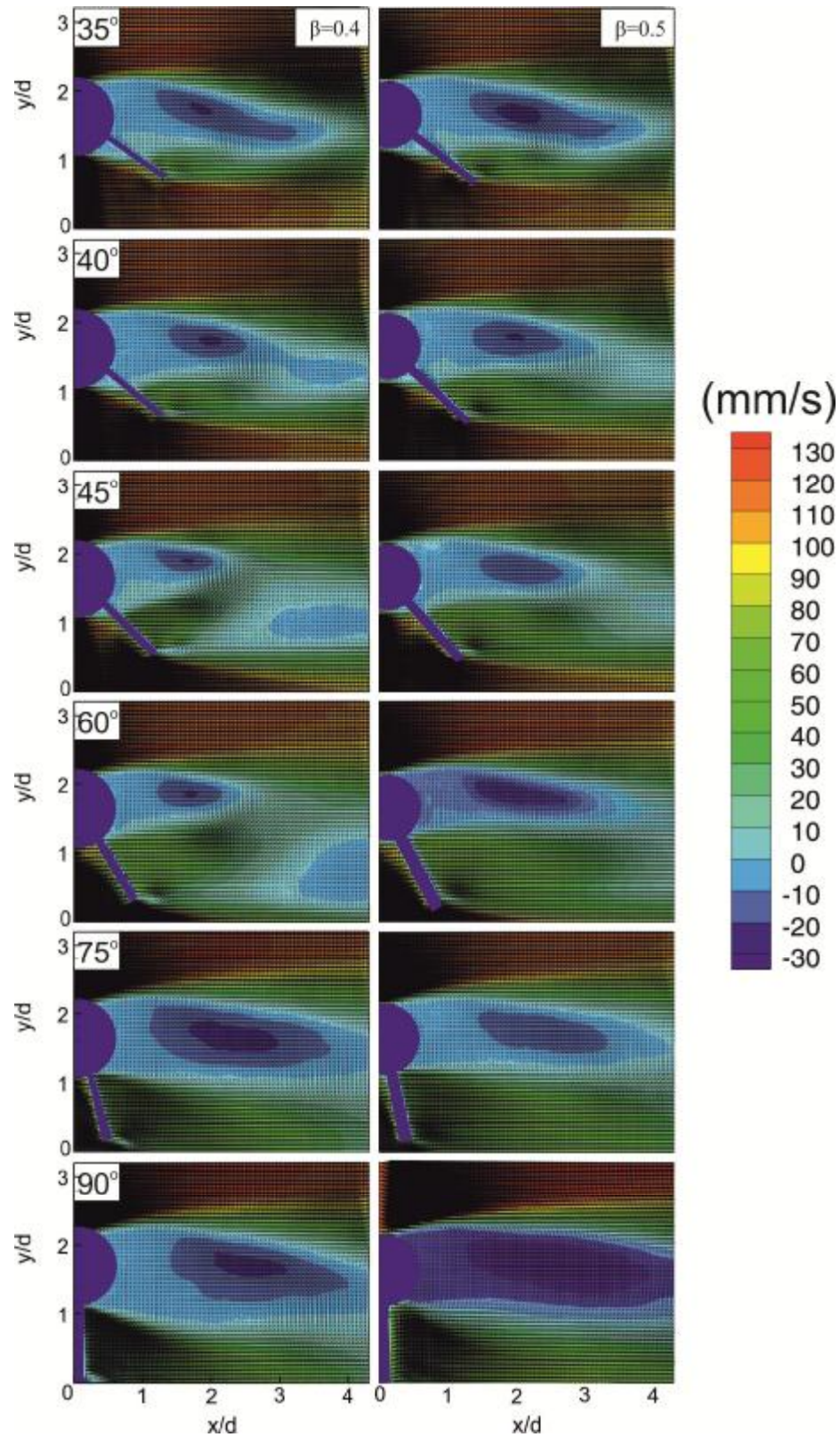


Figure 4. 19a. Time averaged velocity vector field, $\langle \mathbf{V} \rangle$ superposed with the streamwise velocity contours, $\langle u \rangle$ for $35^\circ \leq \theta \leq 90^\circ$ plate angles of $\beta = 0.4$ and 0.5

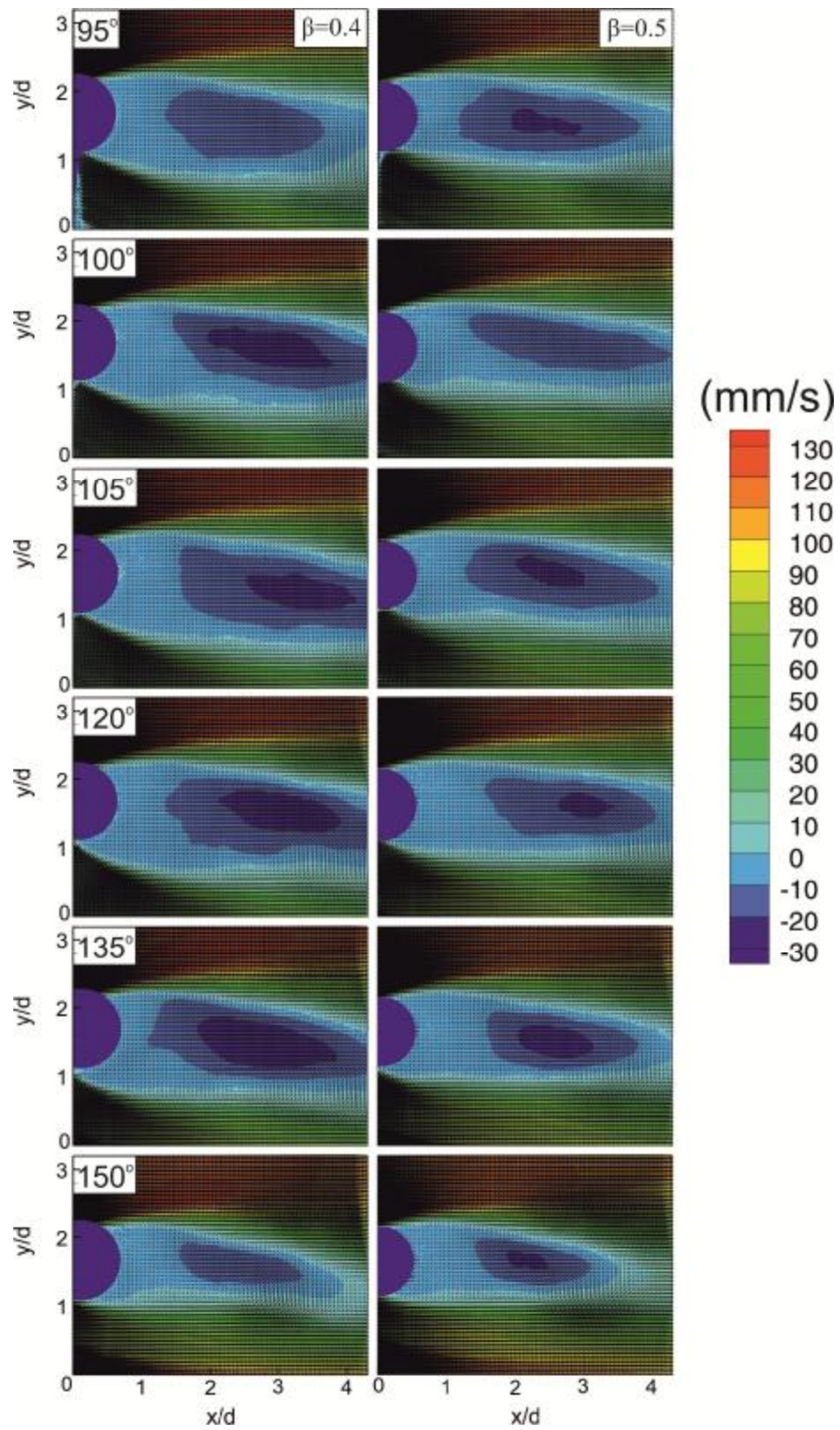


Figure 4. 19b. Time averaged velocity vector field, $\langle V \rangle$ superposed with the streamwise velocity contours, $\langle u \rangle$ for $95^\circ \leq \theta \leq 150^\circ$ plate angles of $\beta=0.4$ and 0.5

The change in velocity vector field and streamwise velocity distribution by further increase in porosity, i.e. $\beta=0.6$ and 0.7 is shown in Figures 4.20a and 4.20b for all plate angles. It is expected that increasing the porosity would result in higher momentum injection through the meshes, hence into the near wake region. Therefore, the velocity deficit in the wake region would decrease by increasing porosity of screen since more fluid is injected into the cylinder wake. This situation is clearly observed when looking at both figures.

The velocity deficit in the wake slightly increases up to the plate angle of $\theta=100^\circ$ for $\beta=0.6$ which corresponds to a larger cylinder wake. For $\theta \geq 105^\circ$, the length of the wake gradually decreases whereas the velocity vectors get higher magnitudes around the reduced velocity region.

Increasing the porosity value to the level of $\beta=0.7$ causes a sudden change in both wake length and width when compared to the first and second row of Figures 4.20a and 4.20b. The wake length dramatically decreases for $\theta=40^\circ$ compared to the plate angle of $\theta=35^\circ$, but it gradually increases by increasing plate angle to a value of $\theta=95^\circ$. For $\theta \geq 95^\circ$ the length of the wake is dramatically reduced compared to the lower porosity values.

It can be concluded from the velocity vectors distributions which are superposed with streamwise velocity contours that the porosity range of $0.4 \leq \beta \leq 0.6$ is effective on elongating the wake region. Furthermore, the wake region is being shrink which is an indication of controlling the flow as well as reducing the drag force exerted on the body since the drag force is directly related with the pressure differences between the front and rear points of the cylinder where the pressure difference is caused from the velocity difference.

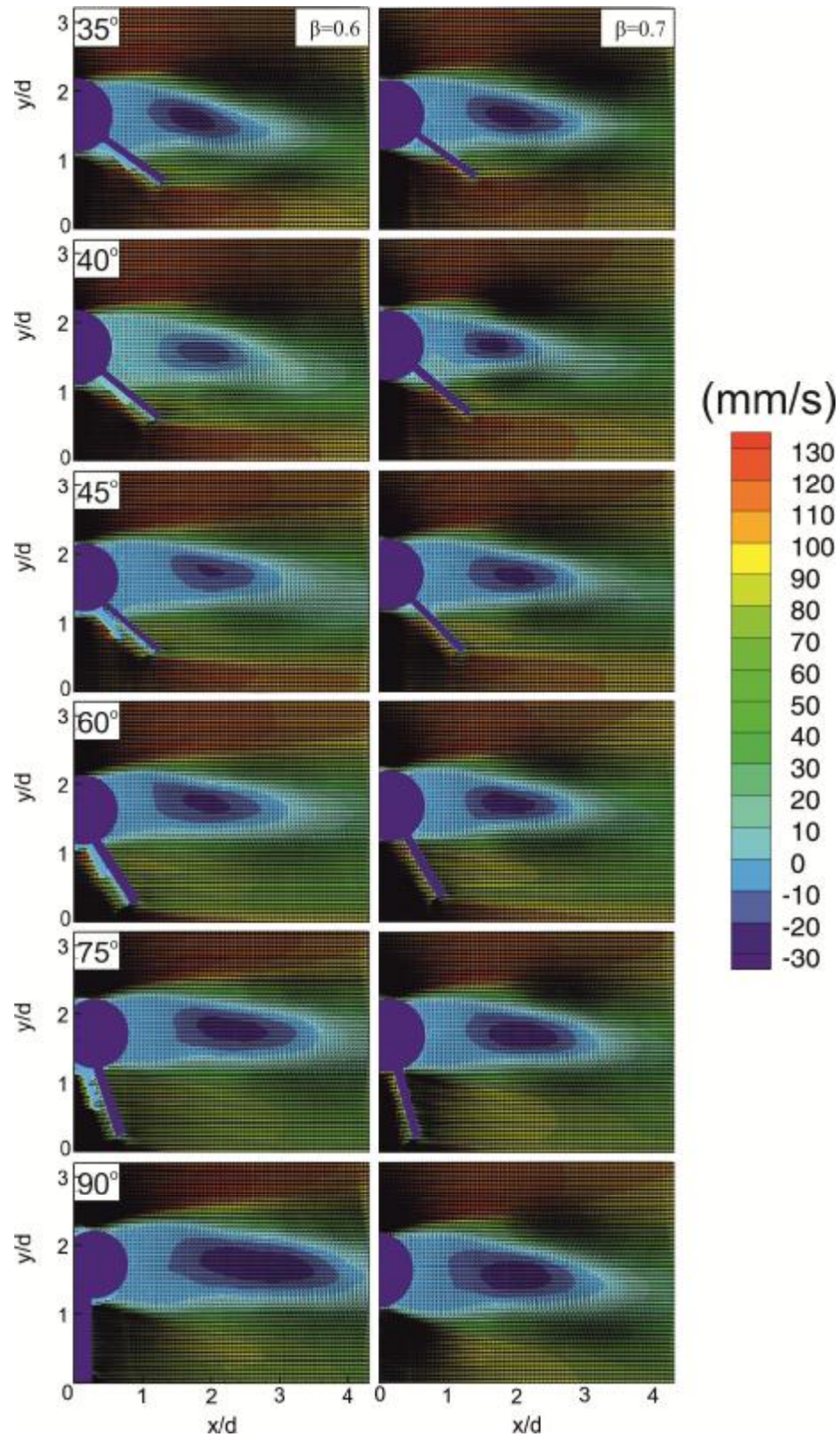


Figure 4. 20a. Time averaged velocity vector field, $\langle V \rangle$ superposed with the streamwise velocity contours, $\langle u \rangle$ for $35^\circ \leq \theta \leq 90^\circ$ plate angles of $\beta=0.6$ and 0.7

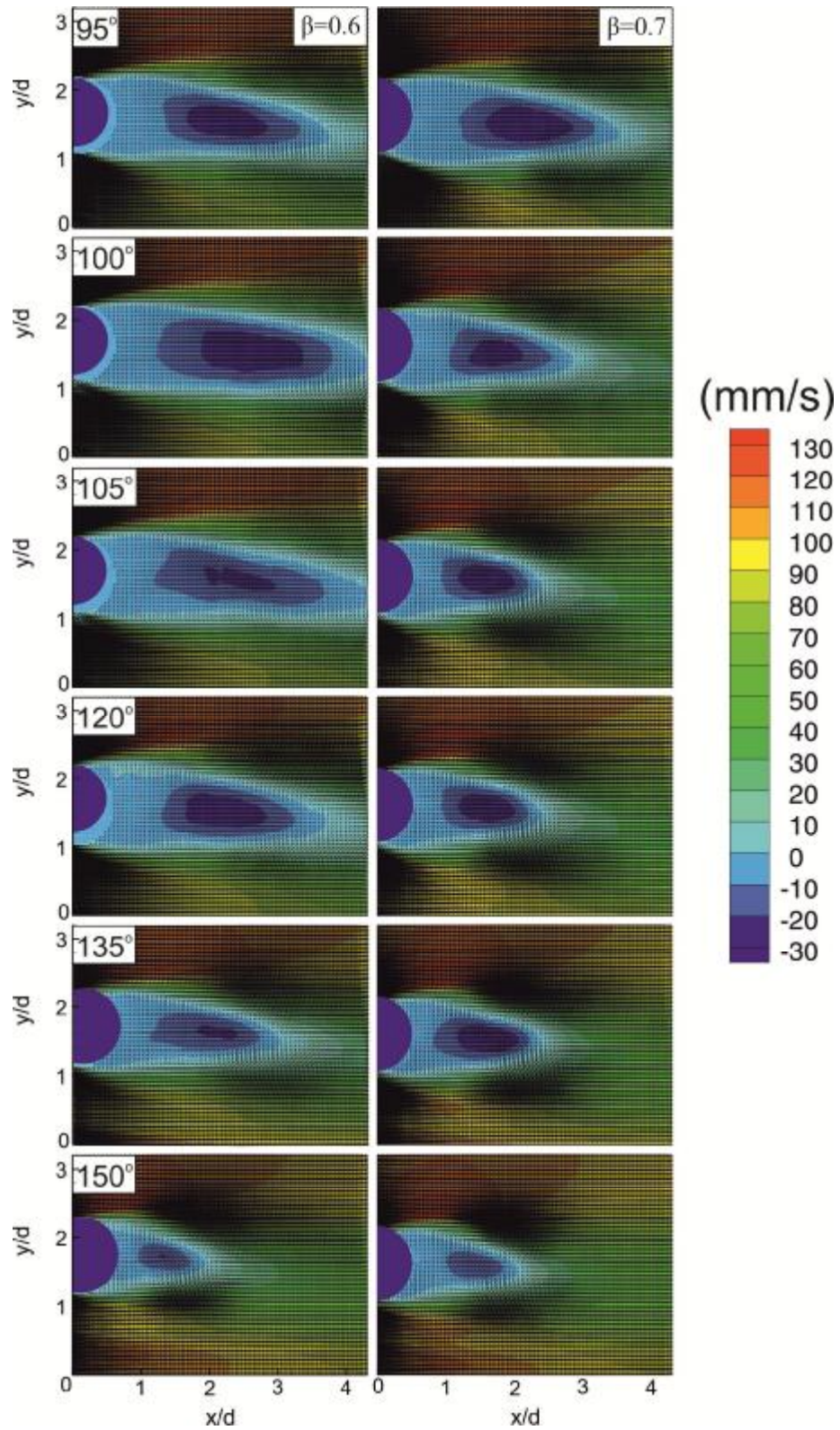


Figure 4. 20b. Time averaged velocity vector field, $\langle V \rangle$ superposed with the streamwise velocity contours, $\langle u \rangle$ for $95^\circ \leq \theta \leq 150^\circ$ plate angles of $\beta = 0.6$ and 0.7

4.4. The Mechanism of Vortex Shedding Suppression

In this section, the vortex shedding suppression mechanism will be analyzed in order to have better understanding of how the permeable plate controls the unsteady flow downstream of the cylinder. The physics of the mechanism will be explained using the instantaneous dye visualization pictures and the vorticity contours of PIV data. For all cases the large-scale colored solid and dashed lines present the positive and negative vorticity layers, respectively. For the permeable plate cases, the data are given starting with the ineffective angles, up to the effective plate angles on flow control, i.e. for $\theta=0^\circ$, 15° , 30° , 45° and 60° .

Figure 4.21 shows the results of natural cylinder case consisting the dye picture and instantaneous vorticity contours. In dye visualization picture, the roll-up of the upper and lower shear layers are obviously seen which results in alternating vortex shedding and consequent Karman Vortex Street downstream of the cylinder. It can be said that the instantaneous vorticity contours evaluated from PIV data are in good agreement with dye picture where the interaction of upper and lower shear layers and consequent vortex shedding are clearly seen.

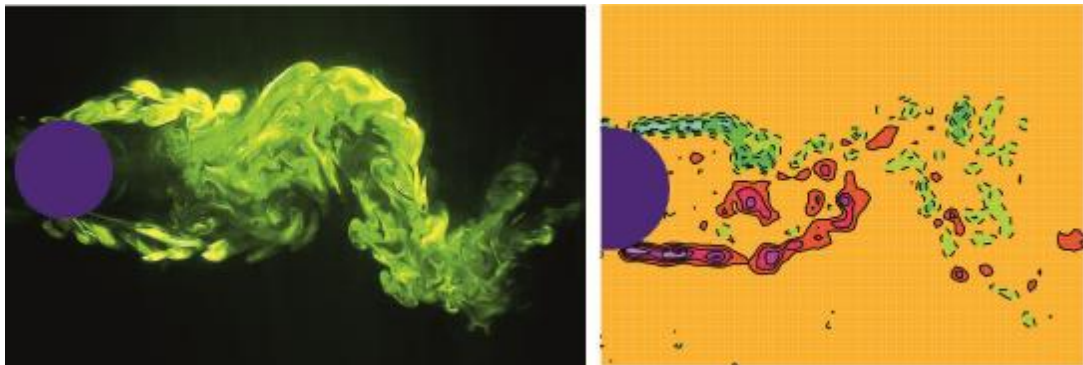


Figure 4.21. Instantaneous dye visualization picture and vorticity contours for natural cylinder

As mentioned in previous sections, attaching a permeable plate along the midspan of a cylinder and giving a certain angle to the plate would directly affect on the shear layer it deals with (in our study the lower shear layer). Figure 4.22 presents the instantaneous data for the permeable plate of $\beta=0.4$ for specified plate angles. For

the case of $\theta=0^\circ$, the upper and lower shear layers interact with each other at trailing edge of the permeable plate. This means that the vortices formed by the shear layers cannot roll-up through the meshes because of the low porosity value, therefore well-organized Karman Vortex Street is formed in farther downstream regions. Increasing the plate angle to a value of $\theta=15^\circ$ results in an increase in the distance of first contact point of upper and lower shear layers, however the formation of vortex shedding and Karman Vortex Street still obviously seen on both dye picture and vorticity contours. Up to $\theta=15^\circ$, the length of the plate was not adequate to interfere with the generated shear layer emanating from the lower shoulder of the cylinder, but increasing the plate angle to a value of $\theta=30^\circ$ would be enough to have a direct interaction as seen by the third row of Figure 4.22. At $\theta=30^\circ$, the upper shear layer rolls-up into the wake region with the absence of lower shear layer. From this plate angle to the higher ones, the blockage of permeable plate on the formation of lower shear layer takes place. Besides, the fluid flows through the meshes resulting in a momentum transfer into the wake region changing by the plate angle. On the other hand, the amount of momentum injection through the meshes is very low since the porosity value is $\beta=0.4$ which is the lowest porosity used in the present work. Increasing the plate angle to $\theta=45^\circ$ and 60° results in a complete suppression of vortex shedding since the formation of a lower shear layer is significantly blocked by the permeable plate which results in the absence of a vortex interaction between upper and lower shear layers even there exists a low-magnitude shear layer (compared with the upper shear layer) generated by the fluid flowing through wire meshes. This low-magnitude lower shear layer is insufficient to interact with the upper shear layer; hence the upper shear layer takes the dominant role on the flow structure. Finally, the vortex shedding is eliminated and consequent formation of Karman Vortex Street is prevented for $\theta=45^\circ$ and 60° . Furthermore, instead of large scale Karman Vortex Street, small-scale Kelvin- Helmholtz vortices are generated especially on the upper shear layer as seen by the dye visualization pictures.

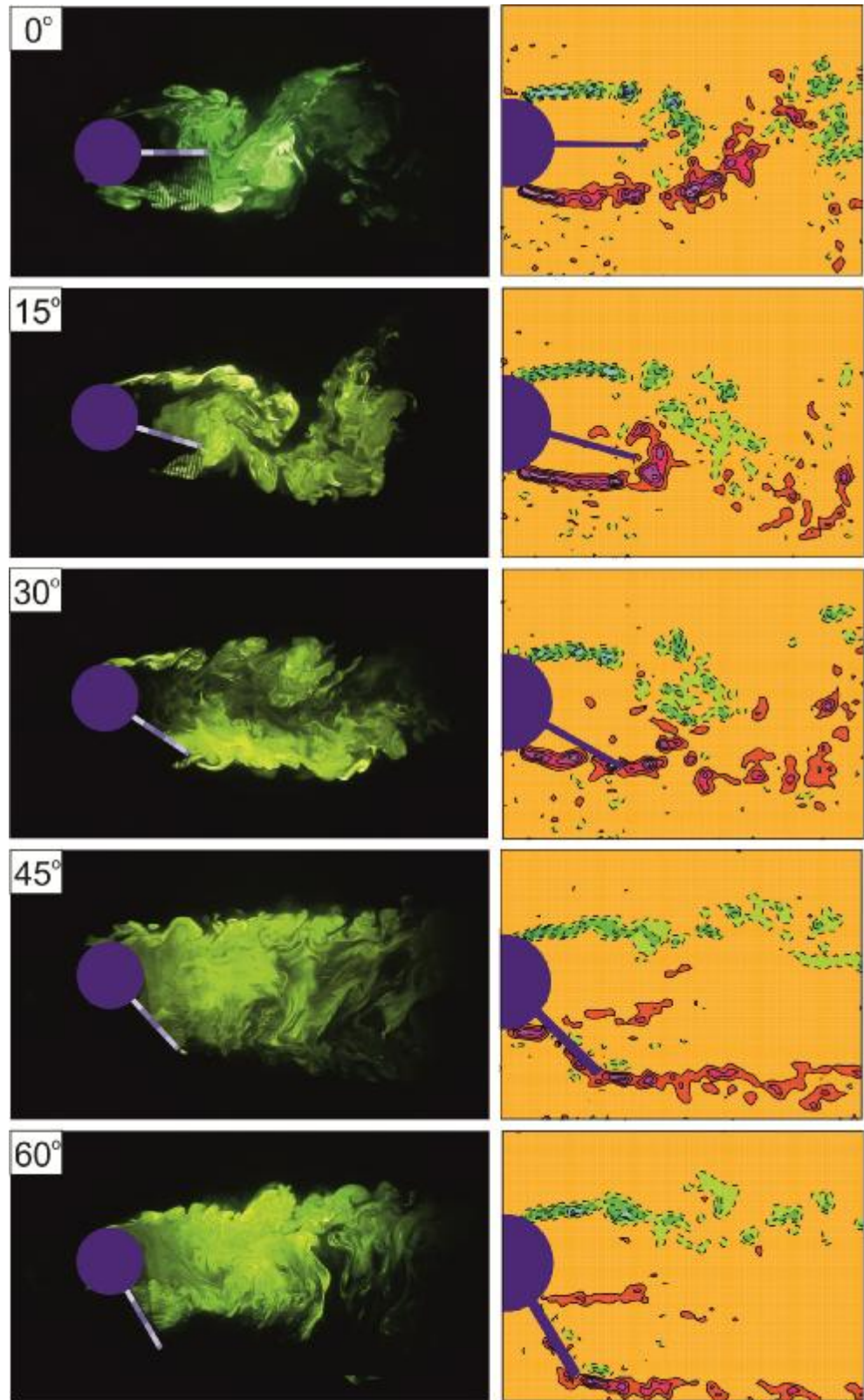


Figure 4.22. Instantaneous dye visualization pictures and vorticity contours for the permeable plate cases of $\beta=0.4$

As evaluated by the time-averaged results on previous sections, it is expected that the porosity of $\beta=0.5$ would affect the flow structure similar to the case of $\beta=0.4$. Figure 4.23 shows the instantaneous flow results of $\beta=0.5$ by means of dye pictures and vorticity contours. Similar with the $\beta=0.4$ case, the vortex formation and interaction of the vortices are evaluated and vortex formation length is increased for the plate angles of $0^\circ \leq \theta \leq 30^\circ$. For $\theta=45^\circ$ and 60° plate angles, the flow structure dramatically changes as expected, i.e. interaction of the shear layers is prevented, vortex shedding is suppressed and consequent Karman Vortex Street is completely disappeared downstream of the cylinder. However, for both porosity values ($\beta=0.4$ and 0.5) the width of the cylinder wake gradually increases by increasing plate angle.

Further increase in the plate angle is presented in Figure 4.24 which includes the instantaneous dye visualization pictures and vorticity contours for $\beta=0.6$. For the plate angles of $0^\circ \leq \theta \leq 30^\circ$, the trend is similar with the lower porosities with slight changes. The mechanism of vortex suppression starts with the plate angle of $\theta=45^\circ$ and the difference in different porosity values is seen by last two rows of the Figure 4.24. Here, the flow structure observed by dye pictures seems to be similar with the lower porosity cases, however examining the instantaneous vorticity contours brings out the details: Lower shear layer generated by the fluid passing through the meshes increases in its magnitude since the open area of the permeable plate increases by porosity, therefore greater magnitude- lower shear layer is now capable to interact with the upper shear layer. This situation is clearly indicated by the results of $\theta=45^\circ$ where the lower vorticity now has enough momentum to roll-up into the wake resulting in the interaction with the upper shear layer, finally forming the vortex shedding. Furthermore, the formation of Kelvin- Helmholtz vortices with high frequency occurs dominantly on the upper shear layer of the cylinder. However this shedding of the vortices is limited in a small area of the wake. For the plate angle of $\theta=60^\circ$, there is still an interaction of upper and lower shear layers, however it can be said that this plate angle is on one hand more effective on the elimination of vortex shedding, on the other hand increasing the formation length and shifting the Karman Vortex Street.

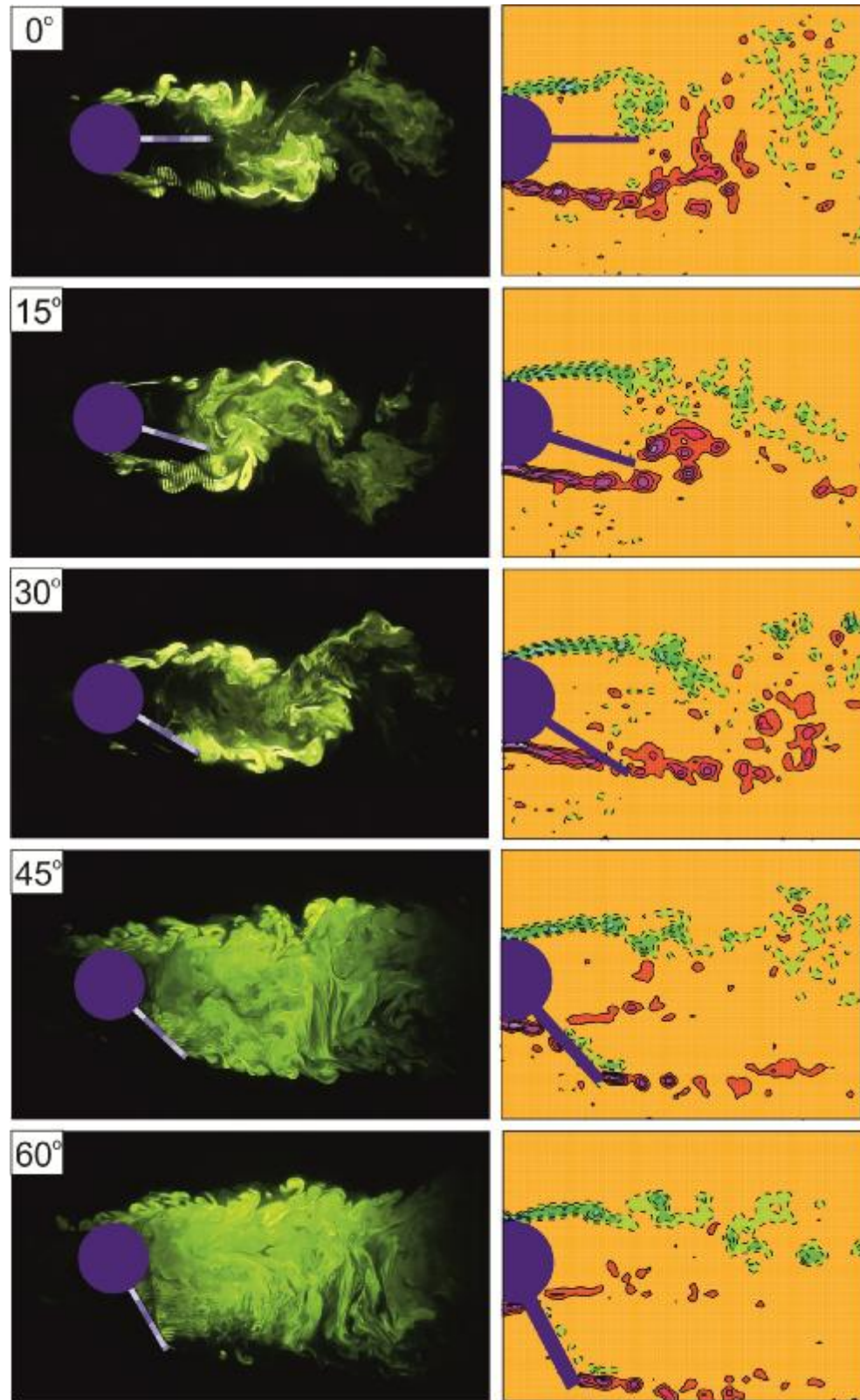


Figure 4.23. Instantaneous dye visualization pictures and vorticity contours for the permeable plate cases of $\beta=0.5$

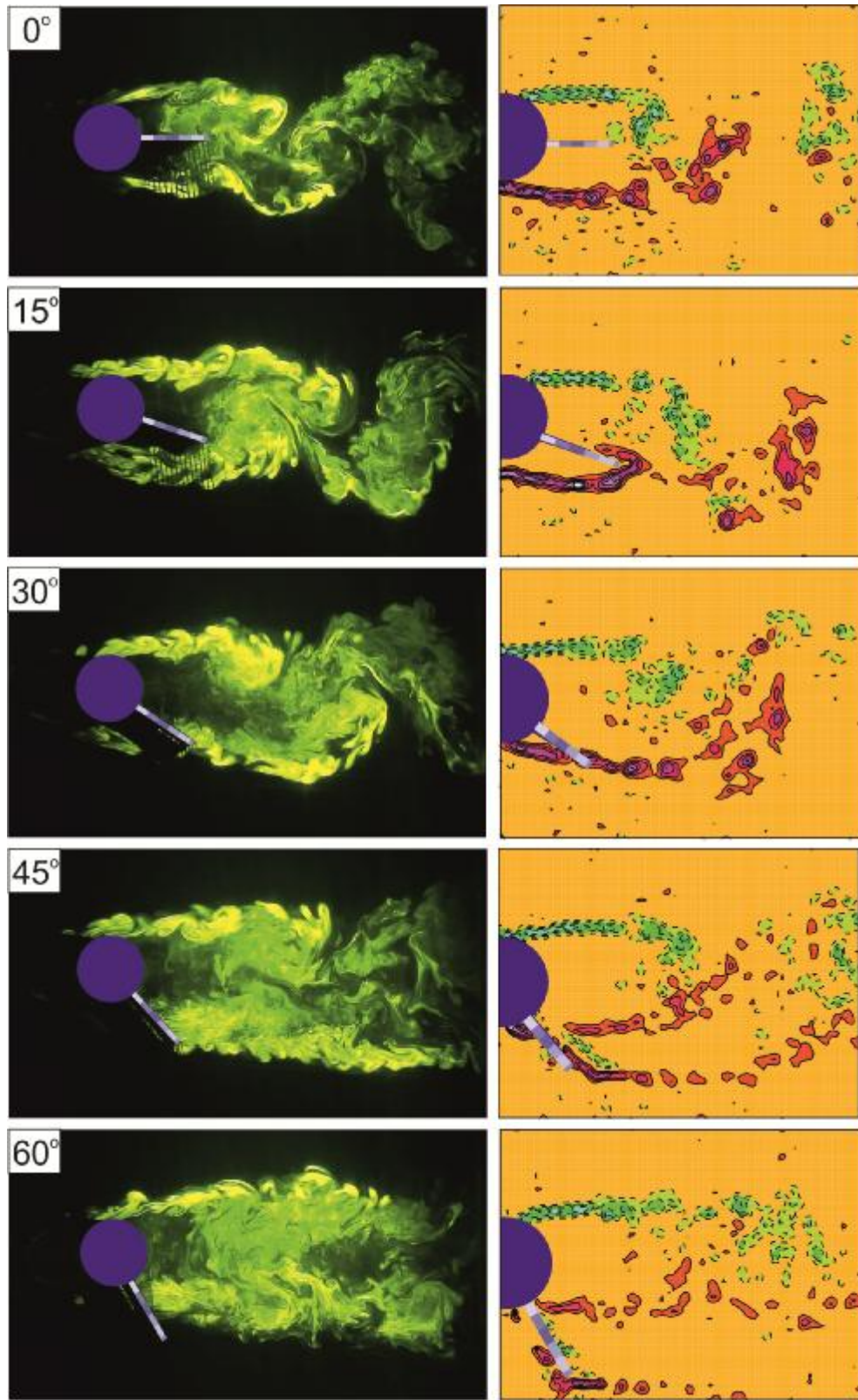


Figure 4.24. Instantaneous dye visualization pictures and vorticity contours for the permeable plate cases of $\beta=0.6$

Figure 4.25 includes the instantaneous dye visualization pictures and vorticity contours for $\beta=0.7$. It is obvious that the plate angles of $0^\circ \leq \theta \leq 30^\circ$ are effective on increasing the vortex formation region and attenuating the vortex shedding, however there exists large scale Karman vortices, furthermore the shear layers interact with each other for these plate angles. Instantaneous vorticity contours for the plate angle of $\theta=45^\circ$ point out that the magnitude of lower shear layer generated by the fluid passing through the permeable plate significantly increases and rolls-up into the wake region. This is due to the increase in porosity value resulting by the increased open area on larger mesh sizes such as $\beta=0.7$. As a result, different than with the lower porosity values, the vortex shedding takes place in the wake region with increase in length and width for $\theta=45^\circ$ and 60° plate angles. However it should also be pointed out that the vortex street formed by the cases of $\beta=0.7$ for plate angles of $\theta=45^\circ$ and 60° covers a smaller region in its width compared to the smaller plate angles, as well as the natural cylinder.

Finally, it is concluded from observation of dye visualization and PIV results that the suppression mechanism successfully works for the porosities of $0.4 \leq \beta \leq 0.6$ which are effective on suppressing the vortex shedding (for effective ranges mentioned in previous sections) and consequent Karman Vortex Street by directly preventing the formation of lower shear layer on the bottom shoulder of the cylinder. However, formation of the high-frequency Kelvin- Helmholtz vortices is obvious along the upper shear layer of the cylinder for the controlled cases ($\theta \geq 45^\circ$). Furthermore the porosity value of $\beta=0.6$ is defined as the transition porosity since the passing fluid through the meshes starts to have enough momentum to generate the lower shear layer but still not enough to roll-up into the wake and interact with the upper shear layer.

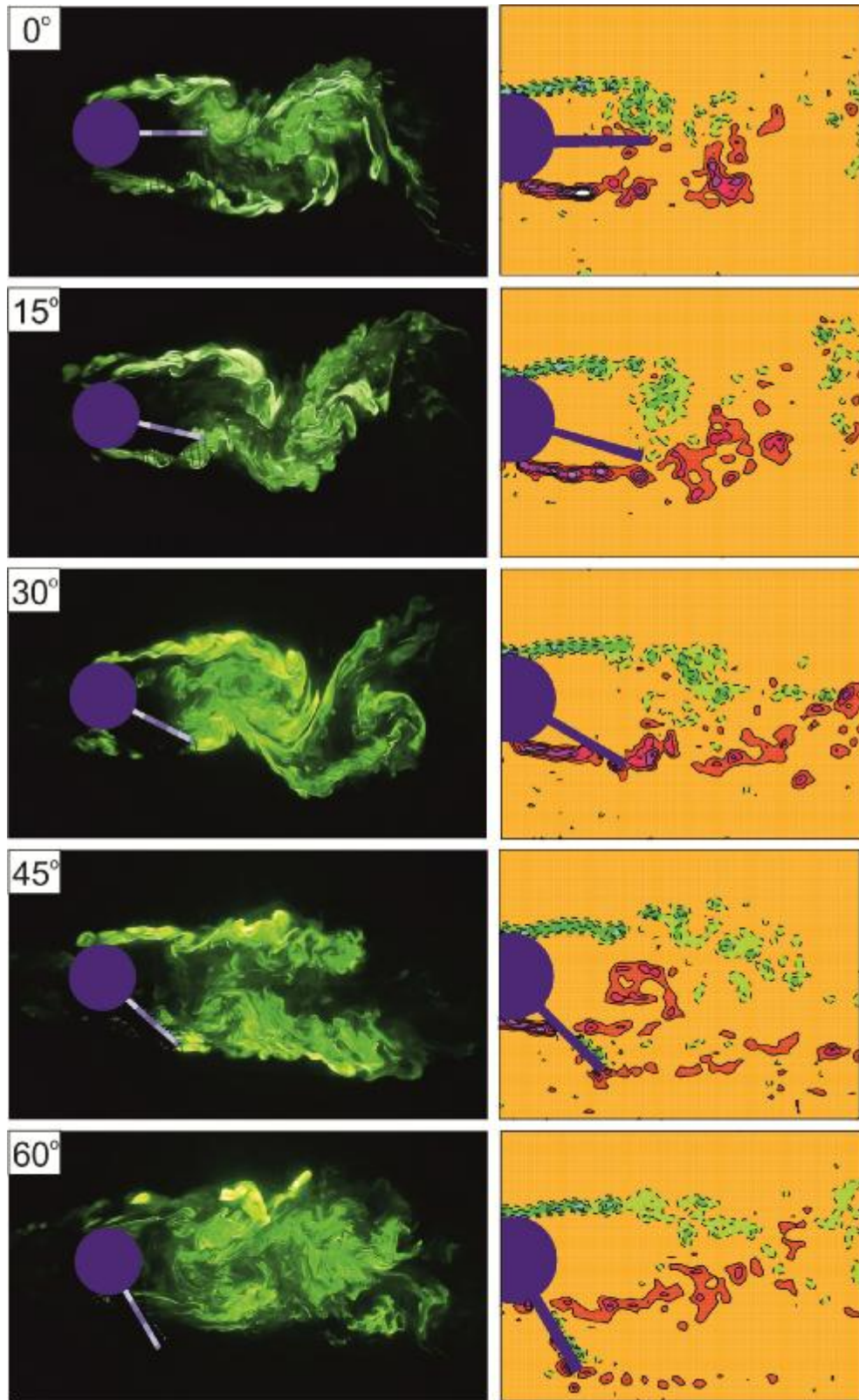


Figure 4.25. Instantaneous dye visualization pictures and vorticity contours for the permeable plate cases of $\beta=0.7$

4.5. FFT Analysis for Analyzing the Vortex Induced Vibrations

In this section, the Fast Fourier Transformation (FFT) technique was used to evaluate the frequency of vortex shedding for various points. Determination of the vortex shedding frequency gives us the frequency of vibration on the body and the amount of vortex induced vibrations (VIV). Therefore it is necessary to analyze and find out the effect of permeable plates on reducing or eliminating the vortex shedding frequency downstream of the cylinder.

4.5.1. FFT Analysis for the Natural Cylinder

The Strouhal number of a circular cylinder as a function of Reynolds number is shown in Figure 4.26 where the Strouhal number is approximately 0.2 for a wide range of Reynolds number, i.e. $250 < Re_d < 2 \times 10^5$.

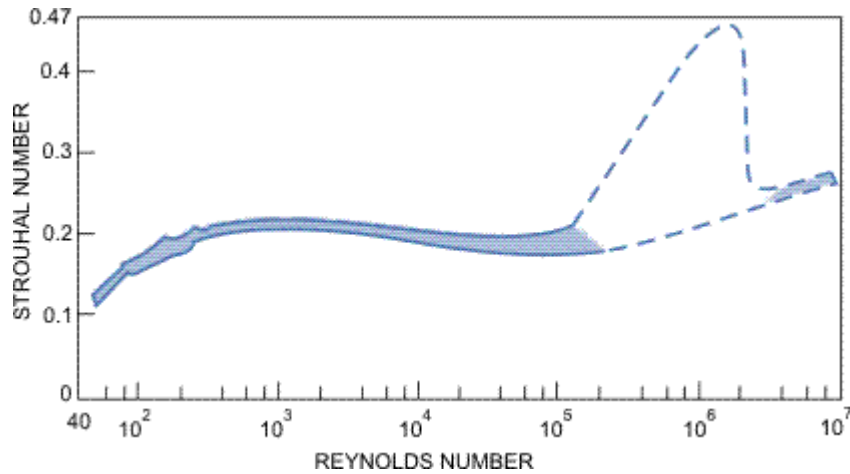


Figure 4.26. The variation of Strouhal number with Reynolds number for a circular cylinder (Blevins, 1990)

Figure 4.27 presents the natural cylinder case having a dominant frequency of $f=0.38$ (at streamwise location of $x/d=1.6$ and spanwise location of $y/d=2$) corresponding to a Strouhal number ($St=f d/U_\infty$ where d is the diameter of cylinder, U_∞ is the free-stream velocity in streamwise direction) of $St=0.2$ which is in good agreement with the results of Blevins (1990) as seen in Figure 4.26. A clear and

distinct peak of frequency shown in Figure 4.27 is an indication of the existence of a dominant vortex structure owing to vortex shedding and consequent vibrations of the vortices. Therefore it is essential to reduce or eliminate the dominant vortex shedding frequency occurred downstream of the cylinder using the flow control techniques.

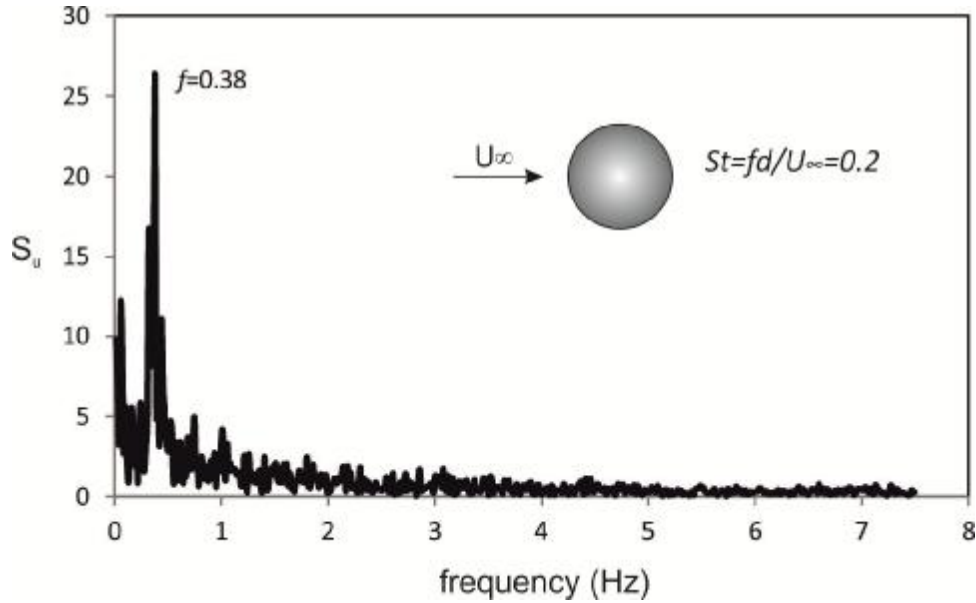


Figure 4.27. Spectra of streamwise velocity fluctuations for the natural cylinder

4.5.2. FFT Analysis for $\theta=0^\circ$ cases

Use of a permeable plate with $\theta=0^\circ$ plate angles (splitter plate) are presented within this section. For all porosity values, the occurrence of a dominant frequency was searched by FFT analysis on the upper shear layer (at $y/d=2$) of the cylinder for three different points of $x/d=2.0$, 2.4 and 2.8 in streamwise directions. These locations are designated by 1, 2 and 3 on figures.

Figure 4.28 shows the results for the solid plate ($\beta=0.0$) where a dominant frequency having a value of $f=0.23$ is evaluated on the upper shear layer and the magnitude of S_u slightly increases while fluid moves from point 1 to 3. Taking into account the frequency result of natural cylinder as being $f=0.38$, a reduction of almost 40% can be achieved using a solid plate with $\theta=0^\circ$ plate angle (splitter plate).

Using a permeable plate instead of a solid one brings out some slight changes on the flow structures as indicated by the results presented in previous sections. Variations in flow structures are due to the ability of fluid to pass through the meshes when compared with the solid plate. Therefore, it is realized that the vortex shedding frequency is directly affected by the permeable plates.

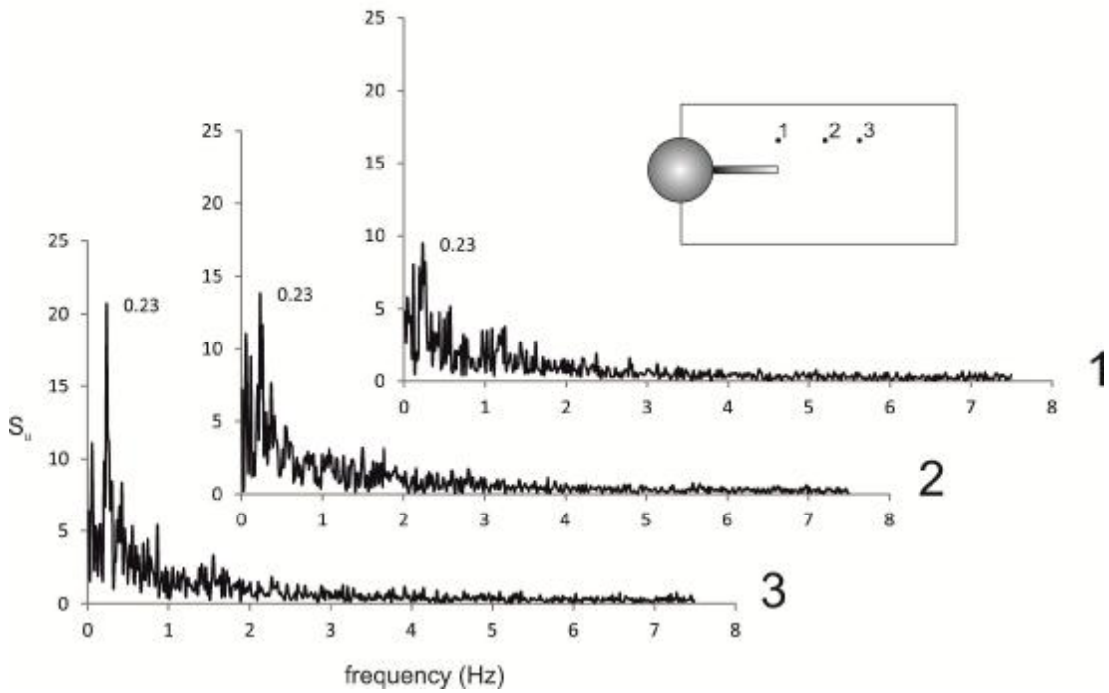


Figure 4.28. Spectra of streamwise velocity fluctuations for the solid plate ($\beta=0.0$) at the plate angle of $\theta=0^\circ$

Figure 4.29 presents the power spectra of streamwise velocity fluctuations for the porosity value of $\beta=0.4$. At farther downstream regions of the cylinder on upper shear layer, namely points 1, 2 and 3, it is clearly seen that a dominant frequency having a value of $f=0.34$ exists and gradually develops in its amplitude. The value of the frequency is lower compared to the natural cylinder case. A reduction of 11% is takes place by using a permeable plate having $\beta=0.4$ porosity while shifting the location of dominant frequency to further downstream regions.

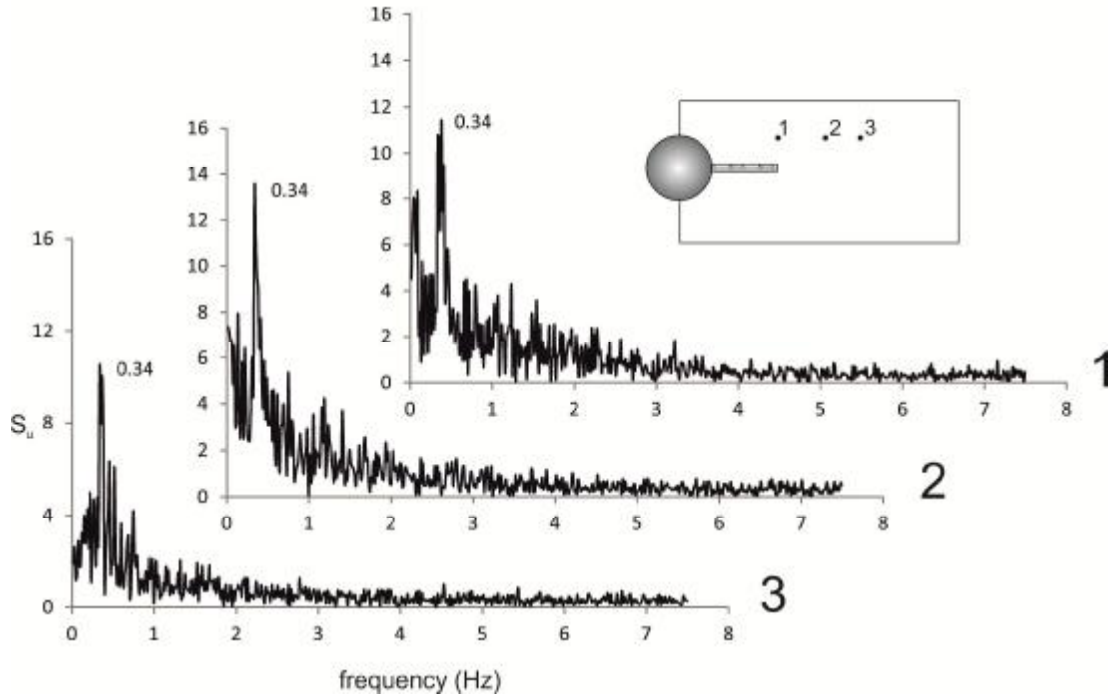


Figure 4.29. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.4$ at the plate angle of $\theta=0^\circ$

The results for $\beta=0.5$ case are presented in Figure 4.30. It is seen that a frequency of $f=0.35$ takes place at point 1 as being an indication of reduction of vibrations in near wake region, however this dominant frequency increases by increasing the streamwise locations. The dominant frequency takes a value of $f=0.38$ at locations 2 and 3 where this frequency value is the same with the natural cylinder case. Although the same frequency evaluated, the amplitude is lower than that of the natural cylinder.

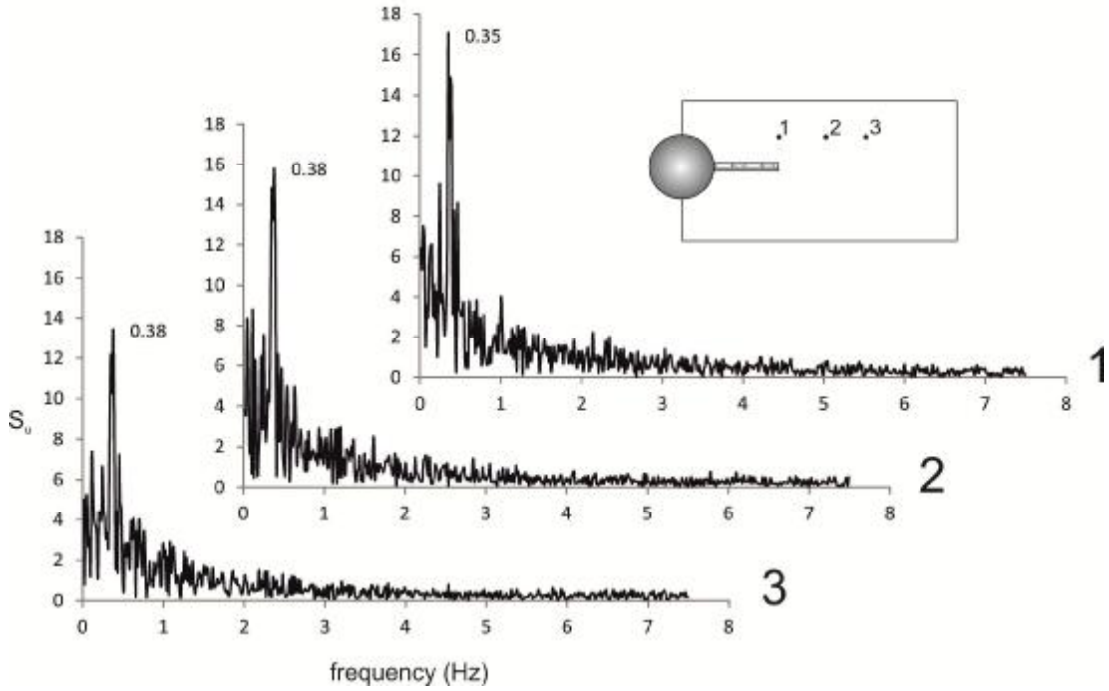


Figure 4.30. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.5$ at the plate angle of $\theta=0^\circ$

Increasing the porosity to $\beta=0.6$ results in occurrence of a dominant frequency of $f=0.38$ even for location 1 as can be seen in Figure 4.31. This frequency value (as being same with the natural cylinder frequency) is valid also for points 2 and 3 where its amplitude increases from point 1 to point 2 whereas decreases from point 2 to point 3, so it can be concluded that the repeatability of this frequency is higher at point 2. Nevertheless, it should be pointed out that the amplitudes of the S_u are lower than that of for the natural cylinder.

The power spectra of streamwise velocity fluctuations for the porosity of $\beta=0.7$ is shown in Figure 4.32. Similar with the case of $\beta=0.6$, a dominant frequency is obtained in all points, however the value of the frequency is found to be lower: $f=0.37$ even the amplitudes are higher compared to $\beta=0.6$. When compared with the natural cylinder, the frequency is reduced by 2.6% whereas the amplitudes of are closer to each other.

Finally it can be concluded that use of a permeable plate of $\beta=0.4$ porosity having $\theta=0^\circ$ is effective on reducing the velocity fluctuations (almost 11%) in the

near wake of a cylinder by shifting the location of dominant frequency in downstream directions.

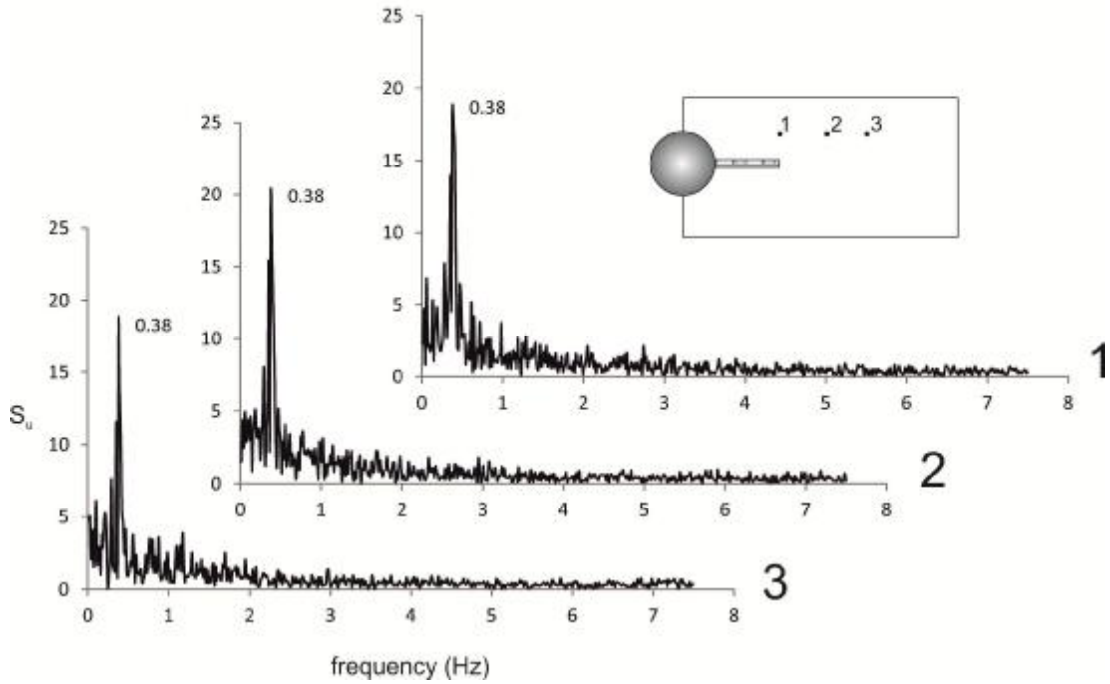


Figure 4.31. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.6$ at the plate angle of $\theta=0^\circ$

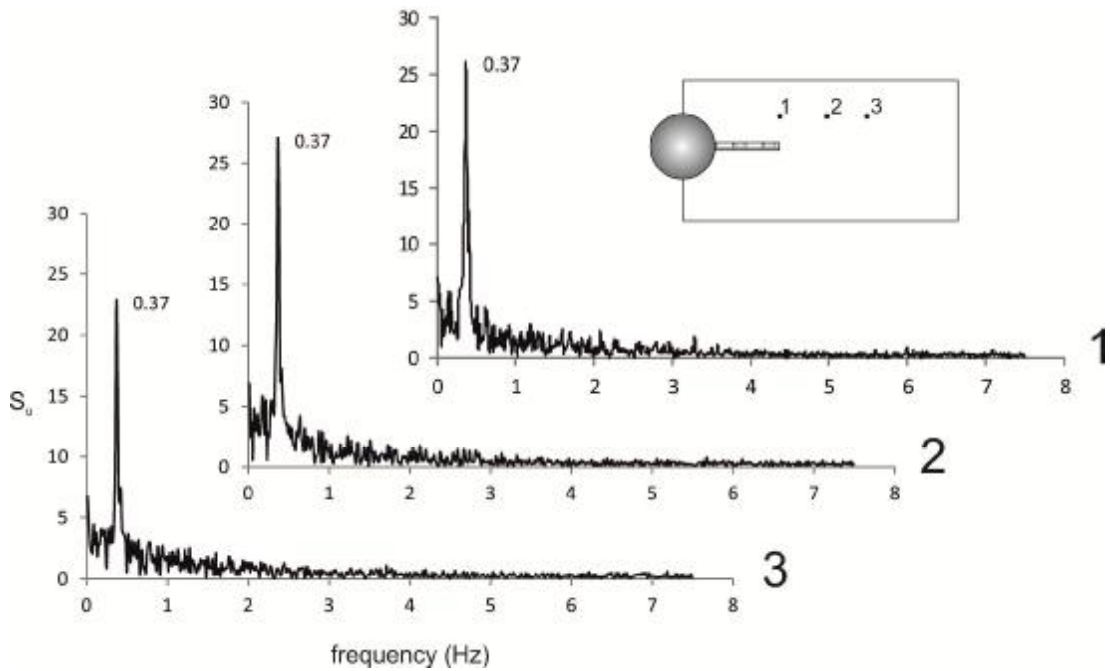


Figure 4.32. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.7$ at the plate angle of $\theta=0^\circ$

4.5.3. FFT Analysis for $\theta=15^\circ$ Cases

The effects of $\theta=15^\circ$ cases on vortex shedding frequency are presented in this section for locations (shown in all figures of this section) 1, 2, 3 and 4 on the upper shear layer and 5, 6, 7 on the lower shear layer which starts to interact with the trailing edge of permeable plate. The locations 1, 2, 3, 4, 5, 6 and 7 correspond to the coordinates of $(x/d, y/d) = (1.6, 2.0), (2, 2.0), (2.4, 2.0), (2.8, 2.0), (1.8, 1.0), (2.4, 1.0)$ and $(2.8, 1.0)$, respectively.

The power spectra of streamwise velocity fluctuations for the solid plate are presented in Figure 4.33. As clearly shown by the graphs there is no dominant frequency evaluated for all points. Absence of a dominant frequency means that there is no periodic motion of the flow at that point, hence the vortex shedding downstream of the cylinder does not exist anymore. However, the upper shear layer shows unstable velocity fluctuations without a certain peak which might be due to small scale eddies formed from the upper shear layer. Therefore the solid plate having $\theta=15^\circ$ is favorable on eliminating the vortex shedding downstream of the cylinder.

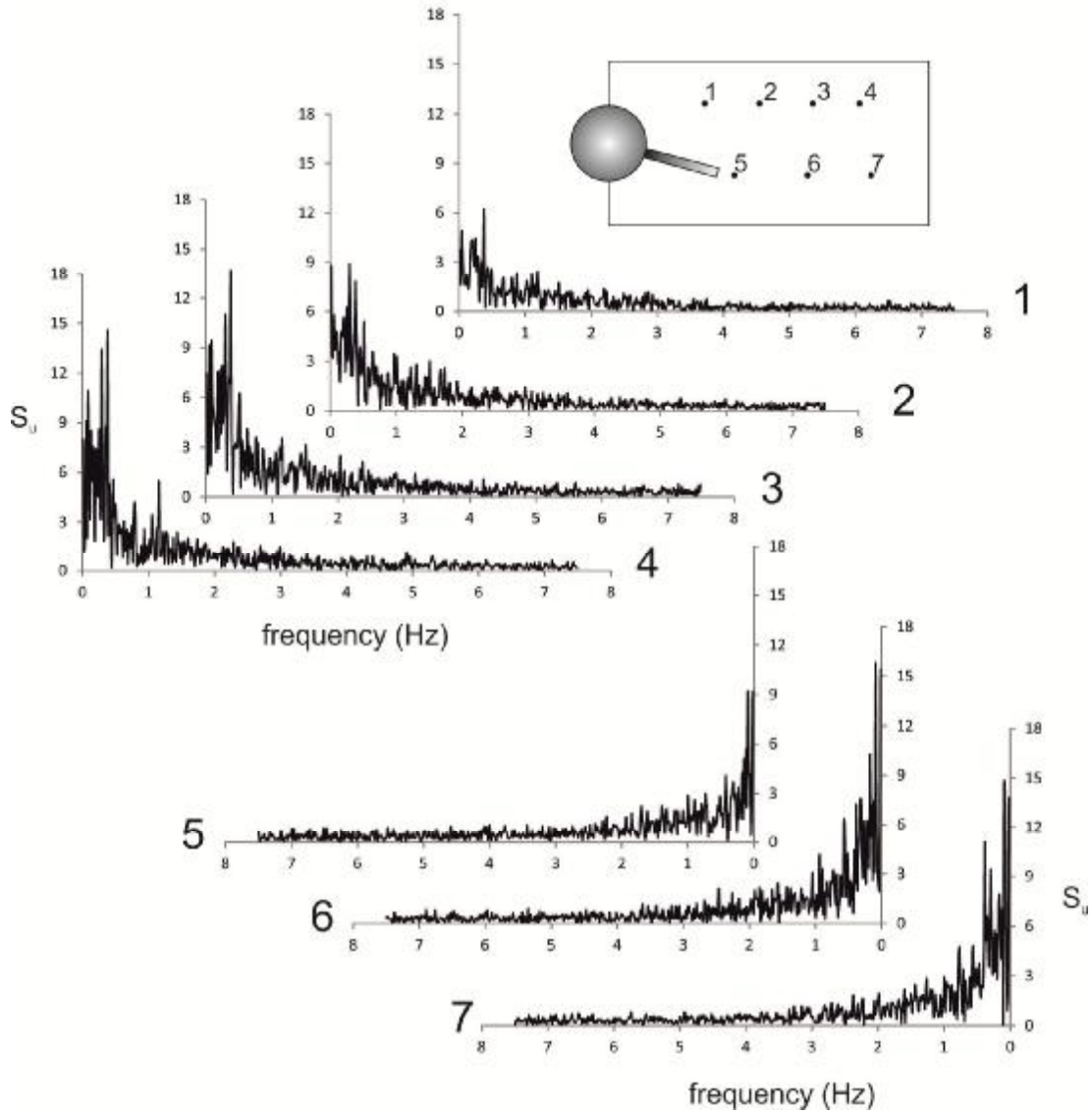


Figure 4.33. Spectra of streamwise velocity fluctuations for the solid plate ($\beta=0.0$) at the plate angle of $\theta=15^\circ$

The results for the porosity of $\beta=0.4$ are shown in Figure 4.34 where occurrence of a dominant peak is clearly observed for each point with various amplitudes. For points 1, 5, 6 and 7, a frequency of $f=0.38$ is evaluated which is same with the frequency of natural cylinder. On farther regions away from point 1, there exist a lower frequency having a value of $f=0.35$ which is lower than the natural cylinder, as well as the frequency evaluated for the lower shear layer. As a conclusion, it is possible to have slight reduction in the frequency upper shear layer using a permeable plate of $\beta=0.4$ for $\theta=15^\circ$ plate angle.

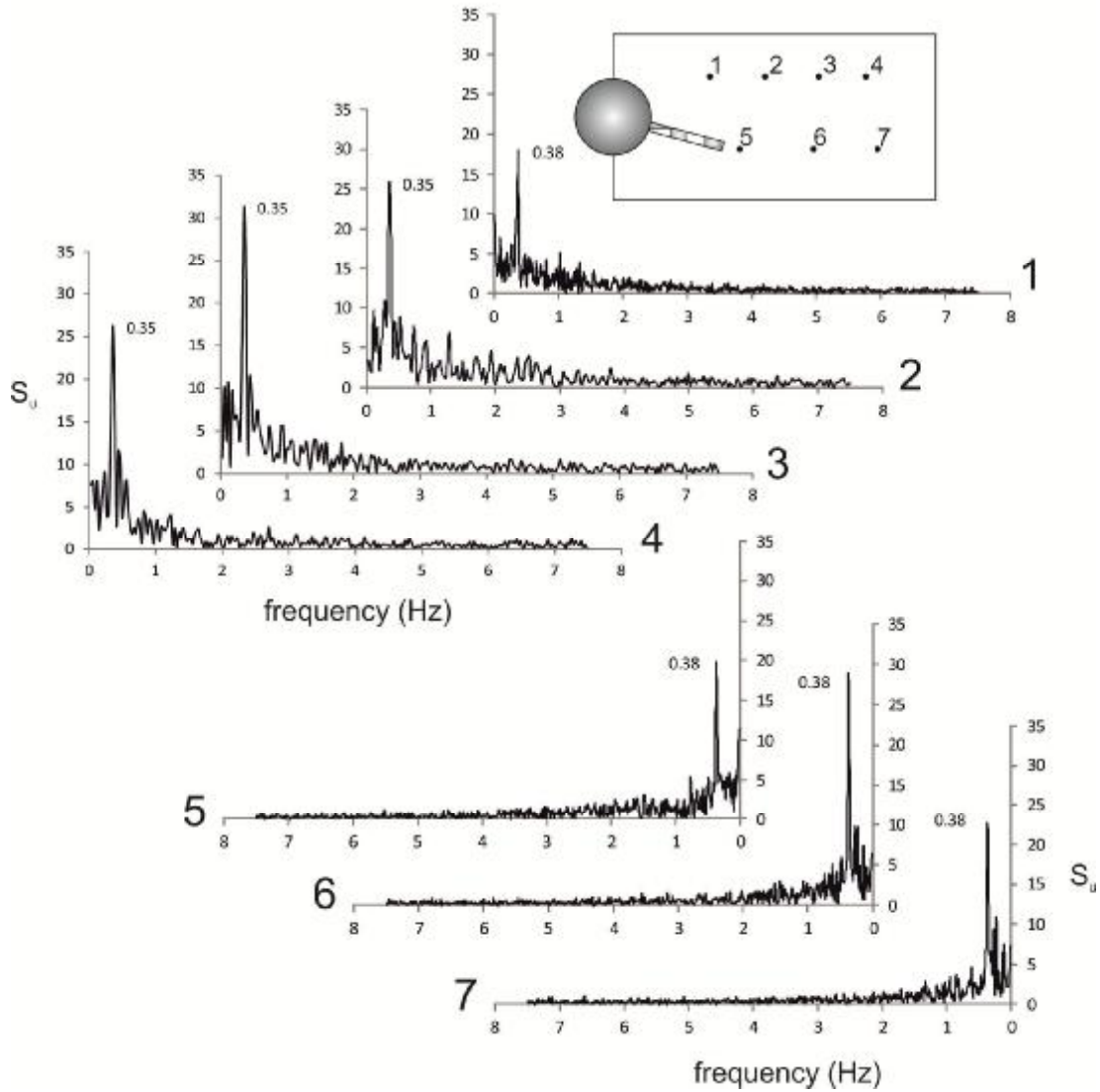


Figure 4.34. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.4$ at the plate angle of $\theta=15^\circ$

Increasing the porosity value to $\beta=0.5$ results in lower dominant peaks of S_u on both upper and lower shear layers as shown in Figure 4.35. For all points, a dominant frequency of $f=0.35$ is evaluated which means an unsteady vortex shedding still exists downstream of the cylinder, however the level of frequency is 8% lower than the natural cylinder frequency. The maximum amplitudes are evaluated at points 3 and 6 for the upper and lower shear layers, respectively.

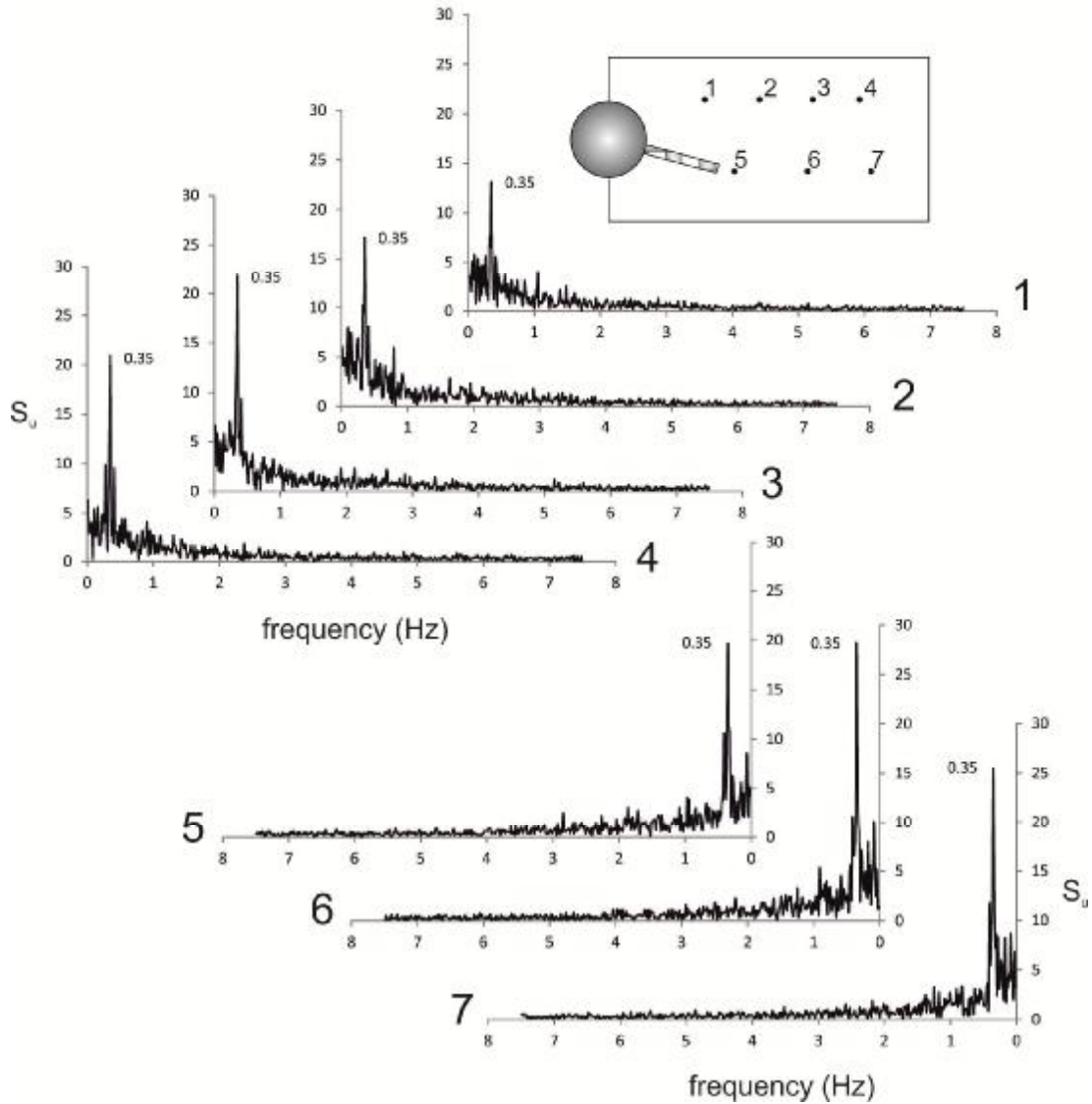


Figure 4.35. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.5$ at the plate angle of $\theta=15^\circ$

It is expected that further increase in porosity would results in evaluation of dominant peaks where Figure 4.36 and 4.37 present the fluctuations of streamwise velocity components for $\beta=0.6$ and 0.7 , respectively. Unsurprisingly, increasing the porosity results in larger amplitude peaks of $f=0.35$, as evaluated by the $\beta=0.5$. The amplitudes, which defines the repeatability of periodic motion of the vortex shedding, increase by increasing porosity and reaches its maximum value at $\beta=0.7$ which is shown in Figure 4.37.

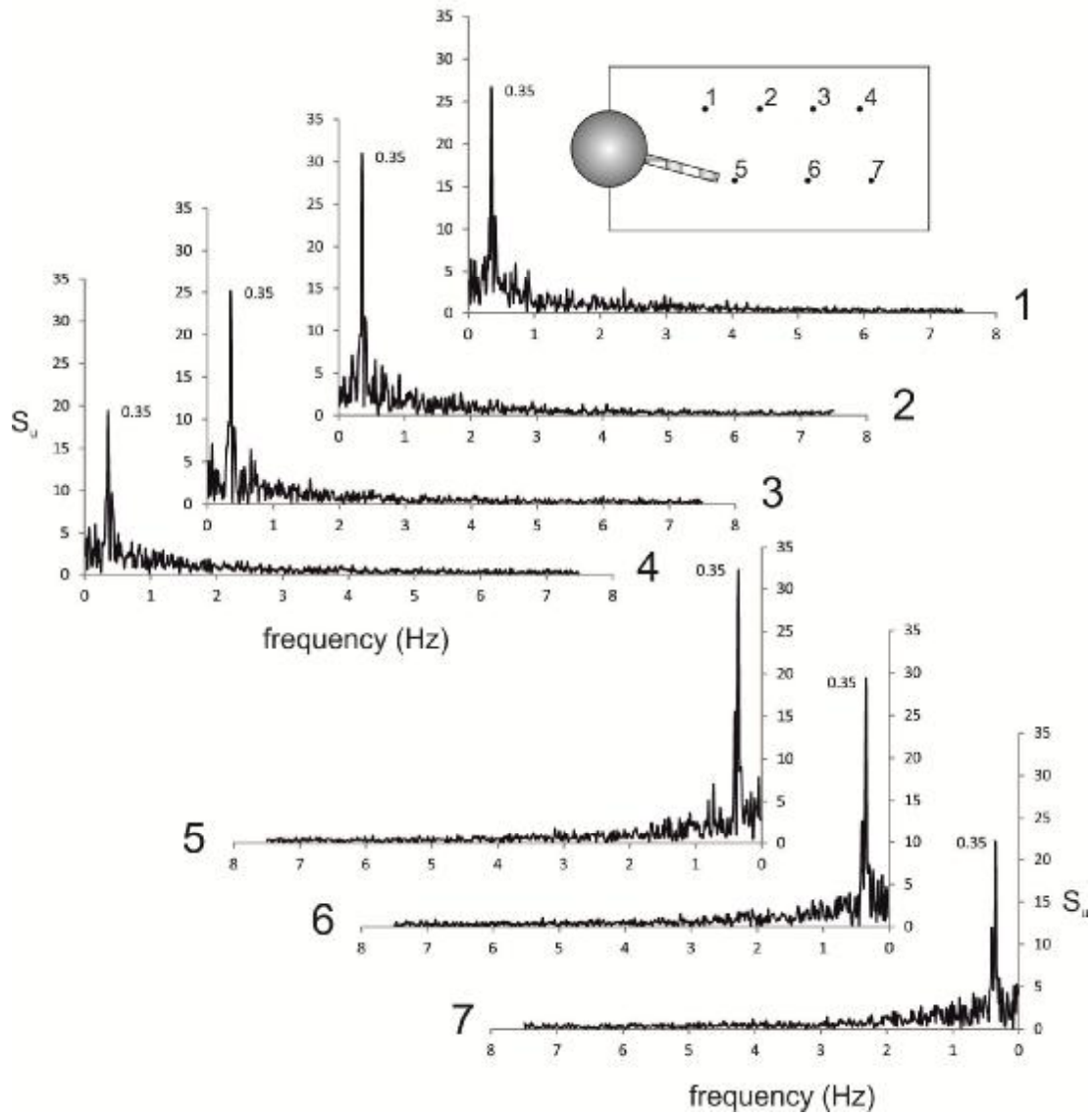


Figure 4.36. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.6$ at the plate angle of $\theta=15^\circ$

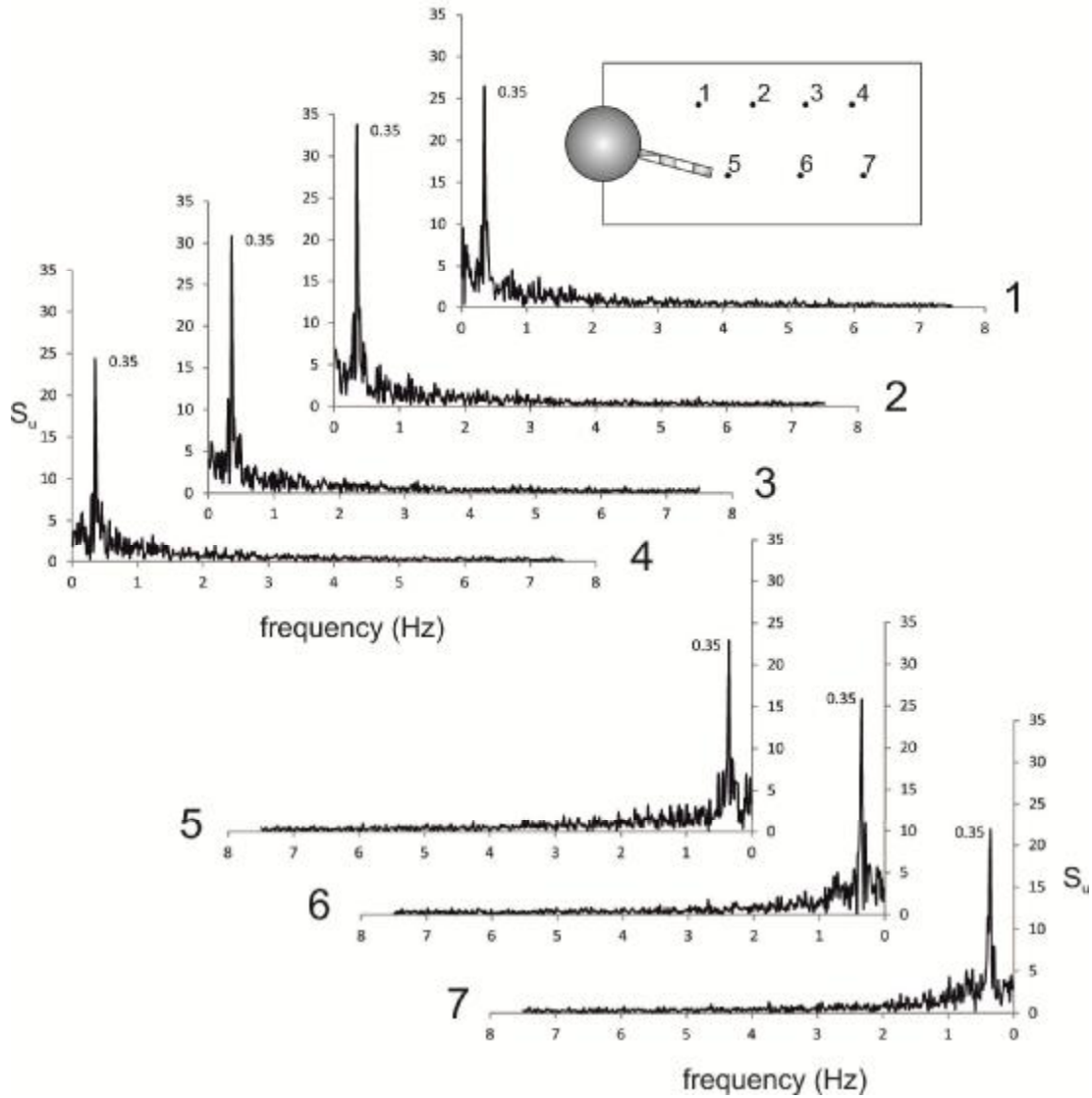


Figure 4.37. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.7$ at the plate angle of $\theta=15^\circ$

4.5.4. FFT Analysis for $\theta=30^\circ$ Cases

The effects of $\theta=30^\circ$ cases on vortex shedding frequency are presented in this section for points (shown in all figures of this section) 1, 2, 3 and 4 on the upper shear layer and 5, 6 for the lower shear layer which directly interacts with the trailing edge of permeable plate. The points 1, 2, 3, 4, 5, and 6 correspond to the coordinates of $(x/d, y/d) = (2.0, 2.0), (2.4, 2.0), (3.2, 2.0), (3.6, 2.0), (2.4, 1.0)$ and $(3.2, 1.0)$, respectively.

The spectra of streamwise velocity fluctuations for the solid plate are presented in Figure 4.38. A dominant peak of S_u having a frequency of $f=0.34$ is obvious on the upper shear layer and the amplitude of this peak gradually increases in downstream direction, namely from point 1 to 4. On the other side, the same frequency value, $f=0.34$ is also evaluated on lower shear layer, namely at points 5 and 6, where the amplitudes are greater compared to the upper shear layer. The evaluated frequency value is lower with respect to the natural cylinder case where a reduction of almost 11% in vortex shedding frequency is achieved by use of a solid plate with $\theta=30^\circ$ plate angle.

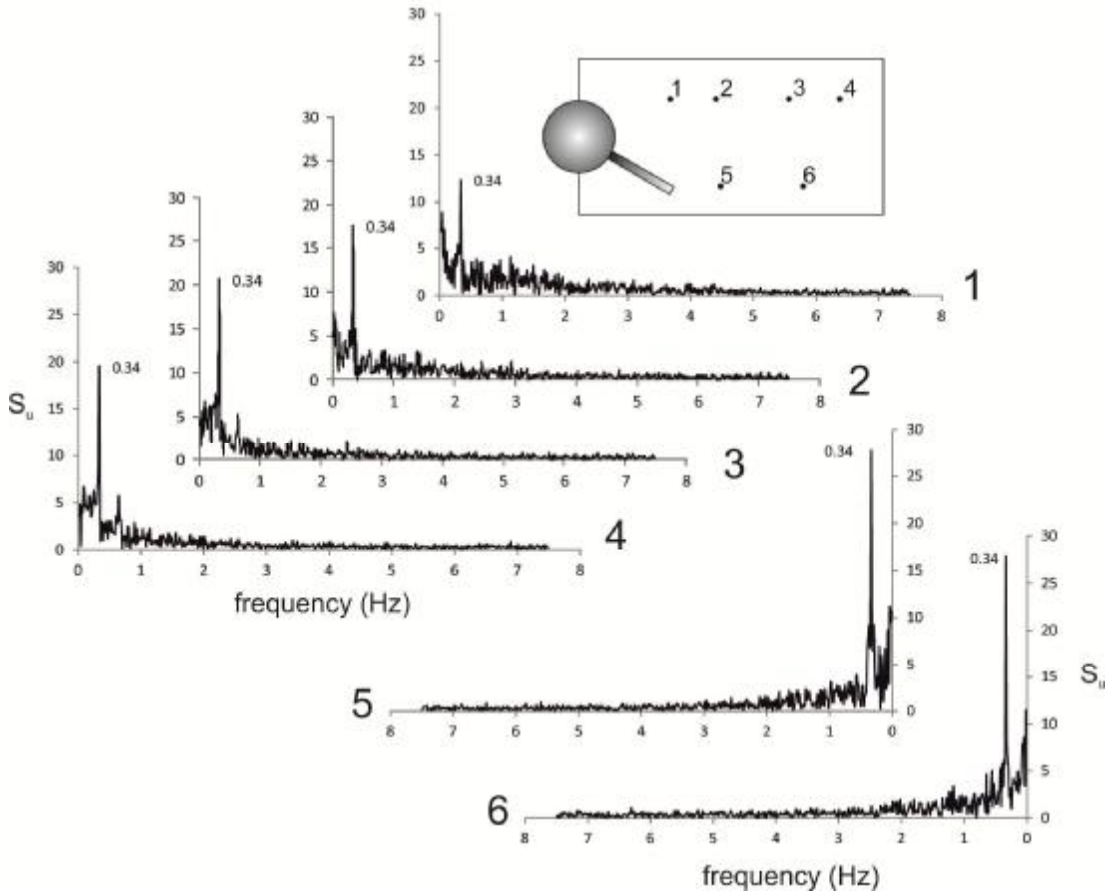


Figure 4.38. Spectra of streamwise velocity fluctuations for the solid plate ($\beta=0.0$) at the plate angle of $\theta=30^\circ$

Figure 4.39 shows the results of permeable plate having $\beta=0.4$ porosity for the determined points. A dominant peak of S_u having a frequency of $f=0.367$ is evaluated in the near wake of the cylinder on the upper shear layer (at points 1 and 2) although the amplitudes are low. Another dominant peak is evident on further downstream regions of upper shear layer, namely points 3 and 4 which has a frequency of $f=0.352$ with larger amplitudes, furthermore this frequency is also evaluated on lower shear layer at points 5 and 6 in which the amplitudes are greater compared to the frequency evaluated on upper shear layer. It should be noted that the frequency values evaluated for the $\beta=0.4$ are still lower than that of for the natural cylinder frequency.

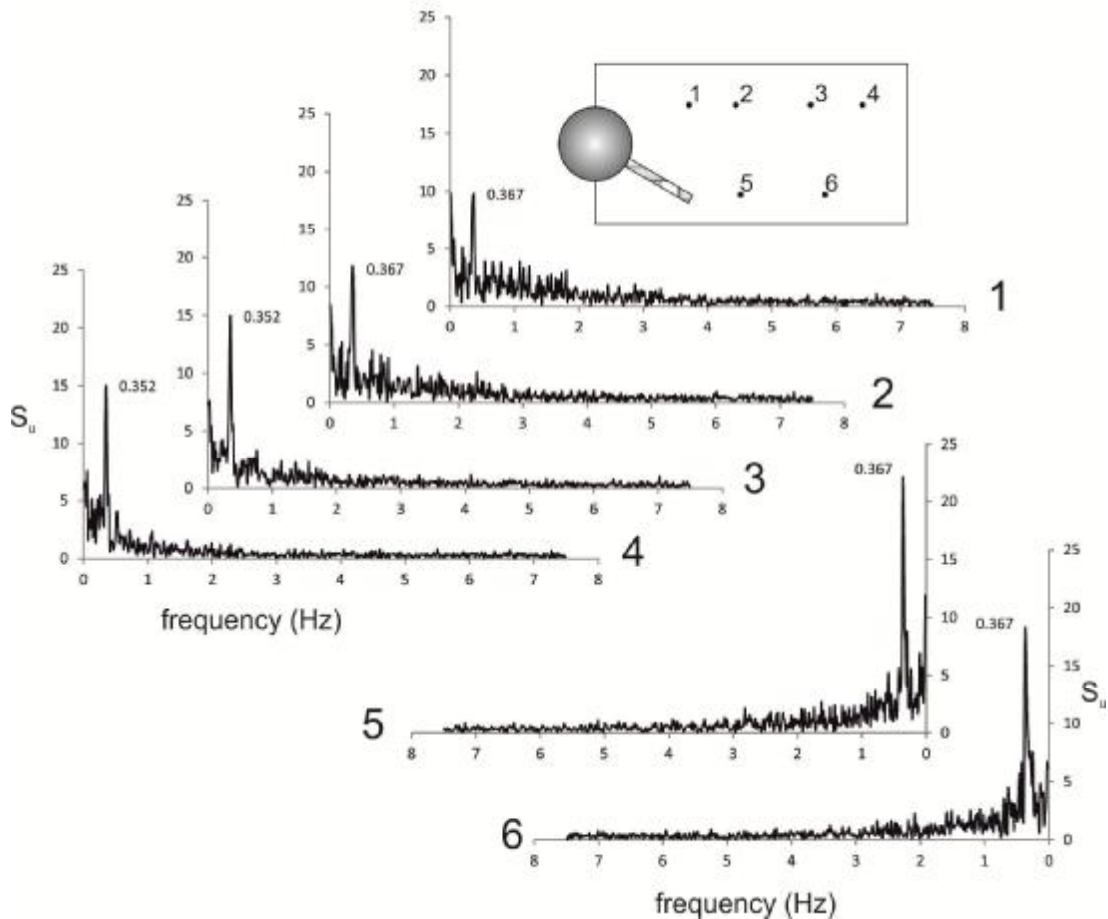


Figure 4.39. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.4$ at the plate angle of $\theta=30^\circ$

When compared to both the $\beta=0.0$ and 0.4 cases, the porosity of $\beta=0.5$ is more effective on reducing the vortex shedding frequency as seen by Figure 4.40 in which no dominant peak, hence a frequency is observed for points 1, 2 and 5 which are the points on upper and lower shear layers in the near wake of the cylinder. On further locations in downstream direction, namely points 3 and 4 on the upper shear layer, there exists a frequency value of $f=0.34$ which is the lower value for this plate angle. Moreover, the frequency of $f=0.34$ is also observed for the point 6 which is on the lower shear layer in the direction of trailing edge of the permeable plate. It can be concluded by these results that using a permeable plate of $\beta=0.5$ with $\theta=30^\circ$ is effective on eliminating the vortex shedding and consequent fluctuations in the near wake, however the shedding of the vortices takes place in further downstream locations.

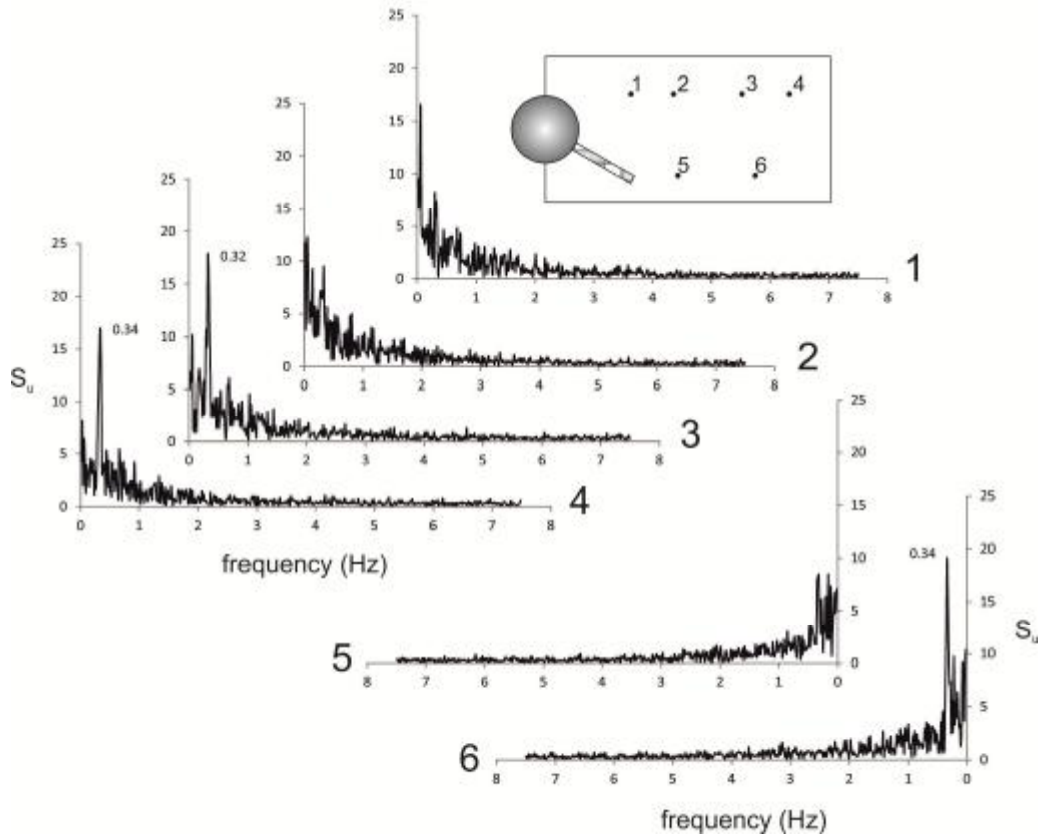


Figure 4.40. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.5$ at the plate angle of $\theta=30^\circ$

Increasing the porosity value to $\beta=0.6$ for this plate angle results in existence of dominant peaks with a frequency of $f=0.35$ for all points in downstream direction as seen by Figure 4.41. Furthermore, the amplitudes of the peaks are closer to each other in all points which means that a vortex shedding with a frequency of $f=0.35$ takes place in all downstream directions.

The results of $\beta=0.7$ are almost similar with the porosity value of $\beta=0.6$ as can be seen by comparing Figures 4.41 and 4.42 where the spectra of streamwise velocity fluctuations of $\beta=0.7$ are presented in Figure 4.42. The vortex shedding frequency of $f=0.35$ is also evaluated for $\beta=0.7$ which results in unsteady vortex shedding in the wake of the cylinder. However, the higher amplitudes are evaluated at points 1, 2 and 5 and gradually diminish at farther points of 3, 4 and 6.

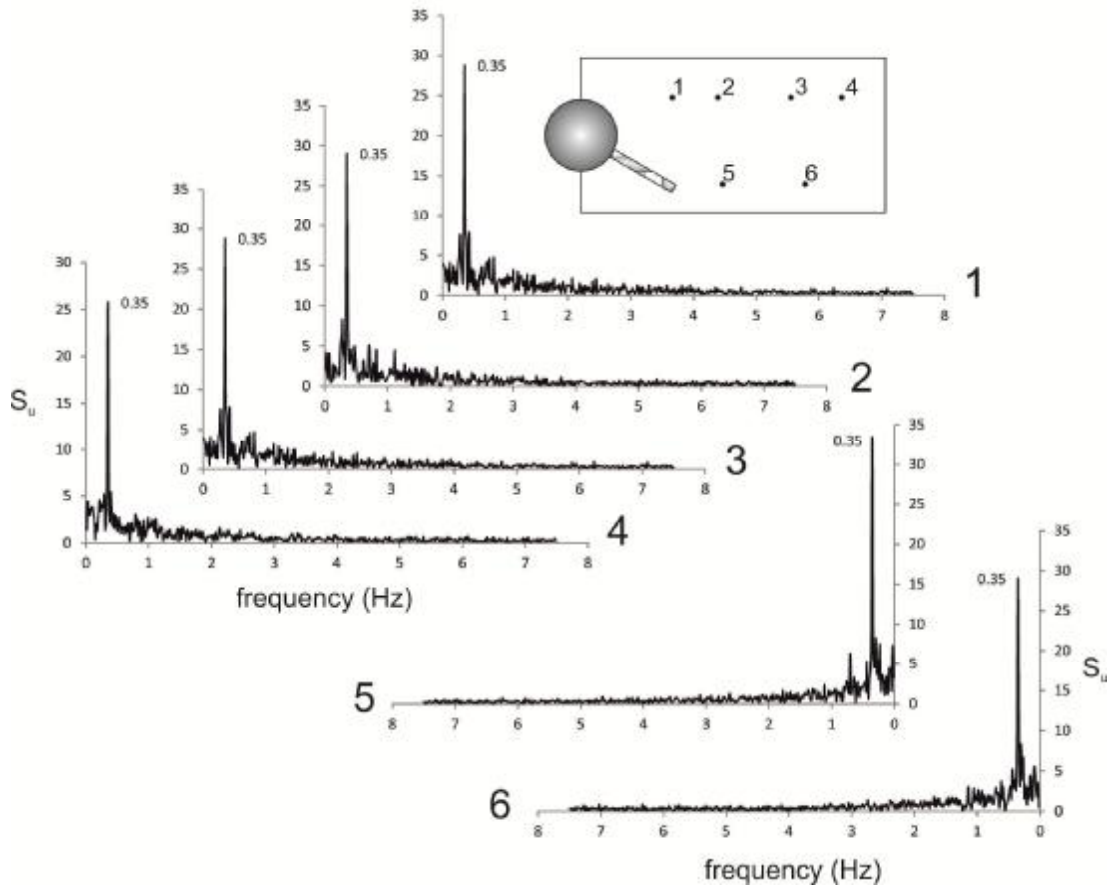


Figure 4.41. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.6$ at the plate angle of $\theta=30^\circ$

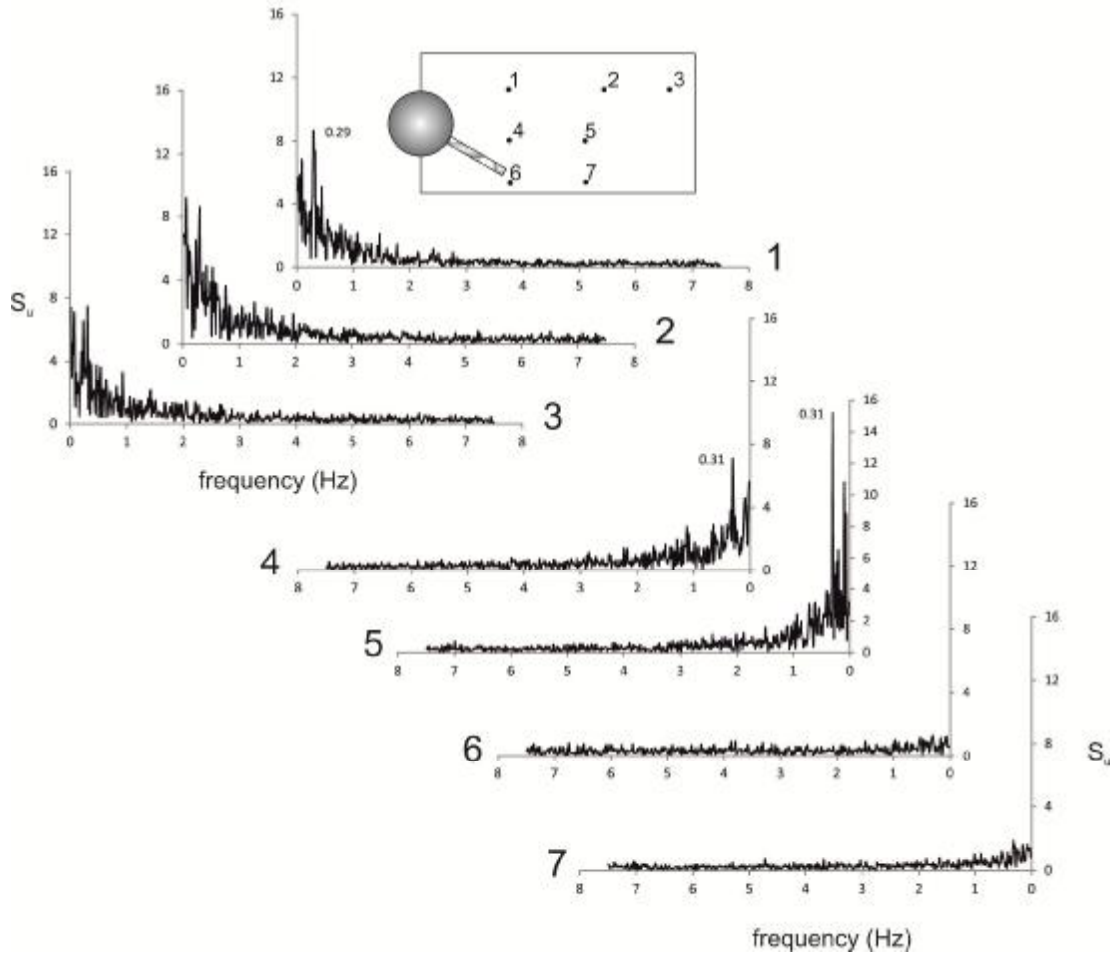


Figure 4.42. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.7$ at the plate angle of $\theta=30^\circ$

It can be concluded by the results of $\theta=30^\circ$ plate angle that the most effective porosity on reducing and eliminating the vortex shedding is $\beta=0.5$ in which occurrence of a dominant peak, hence the existence of vortex shedding is disappeared in the near wake of the cylinder, but occurred in farther downstream directions with a lower frequency value compared to the natural cylinder case.

4.5.6. FFT Analysis for $\theta=45^\circ$ Cases

Different than the smaller angles, there are three different shear layer for $\theta=45^\circ$ since the fluid starts to flow through the permeable plates at this angle of the permeable plate. Therefore, the effects of $\theta=45^\circ$ cases on vortex shedding frequency are presented in this section for points (shown in all figures of this section) of 1, 2, and 3 on the upper shear layer, 4 on the lower shear layer and 5, 6 for the shear layer of trailing edge of the permeable plate. The points 1, 2, 3, 4, 5, and 6 correspond to the coordinates of $(x/d, y/d) = (2.4, 2.0), (2.8, 2.0), (4.0, 2.0), (1.5, 1.3), (2.4, 0.5)$ and $(4.0, 0.5)$, respectively.

The usage of a solid plate with $\theta=45^\circ$ plate angle is presented in Figure 4.43 by means of spectra of streamwise velocity fluctuations. A distinct peak with a frequency of $f=0.23$ is found on the upper shear layer at points 1, 2 and 3 by increasing its amplitude. On the lower shear layer of the cylinder, namely at point 4, there is no dominant peak since the plate is solid and no transition takes place through the solid plate. At points 5 and 6, which are on the shear layer generated by trailing edge of the plate, a frequency of $f=0.23$ and $f=0.22$ exist, respectively. The slight difference in between the points of 5 and 6 is due to the highly unsteady flow by the existence of solid plate.

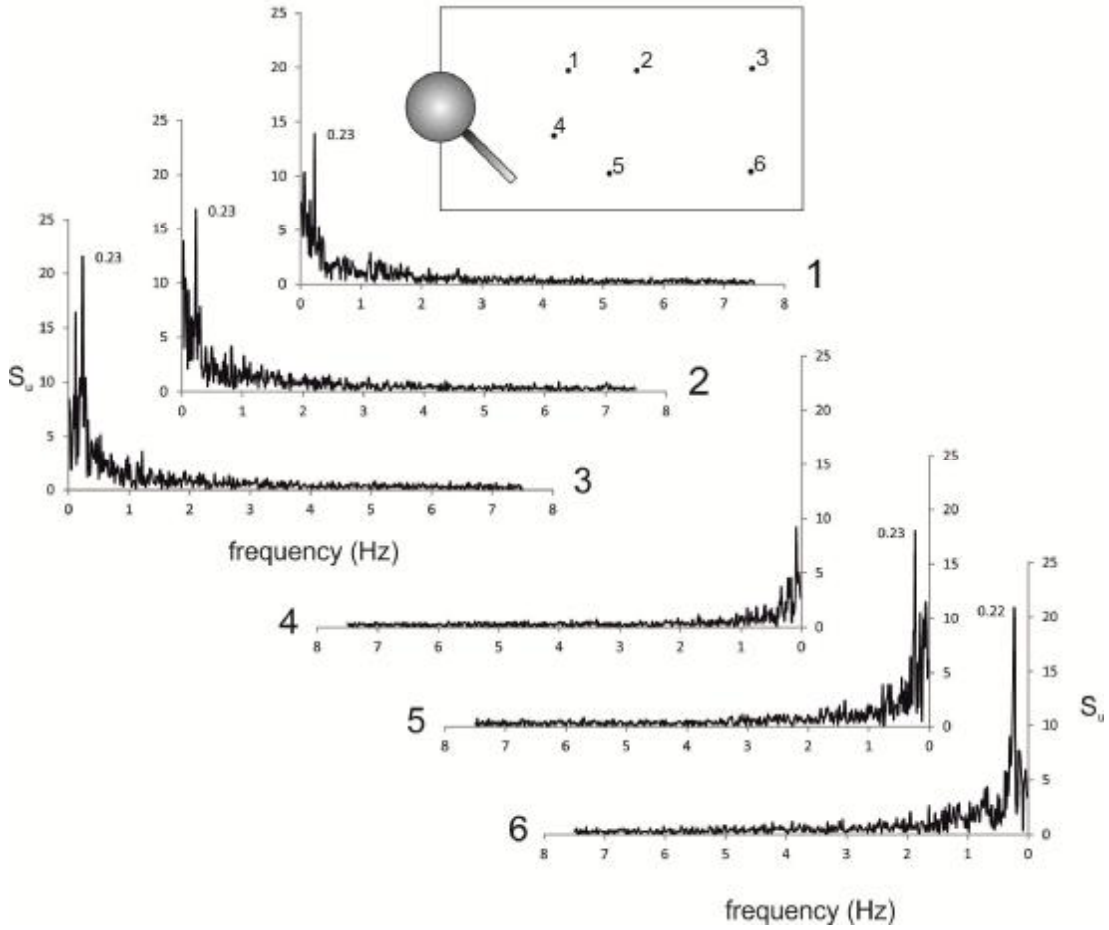


Figure 4.43. Spectra of streamwise velocity fluctuations for the solid plate ($\beta=0.0$) at the plate angle of $\theta=45^\circ$

The case where the permeable plate takes place having a porosity value of $\beta=0.4$ is presented in Figure 4.44. Using a permeable plate would give chance to the fluid to pass through the meshes, so this will increase the momentum in lower shear layer whereas the momentum of the fluid passing from the trailing edge of the plate reduces. Finally, the formation of vortices is significantly affected. This effect is clearly seen by looking at the spectra of streamwise velocity fluctuations for all points in which no dominant peak occur downstream of the cylinder. The vortex shedding is completely eliminated downstream of the cylinder by using a permeable plate of $\beta=0.4$ for $\theta=45^\circ$ plate angle. However, a small amplitude peak is observed at point 6 having a frequency of $f=0.34$. Since the amplitude is very low compared with the natural cylinder, this frequency could be neglected.

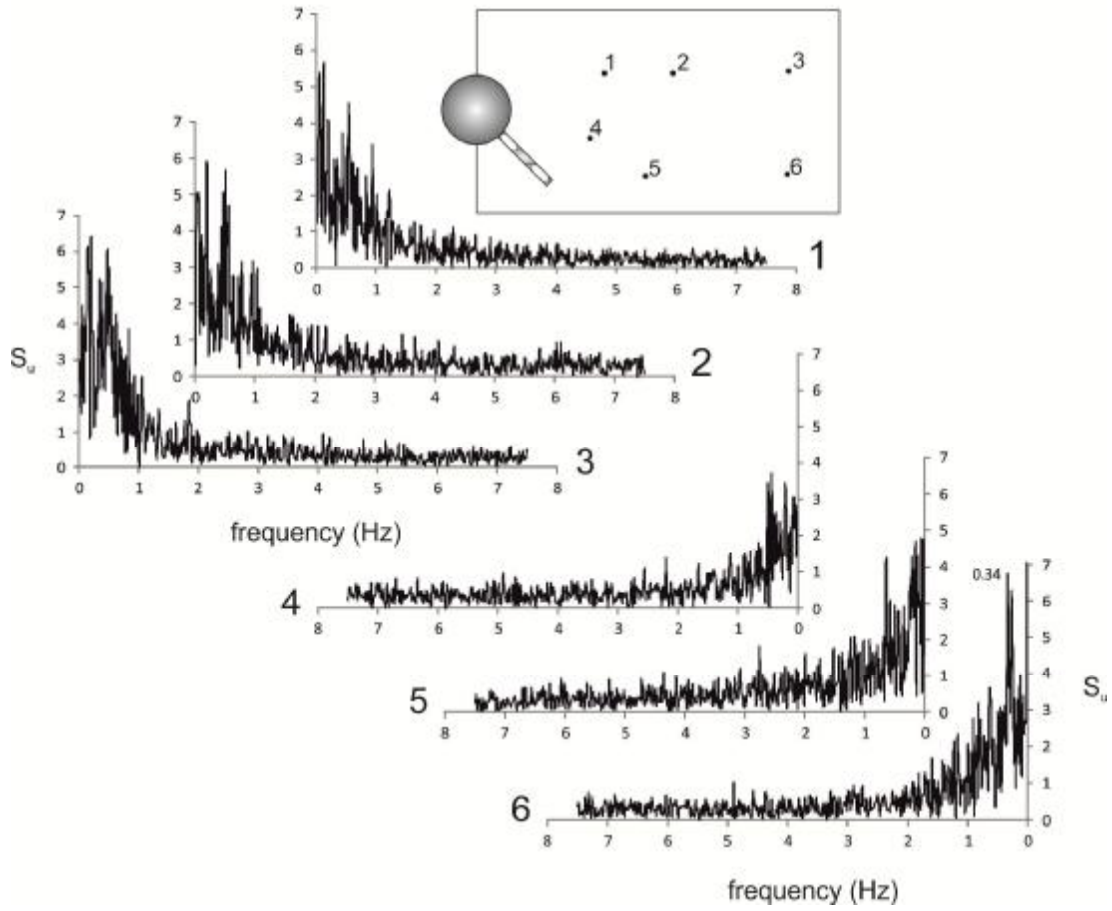


Figure 4.44. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.4$ at the plate angle of $\theta=45^\circ$

Almost the same results are evaluated for the porosity value of $\beta=0.5$ which is shown by Figure 4.45. No dominant peak is observed for the points of 1, 2, 3, 4 and 5 resulting in elimination of vortex shedding, however a small amplitude peak is observed at point 6 having a frequency of $f=0.25$. Since the amplitude is very low compared with the natural cylinder, this frequency could be neglected.

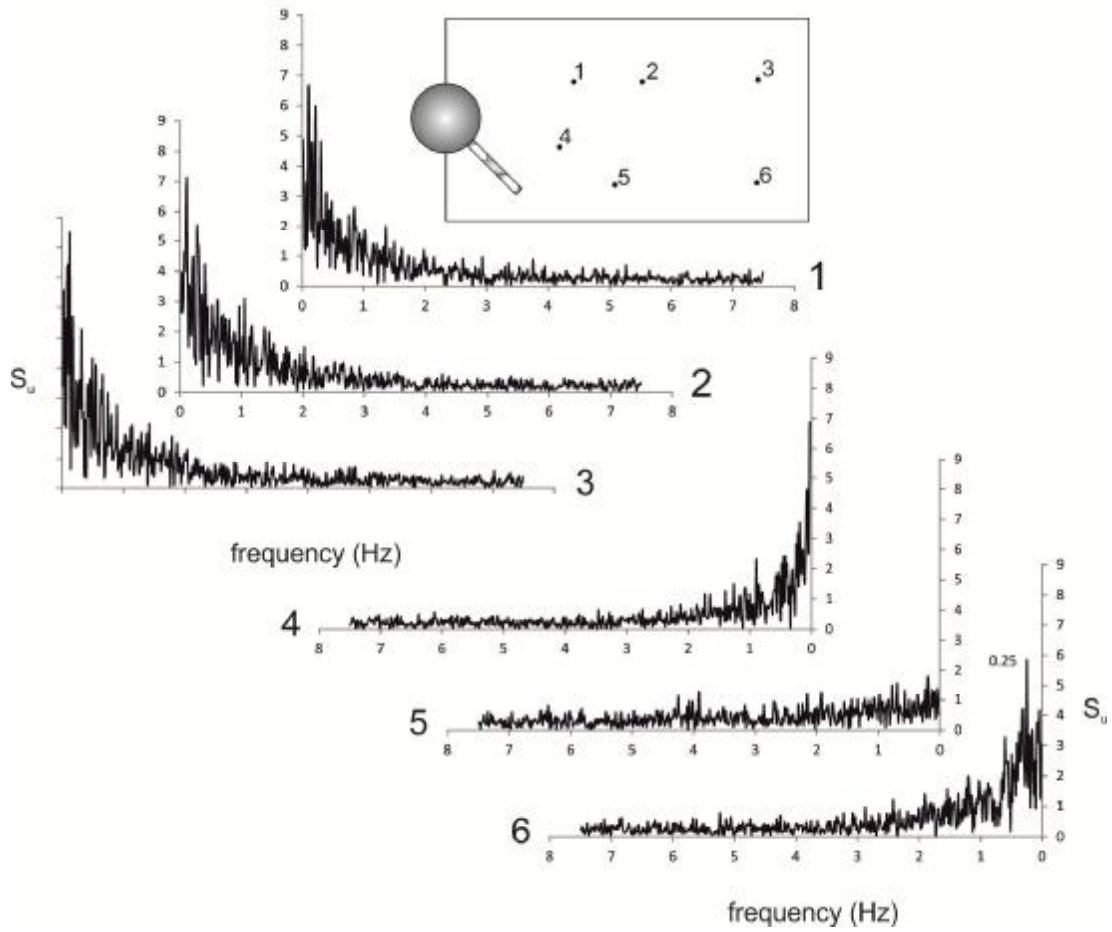


Figure 4.45. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.5$ at the plate angle of $\theta=45^\circ$

The results of a further increase in porosity value to $\beta=0.6$ is presented in Figure 4.46. It can be said that the permeable plate is still effective on eliminating the frequency downstream of the cylinder for this plate angle. There is no dominant peak evident for points of 1, 2, 4 and 5 which means no periodic action takes place within the near wake of the cylinder, however very small amplitudes are observed for the points of 3 and 4 where the frequencies are determined to be $f=0.35$ and $f=0.32$, respectively.

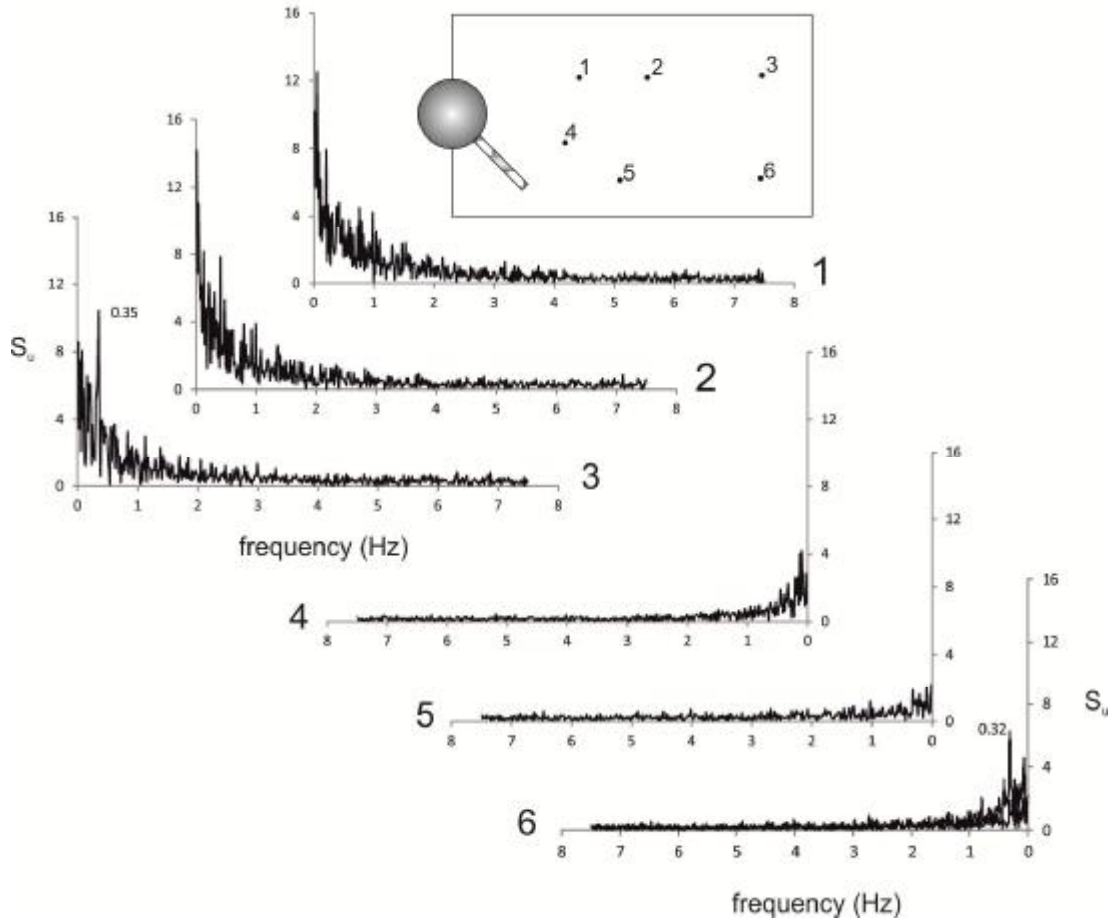


Figure 4.46. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.6$ at the plate angle of $\theta=45^\circ$

Figure 4.47 includes the spectra of streamwise fluctuations for the porosity value of $\beta=0.7$. Here it should be noted that the points are changed according to the high porosity ratio. For $\beta=0.7$, the mesh sizes get their largest value, therefore ineffectiveness of the permeable plate exists for this case which is clearly observed by looking at the frequencies evaluated in Figure 4.47. A dominant peak having a frequency of $f=0.29$ was evaluated at points 1, 2, 4 and 5 as can be seen in the figure. No frequency is observed on the lower shear layer as obvious by the results of point 3. It can be concluded that usage of a permeable plate having a porosity of $\beta=0.7$ is not adequate to completely eliminate the vortex shedding; however it reduces the frequency of the vortex shedding, hence the vortex induced vibrations.

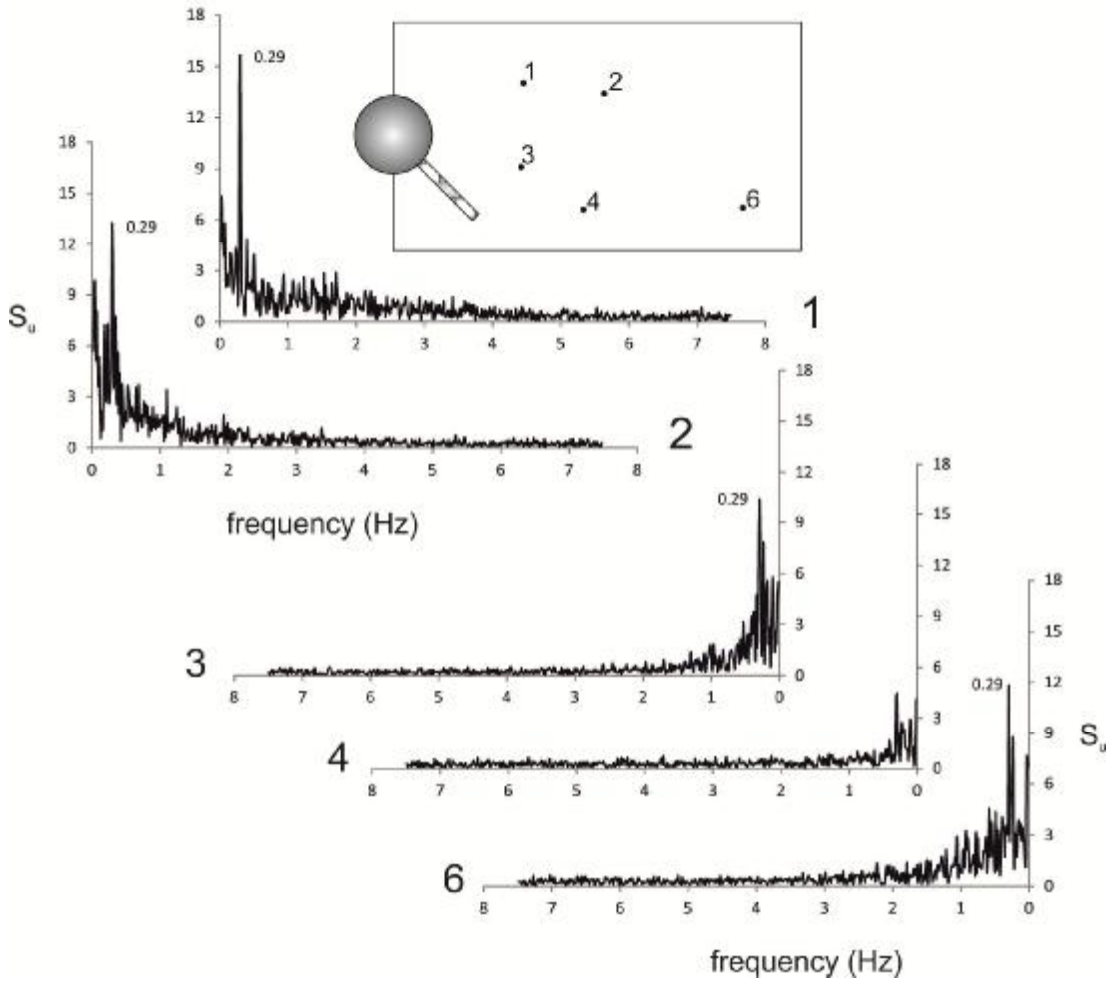


Figure 4.47. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.7$ at the plate angle of $\theta=45^\circ$

4.5.7. FFT Analysis for $\theta=60^\circ$ Cases

Similar physical condition with the $\beta=0.4$ cases is valid for the plate angle of $\theta=60^\circ$ in which the fluid starts to flow through the permeable plates at this angle of the permeable plate. Therefore, the effects of $\theta=60^\circ$ plate angle on suppression of vortex shedding frequency are presented in this section for points (shown in all figures of this section) of 1, 2, and 3 on the upper shear layer, 4 and 5 on the lower shear layer and 6, 7 for the shear layer of trailing edge of the permeable plate. The points 1, 2, 3, 4, 5, 6 and 7 correspond to the coordinates of $(x/d, y/d) = (1.6, 2.0)$, $(3.2, 2.0)$, $(4.0, 2.0)$, $(1.6, 1.0)$, $(1.4, 1.0)$, $(1.6, 0.2)$ and $(2.4, 0.2)$, respectively.

Figure 4.48 presents the results of solid plate in which it is clearly seen that a dominant peak with a various amplitude frequency of $f=0.21$ is valid for almost all points except point 4 which is on the lower shear layer of the cylinder. This lower frequency value depicts that using a solid plate having a plate angle of $\theta=60^\circ$ would result in reduced frequency of vortex shedding, as well as the vibrations.

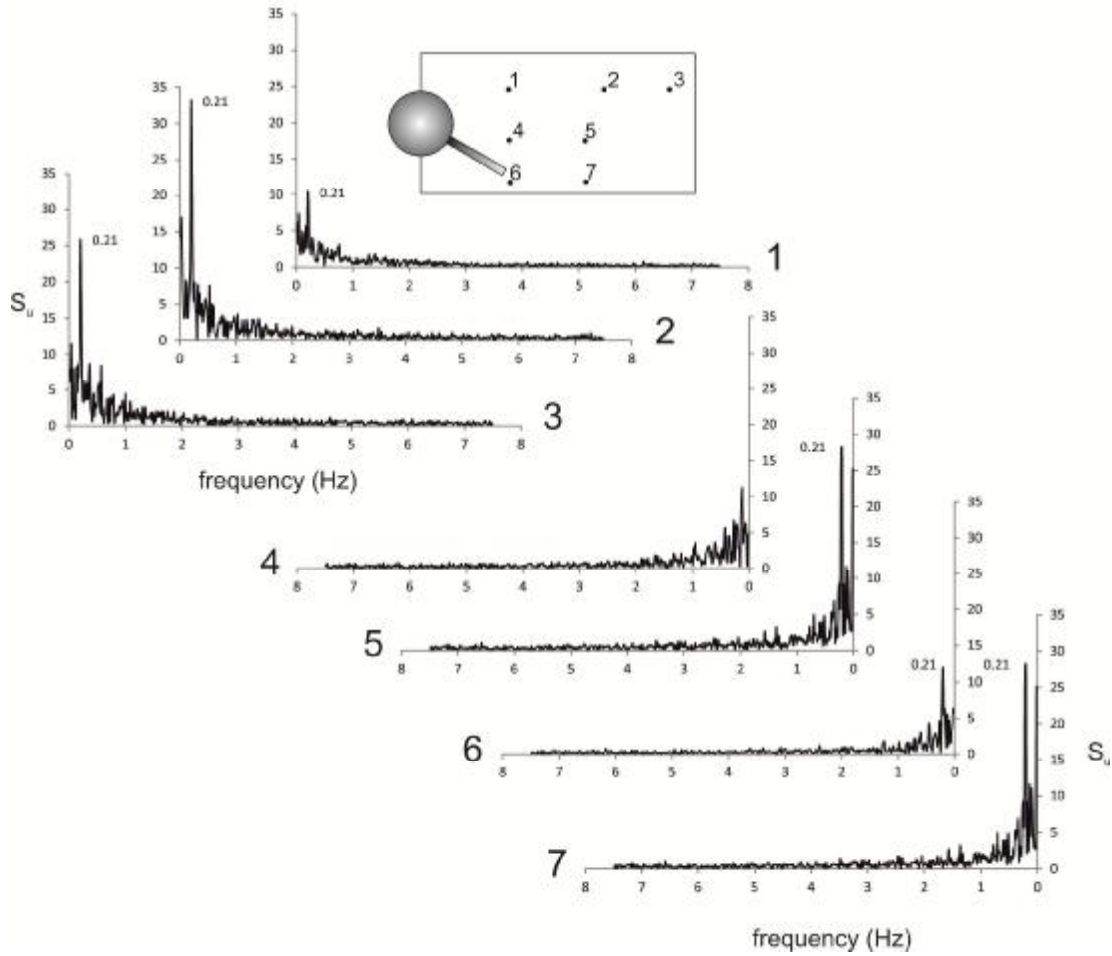


Figure 4.48. Spectra of streamwise velocity fluctuations for the solid plate ($\beta=0.0$) at the plate angle of $\theta=60^\circ$

The use of a permeable plate with the porosity value of $\beta=0.4$ completely suppresses the shedding of the vortices as easily seen by the Figure 4.49 in which there exists no dominant peak for all points. The results of spectra of streamwise velocity fluctuations are in good agreements with the previous results where this case is one of the most effective cases on controlling the flow.

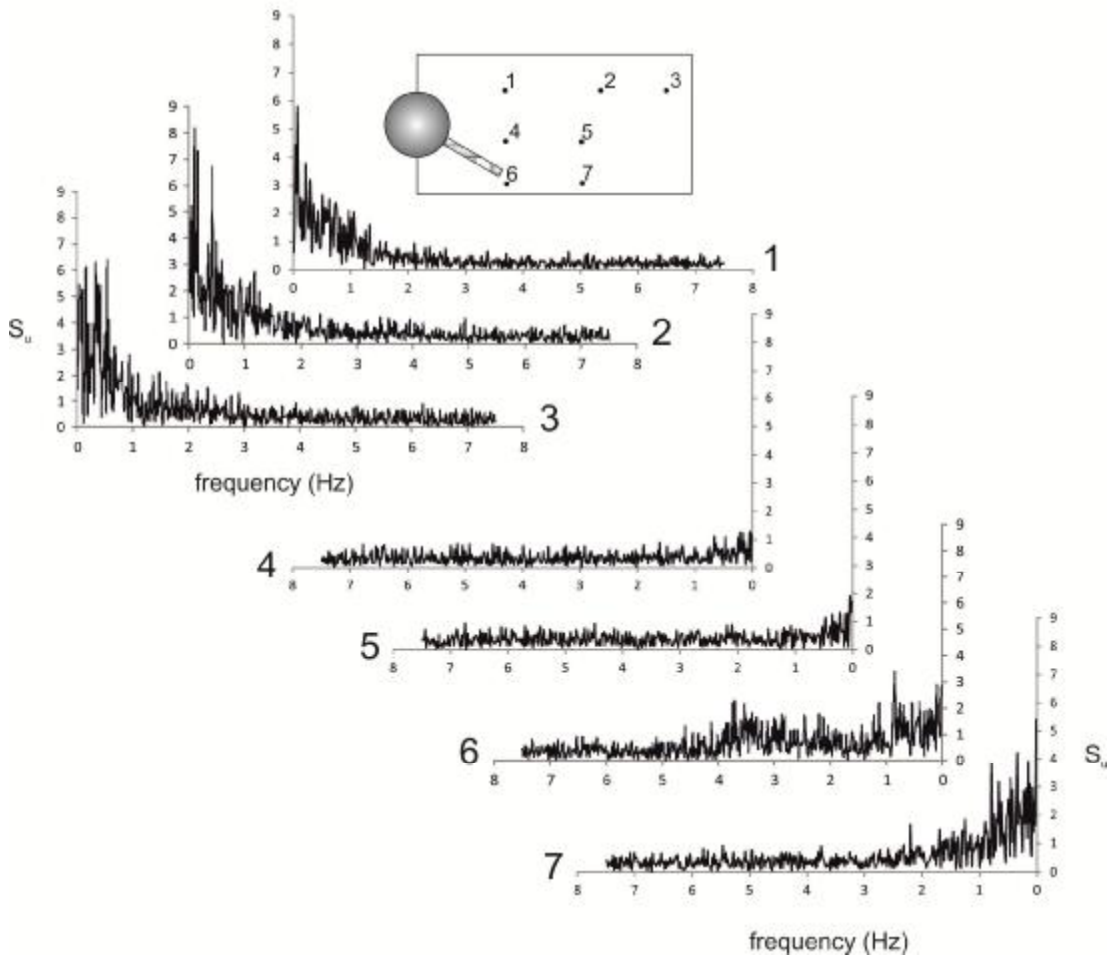


Figure 4.49. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.4$ at the plate angle of $\theta=60^\circ$

Figure 4.50 presents the spectra of streamwise velocity fluctuations for $\beta=0.5$ porosity which are almost same with the porosity of $\beta=0.4$ where no dominant frequency is evident in all points which means the unsteady vortex shedding is completely eliminated downstream of the cylinder.

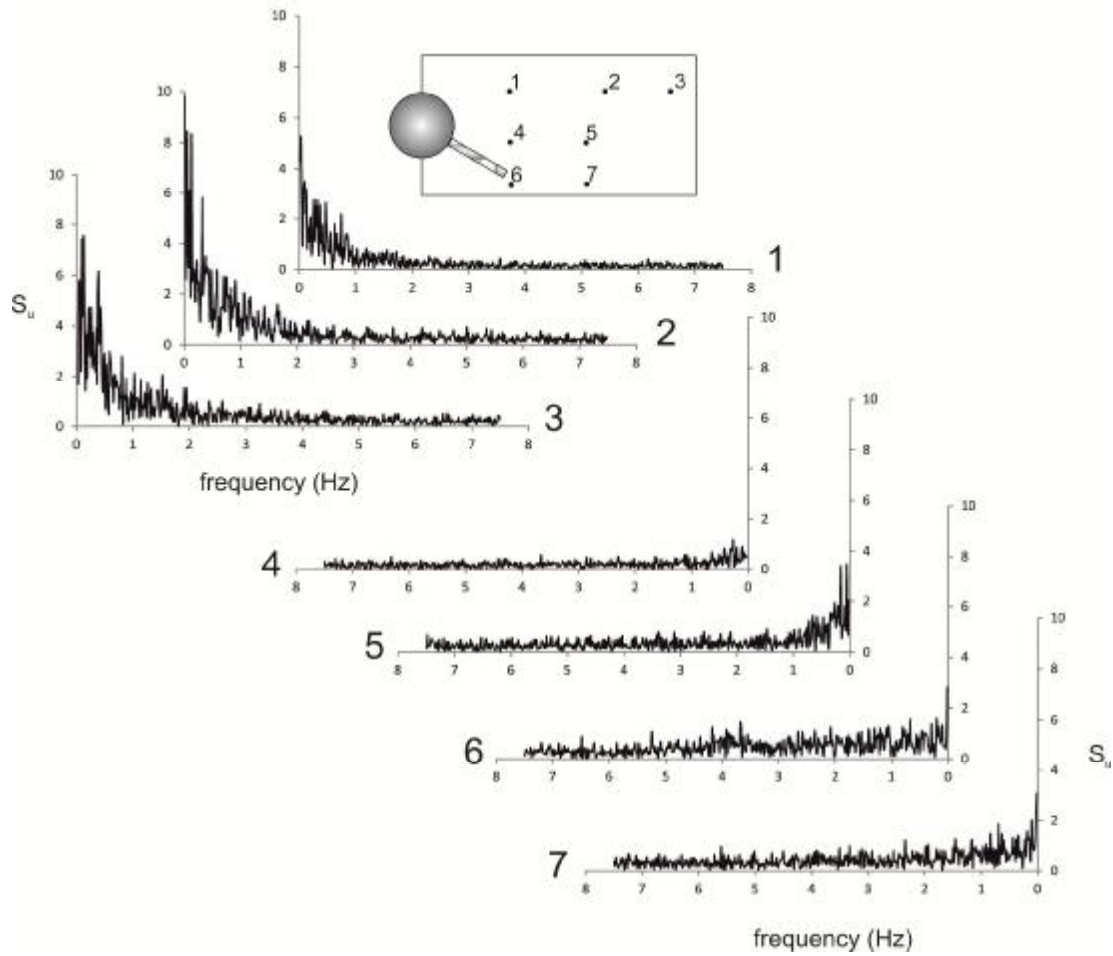


Figure 4.50. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.5$ at the plate angle of $\theta=60^\circ$

The permeable plate is still effective on elimination of vortex shedding even the porosity increases to $\beta=0.6$ as obviously seen by Figure 4.51. No frequency is evaluated at pre-determined points, therefore the vortex shedding is completely eliminated, and hence the vortex induced vibrations are annihilated.

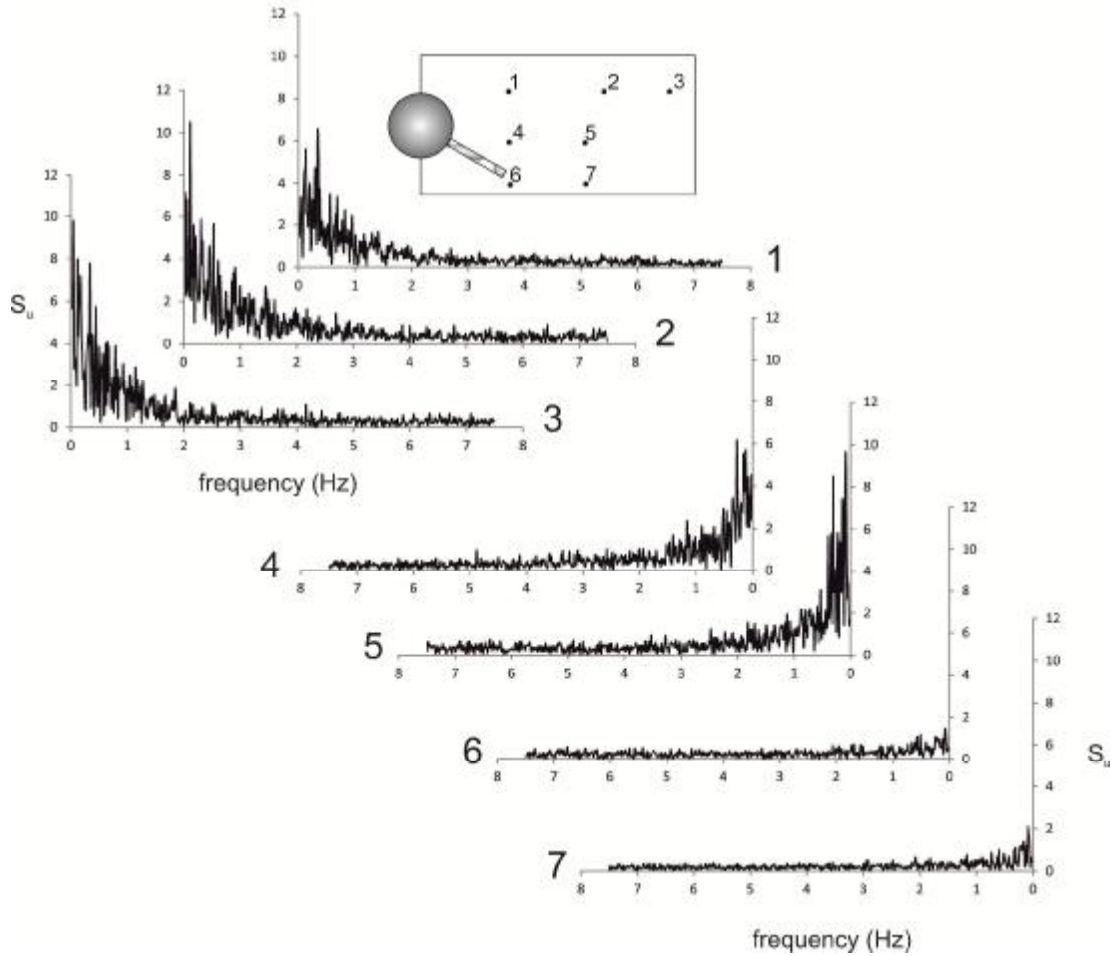


Figure 4.51. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.6$ at the plate angle of $\theta=60^\circ$

As expected by the previous results, increasing the porosity in this plate angle would result in a dominant frequency especially in the near wake. The results are presented in Figure 4.52 which are evaluated for $\beta=0.7$ porosity. A dominant peak with a frequency value of $f=0.29$ is evident on the upper shear layer, i.e. in the near wake corresponding to the occurrence of vortex shedding. A higher peak is observed on the lower shear layer which is evaluated as $f=0.31$ and increase in its amplitude from point 4 to 5. Different than all the other cases discussed before, there are peaks on the lower shear layer, which is due to the high amount of fluid passed through the meshes since the size of the meshes increase by increasing porosity. It can be concluded by this result that $\beta=0.7$ is the transition point of lower shear layer to

generate a dominant peak, furthermore, the vortex shedding is now in between the vortices generated from the upper shear layer and the lower shear layer.

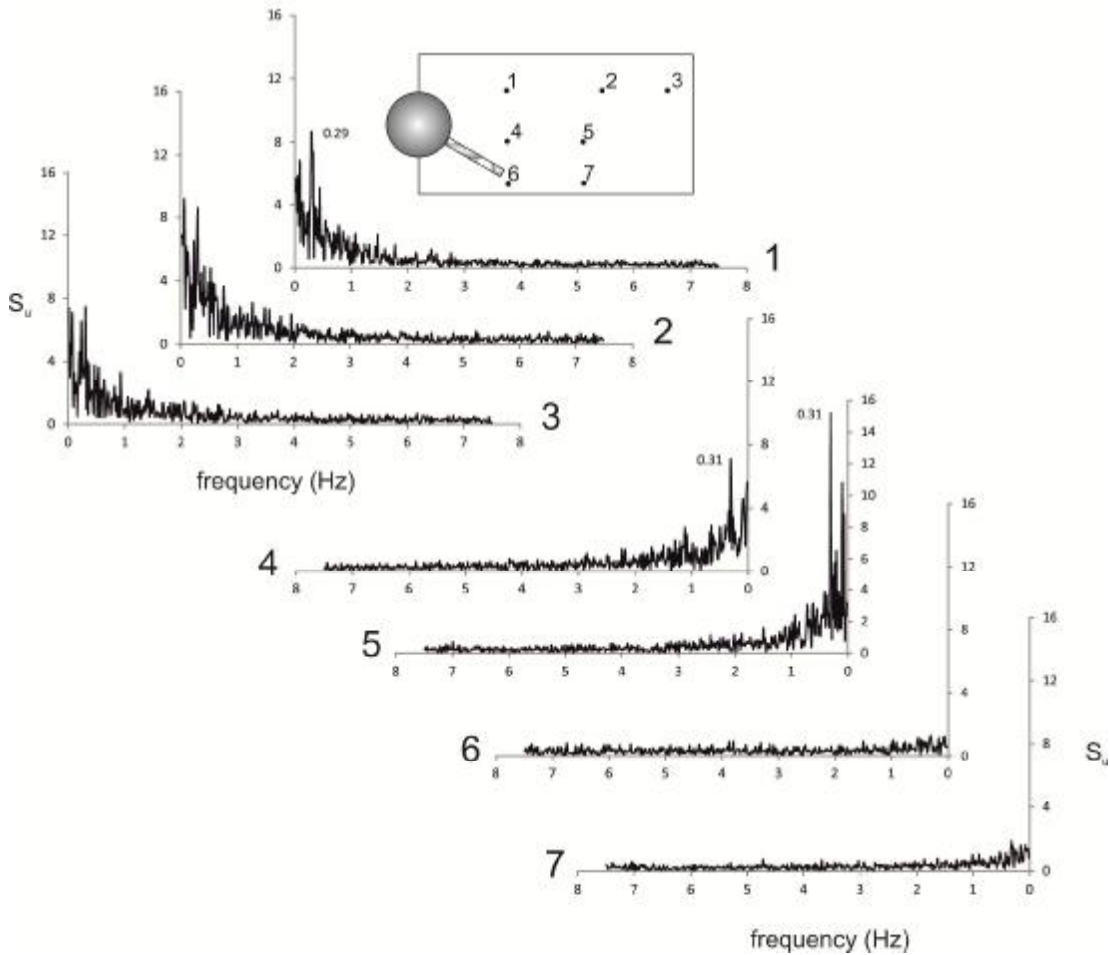


Figure 4.52. Spectra of streamwise velocity fluctuations for the permeable plate of $\beta=0.7$ at the plate angle of $\theta=60^\circ$

4.6. Special Case; Permeable Plate Attached on Separation Point

The separation point for laminar boundary layer is defined as $\alpha=80^\circ$ (starting with the stagnation point of the cylinder in clockwise direction) within the literature. In this study, the use of a permeable plate with $\theta=100^\circ$ coincides with the separation point. Therefore the control of the flow by “separation point control” takes place and should be analyzed in details. Throughout the thesis, the effect of permeable plates on flow control was mentioned occasionally; however no comparison between different porosity values were done. This section presents the results of both qualitative and quantitative measurements for permeable plate used at separation point of the cylinder.

Figure 4.53 presents the pictures of visualized flow field for natural cylinder, as well as the controlled cases for all porosity values. First of all, the rolling up of the vortices from upper and lower shoulder of the cylinder is obviously seen for the natural cylinder resulting in well-known Karman Vortex Street in its downstream direction. When the permeable plate having $\beta=0.4$ porosity takes place on the separation point of the cylinder, the flow structure dramatically changes where the rolling up of the vortices from upper and lower shear layer is eliminated and the vortex street completely disappeared in the current flow field. Similar behaviors are seen for the porosity value of $\beta=0.5$, however there exists high- frequency Kelvin-Helmholtz vortices on the upper shear layer of the cylinder, again with completely eliminated vortex shedding for the current flow field. The main mechanism of the vortex shedding suppression could be explained as follows; since the permeable plate is located at the separation point of the cylinder, the flow comes across with the plate and cannot be able to roll-up into the wake region and entrain high momentum from the shear layer. Besides, the permeable plate which is constructed from woven screen makes this high momentum flow to a regulated uniform structure. Therefore even the upper shear layer rolls-up into the wake region, it does not come across with the existence of a counter shear layer to be interacted and finally the formation and consequent interaction of the vortices are eliminated, finally the vortex shedding is accurately suppressed. However, this mechanism works accurately up to a certain

value of the porosity which is also the aim of current study. As the porosity increases, the open area on the permeable plate, i.e. the sizes of the mesh increases, too. The picture for the porosity of $\beta=0.6$ shows this truth, as being the transition porosity from a complete suppression to attenuation of the vortex shedding. For $\beta=0.6$, more fluid passes through the meshes and the momentum entrainment into the wake from the lower shear layer slightly starts. As a result the interaction and shedding of the vortices in further downstream locations take place for this porosity value. It is expected that as $\beta=0.6$ is the transition value for flow control, further increase in porosity would result in obvious vortex shedding downstream of the cylinder as seen in Figure 4.53 for $\beta=0.7$ case. The rolling up of the vortices from upper and lower shear layer starts and the vortices are now being interacted closer to the rear stagnation point of the cylinder compared to the previous porosity value, $\beta=0.6$. The visualization picture for $\beta=0.8$ is similar with the previous porosity value, however the vortex shedding starts near to the cylinder base as observed for the natural cylinder. Although the case of $\beta=0.7$ and 0.8 are not useful for a complete suppression of the vortex shedding, they are effective on shifting the vortex formation region to farther regions downstream of the cylinder which is also an important issue on flow control around bluff bodies.

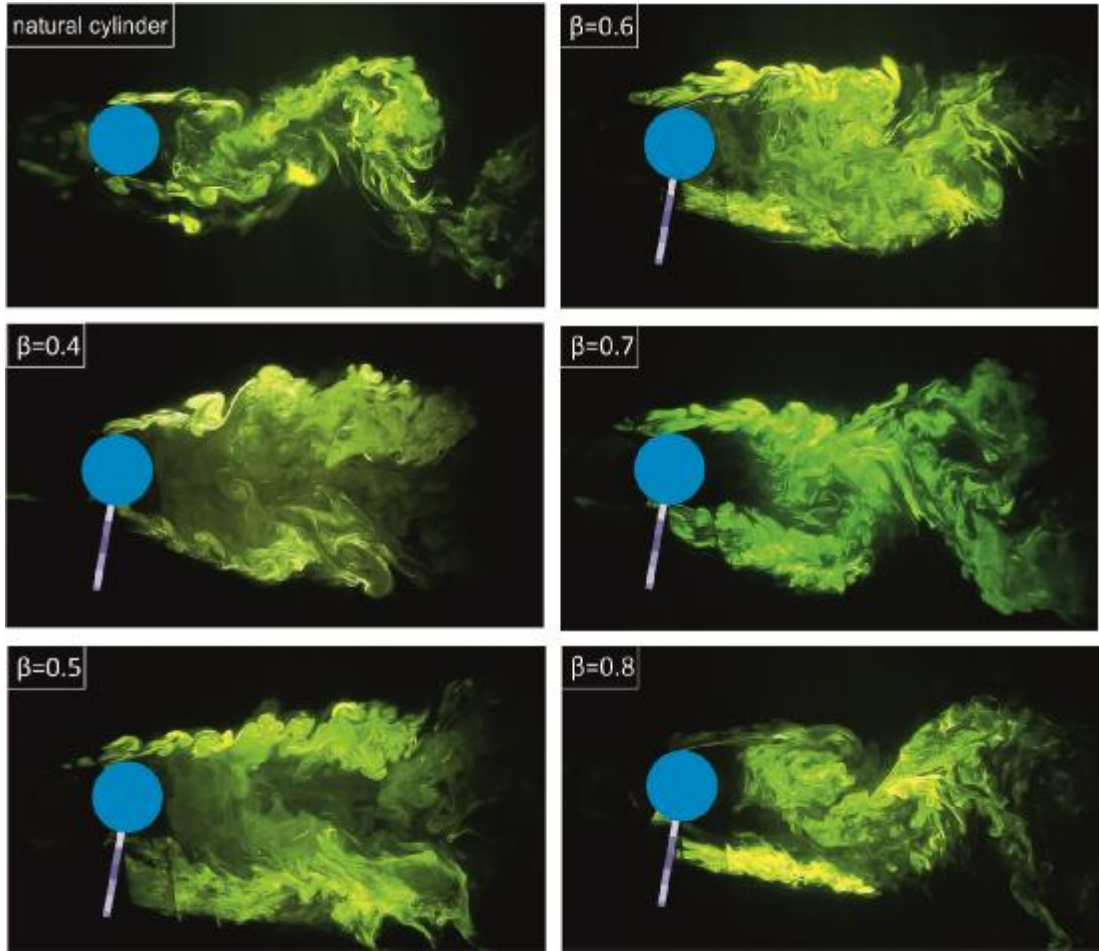


Figure 4.53. Dye visualization pictures for natural cylinder and all porosities

Even the flow visualization results show the effect of permeable plate on flow control of unsteady vortical structure; it is important to decide whether the permeable plate is useful or not by analyzing quantitative PIV results. Figure 4.54 presents the time averaged Reynolds Shear Stress concentrations, $\langle u'v' \rangle$, vorticity contours, $\langle \omega \rangle$ and streamline topologies, $\langle \psi \rangle$ for all cases, as well as the natural cylinder. The blue and red lines present the negative and positive values, respectively for Reynolds shear stress and vorticity contours. The incremental value of Reynolds shear stress concentrations was taken as 0.01 where it was taken as 4 s^{-1} for vorticity contours. There exist small-scale clusters of Reynolds shear stress in the near wake of cylinder where the large-scale ones are formed in far wake. Decreasing the intensity of Reynolds shear stress means the prevention of passing momentum from one layer to another, in other words; fluctuations. It is clearly seen from Figure 4.54 that using a

permeable plate attached on the separation point of cylinder significantly reduces the concentrations of Reynolds shear stress, especially for $\beta=0.4$ and 0.5 . Furthermore, the formation of Reynolds shear stress contours exists in farther regions according to the base of the cylinder compared with the natural cylinder case. The positive Reynolds shear stress contours are almost disappeared with respect to the negative ones which mean that the permeable plate dramatically reduces the fluctuations in lower shear layer while also effectively reducing in the upper shear layer, i.e. when compared with the natural cylinder the reduction in maximum Reynolds shear stress value for lower shear layer is found to be 85% and 86% for $\beta=0.4$ and 0.5 , respectively. However, the reductions are found to be 63% and 59% for $\beta=0.4$ and 0.5 , respectively for the upper shear layer. Increasing the porosity to $\beta=0.6$ results in a slight increase in concentrations of Reynolds shear stress which means the permeable plate starts to lose its efficiency on flow control, however the percent reductions in maximum values are still significant; 52% for upper and 79% for lower shear layer compared with the natural cylinder. It was depicted that the cases of $\beta=0.7$ and 0.8 were not able to completely eliminate the vortex shedding downstream of the cylinder, however they were effective on increasing the vortex formation region. The Reynolds shear stress concentrations for $\beta=0.7$ and 0.8 shows that the intensity of the fluctuations increases, especially for $\beta=0.8$, however the values are still lower with respect to the values of natural cylinder. The reductions of 45% and 40% is achieved on the upper shear layer for $\beta=0.7$ and 0.8 , respectively where the reductions for the lower shear layer are evaluated as 56% and 27% $\beta=0.7$ and 0.8 , respectively.

The second column of Figure 4.54 presents the time averaged vorticity contours where the upper and lower shear layers get closer to each other for the natural cylinder which corresponds to the interaction of shear layers at the location of closing up. Therefore it is important to eliminate this close up of the shear layers to each other in order to control and suppress the vortex shedding. As can be seen by the vorticity results of $\beta=0.4$ and 0.5 , the upper and lower vorticity contours do not get close to each other, while the upper shear layer elongates in downstream direction, the lower vorticity layer moves away from the centerline of the cylinder, this is why the vortex shedding is completely suppressed. Furthermore, the

magnitudes of the vortices are reduced compared to the natural cylinder. The reduction in lower vorticity layer gets its maximum value for $\beta=0.5$ case which means that almost no vortical structure takes place for the lower shear layer. The permeable plate of $\beta=0.6$ generates a similar vortical structure with the previous porosity values, i.e. both upper and lower shear layer do not tend to move into the wake region and elongates in downstream direction. Increasing the porosity value to $\beta=0.7$ decreases the elongation in the shear layers while increasing the vorticity magnitudes compared with the lower porosities. Finally, for the porosity of $\beta=0.8$, the vorticity contours get a similar shape with the natural cylinder where the vorticity layers get closer to each other in the wake region and the lengths of the layers decrease compared with the lower porosity values. Furthermore, the lower vorticity layer is smaller in length with respect to the lower vorticity layer of the natural cylinder due to the presence of permeable plate.

The time-averaged streamline topologies shown in the third column of Figure 4.54 clearly presents how the flow structure affected by presence of the permeable plate. The results of the natural cylinder are similar with the literature where two symmetrical focal points and a saddle point exist in the wake of the cylinder. To control the unsteady vortical flow downstream of the cylinder, it is important to increase the wake length to farther downstream locations. For $\beta=0.4$, there exist three focal points while the saddle point which is supposed to be formed behind the focal points cannot be seen throughout current flow field. The similar flow structure exists for the porosity of $\beta=0.5$ except that it has two focal points in the wake region. For both $\beta=0.4$ and 0.5 cases, both wake widths and vortex formation lengths are significantly increased which is an indication of effective control of the unsteady flow structure downstream of the cylinder. This is the case for higher porosities of $\beta=0.6$ and 0.7 , however, the increase in the vortex formation length slightly decreases by increasing porosity as clearly seen by Figure 4.54. The time-averaged streamlines for $\beta=0.8$ is very similar with the ones belong to the natural cylinder, which means this porosity value is not effective on controlling the flow.

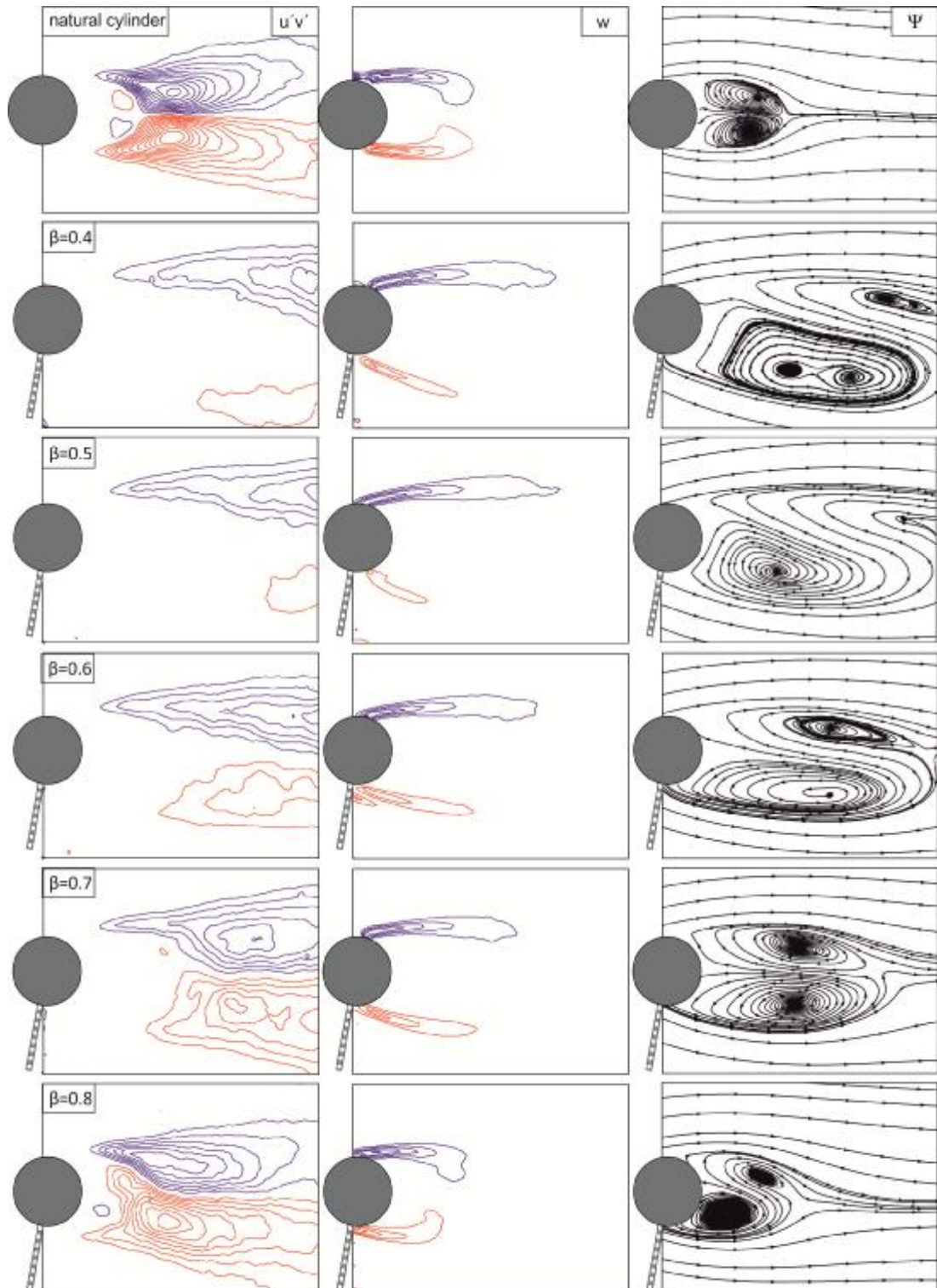


Figure 4.54. Time averaged Reynolds Shear Stress concentrations, $\langle u'v' \rangle$, vorticity contours, $\langle \omega \rangle$ and streamline topologies, $\langle \psi \rangle$ for all cases, as well as the natural cylinder

The spatial distribution of turbulent kinetic energy (TKE), streamwise Reynolds normal stress $\langle u'u' \rangle$, spanwise Reynolds normal stress $\langle v'v' \rangle$ are presented in Figure 4.55 for all cases, as well as the natural cylinder. These parameters have been made dimensionless by dividing them with the square of the depth-averaged free-stream velocity, U^2 for better comparisons. Both minimum and incremental values were taken as 0.01 for TKE, $\langle u'u' \rangle$ and $\langle v'v' \rangle$. The distribution of TKE, which is shown in the first column of Figure 4.55 shows that the fluctuations in the wake region dramatically decreases by use of a permeable plate, especially for the porosities of $\beta=0.4$ and 0.5 . The peak generated approximately $1.5d$ away from the cylinder base for the natural cylinder does not exists for the controlled cases of $0.4 \leq \beta \leq 0.6$, instead there are two independent peaks generated on the upper and lower layer of the cylinder. This is due to the prevention of vortex interaction in downstream direction; however the TKE values are generated throughout the whole flow field is significantly reduced compared with the natural cylinder. Increasing the porosity value to $\beta=0.7$ results in reemergence of the peak along the centerline of the cylinder which comes on to the meaning of increased fluctuations in the cylinder wake, hence the occurrence of vortex shedding starts.

The fluctuations in streamwise directions are presented by streamwise Reynolds normal stress in second column of Figure 4.55. A double peak occurs in close region to the cylinder base for the natural cylinder which is disappeared when the permeable plate is used having porosity values of $\beta=0.4$ and 0.5 . It is clearly seen that the streamwise fluctuations are dramatically reduced by use of the permeable plate and the peak generation is shifted to farther regions, especially for $0.4 \leq \beta \leq 0.7$. Further increase in porosity to $\beta=0.8$ results in regeneration of double peak in the wake, however the magnitudes are still lower compared with the natural cylinder.

The third column of Figure 4.55 presents the spanwise fluctuations in terms of Reynolds normal stresses where a single peak is generated in the wake region for the natural cylinder case. The effect of permeable plate gets its higher intense on reducing the fluctuations for spanwise direction, i.e. just two contours are evaluated for $\beta=0.4$ and 0.5 which means the fluid flows without any fluctuations in spanwise directions and finally the vortex formation and consequent interaction are effectively

prevented. For $\beta=0.6$, the plate is still very effective on reducing the spanwise fluctuations, however a single peak exists on the last streamwise location of the current flow field which is an indication of shifting the spanwise fluctuations to farther locations. Increasing the porosity decreases this shifting by increasing the intensity of spanwise fluctuations which is also an indication of reduction in the effectiveness of permeable plates on flow control. The higher porosity value, $\beta=0.8$ shows a similar contour distribution as evaluated for the natural cylinder; however the magnitudes are still lower in comparison.

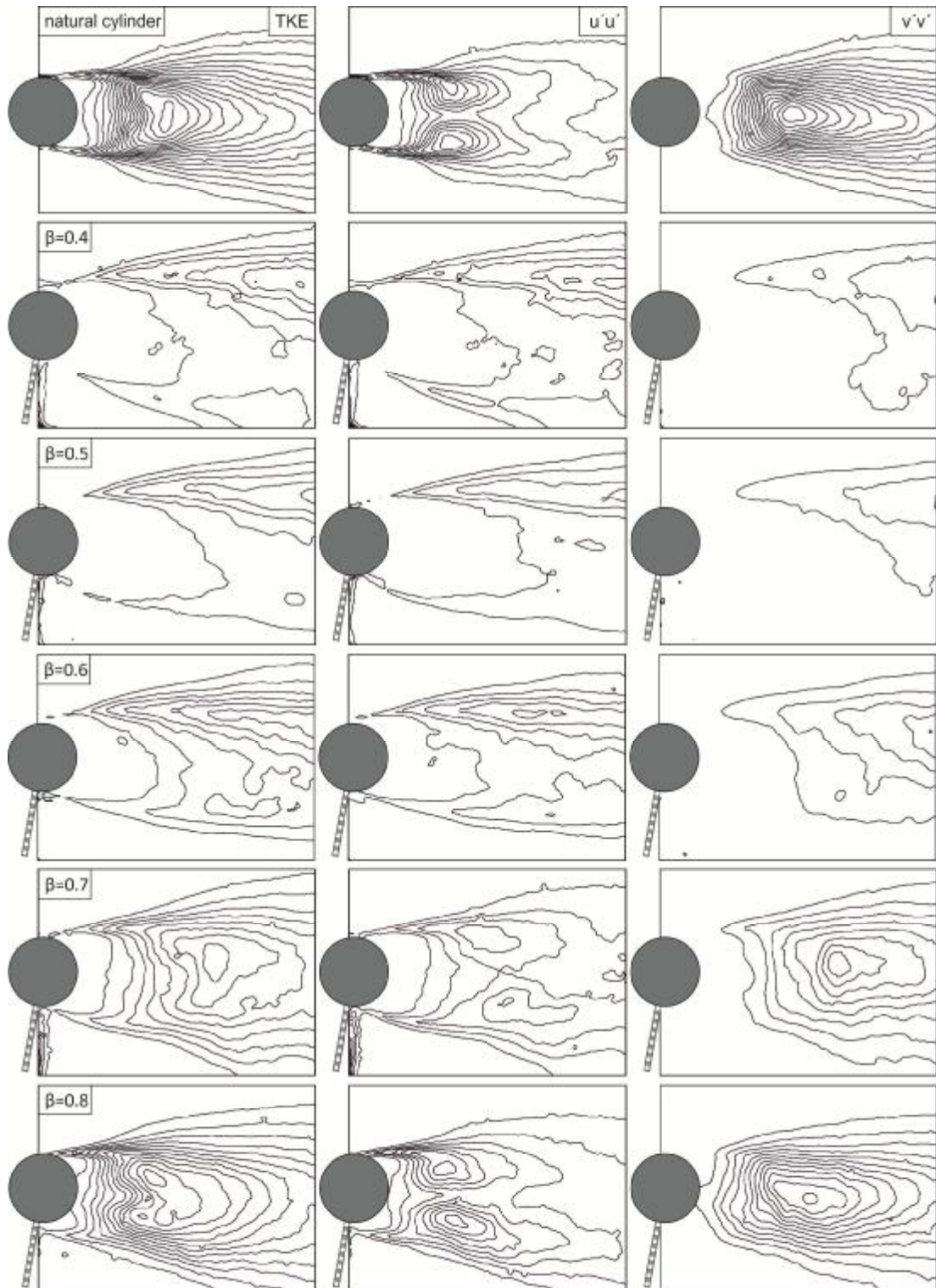


Figure 4.55. The spatial distribution of turbulent kinetic energy (TKE), streamwise Reynolds normal stress $\langle u'u' \rangle$, spanwise Reynolds normal stress $\langle v'v' \rangle$ for all cases, as well as the natural cylinder

5. CONCLUSIONS

The phenomenon of vortex shedding and vortex induced vibrations (VIV) around bluff bodies is of practical interest of many fields in engineering since its output has several undesirable effects such as structural vibrations, increase in drag and lift forces on the body. If the frequency of vortex shedding approaches to the natural frequency of bluff body, the case of resonance takes place and consequently sudden structural failures may occur. Although being an advantage for heat exchanger tubes, this is a serious problem in many engineering applications such as bridges, high-rise buildings, chimneys, turbine blades, cooling systems for nuclear power plants, power transmission lines etc. In recent years investigations have been focused on applications in offshore engineering since oil risers (steel pipes) are being broken like a wire by large amplitude VIV's caused by the deep ocean currents. Ocean engineers are considering how to solve this problem because riser systems cost one-third of an entire offshore production unit. Therefore, the effective control of vortex shedding and induced vibrations is essential in these types of engineering applications.

In this study, a passive control method was applied in order to control the flow around a circular cylinder using attached permeable plates constructed by woven screens. The parameters of the study were the porosity, β of the permeable plate, the plate angle, θ and the plate length, L/d . In order to compare with the permeable plate, effect of a solid plate ($\beta=0.0$) was also investigated. The diameter of the cylinder was kept constant as $d=50\text{mm}$ where the free stream velocity was also kept constant as $U=95.2\text{ mm/s}$ which corresponds to a Reynolds number of $Re_d=5000$ according to the cylinder diameter. Experiments were performed using flow visualization and particle image velocimetry (PIV) techniques. The results showed that the use of an attached permeable plate significantly affects the flow structure downstream of the cylinder.

The existence of the permeable plate has two impacts on the flow structure; one is to directly prevent a vortex roll-in from lower side of the cylinder, and secondly injecting high amount of momentum through its meshes which directly

prevents the roll-in of the upper shear layer. Consequently the lower vortex is not formed and upper vortex is not rolled into the wake region and finally the vortex shedding is successfully prevented. According to this physical reality, the fluctuations significantly decrease in the wake of the cylinder for porosity ratios of $0.4 \leq \beta \leq 0.6$ and permeable plate angles of $45^\circ \leq \theta \leq 135^\circ$ which might be useful for elimination of the vortex shedding and consequent Karman Vortex Street, as well as vortex induced vibrations. The maximum reduction in the maximum value of Turbulent Kinetic Energy in the wake region is obtained as 74% (compared with the natural cylinder) for porosity ratio of $\beta=0.4$ and the plate angle of $\theta=45^\circ$ as being an indication of flow control and suppression of the vortex shedding. Furthermore, the effective intervals of both porosity and plate angle are also effective on reduction in Reynolds Shear Stress values evaluated in the wake of the cylinder.

Further analysis showed that the solid plate is not effective on reducing the fluctuations except for some special cases; $\theta=0^\circ$, 15° and 165° , i.e. these cases reduce the concentrations of RSS contours compared with the natural cylinder case, therefore it can be concluded that these plate angles are effective on suppressing the vortex shedding in the near wake of the cylinder. Using a permeable plate was found to be more effective on flow control and the most effective porosity values were determined to be $\beta=0.4$ and 0.5 for a wide range of plate angle; $35^\circ \leq \theta \leq 150^\circ$, however this effective interval gets narrower for higher porosity values, for instance it is $45^\circ \leq \theta \leq 135^\circ$ for $\beta=0.6$; $45^\circ \leq \theta \leq 135^\circ$ for $\beta=0.7$ and $60^\circ \leq \theta \leq 100^\circ$ for $\beta=0.8$.

The effect of plate angle on flow control was also studied concerning the plate lengths of $L/d=0.4$ and $L/d=1.0$, and the results showed that the permeable plate having a length of $L/d=0.4$ is also effective on controlling the unsteady vortical structure in the wake of a cylinder. However, its effectiveness is valid for a limited range of plate angles; $60^\circ \leq \theta \leq 75^\circ$ due to the shorter plate length interacts with the lower shear layer at larger plate angles. Since the effective interval for $L/d=1.0$ was found to be $45^\circ \leq \theta \leq 135^\circ$, this plate length was preferred to analyze in details. However, smaller plate lengths could be suggested to be used for large plate angles.

In order to have better understanding of the vortex shedding suppression mechanism, the results were also analyzed using instantaneous data of PIV and flow

visualization pictures for effective intervals of the parameters. It was concluded that the suppression mechanism successfully works for the porosities of $0.4 \leq \beta \leq 0.6$. This range was found to be effective on suppressing the vortex shedding (for effective intervals mentioned on previous sections) and elimination of consequent Karman Vortex Street by directly preventing the formation of lower shear layer on the bottom shoulder of the cylinder. However, formation of the high-frequency Kelvin-Helmholtz vortices was observed on the upper shear layer of the cylinder for the controlled cases ($\theta \geq 45^\circ$) of $0.4 \leq \beta \leq 0.6$. Furthermore the porosity value of $\beta = 0.6$ was defined as the transition porosity since the passing fluid through the wire meshes starts to have enough momentum to generate the lower shear layer but still not enough to roll-up into the wake and interact with the upper shear layer.

Finally, the FFT analysis showed that the vortex shedding frequency is annihilated or even completely eliminated for the mentioned effective intervals. The use of a solid plate (splitter plate) was found to be accurate on annihilating the shedding frequency reaching to a reduction of 40% compared with the natural cylinder. Furthermore, the use of a solid plate with $\theta = 15^\circ$ completely eliminates the existence of a frequency in the wake of the cylinder.

The effective intervals of the permeable plates having the porosity values of $\beta = 0.4$ and 0.5 were also found to be accurate on reducing the vibrations, i.e. the frequency obtained in the cylinder wake gradually decreases by increasing plate angle, and for $\theta \geq 45^\circ$ the existence of a dominant frequency downstream of the cylinder is completely eliminated within the limit of effective plate angle. The porosity values of $\beta = 0.6$ and 0.7 were also found to be useful on annihilating the vortex shedding frequency; however a complete deterioration of frequency could not be obtained for these porosity values.

According to these findings, the use of a permeable plate on effective intervals could be suggested for the engineering applications concerning the vortex shedding and induced vibrations. Although this study focuses on suppression of vortex induced vibrations, the drag force exerted on the cylinder with attached permeable plate should be studied in order to evaluate the effect of permeable plates on drag force. It is assumed that use of a higher porosity plate would also results in a

lower drag exerted on the cylinder. Therefore the use of a permeable plate with $\beta=0.6$ porosity is suggested to have both lower vibrations, as well as the lower drag force. If one deals with only eliminating the vortex induced vibrations, the porosity values of $\beta=0.4$ and 0.5 could be suggested with plate angles of $45^\circ \leq \theta \leq 135^\circ$. Furthermore, and as being a future work, use of double-attached permeable plates may be used in order to complete elimination of vortex shedding, induced vibrations and the Karman Vortex Street.

This new accurate passive method would be an alternative technique on suppression of VIV's in deep water applications and future works about this topic will be studied concerning the drag force measurements.

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