EFFECT OF CYLINDER SHAPE ON HEAT TRANSFER AND FLUID FLOW

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Abstract

Two dimensional flow over single cylinder with different shapes in the free stream are numerically investigated. Consideration is given to the effect of cylinder seven shapes with the same area, and Reynolds number (Re) on heat transfer and flow phenomena. The study has been carried out for Reynolds number between (200-2000). The study discloses that the heat transfer and pressure drop are effected by cylinder shape. The relationship between Nuselt number (Nu) and angle from stagnation point (θ) also pressure coefficient (Cp) and angle from stagnation point (θ) are determined, the new shapes are compared with circular cylinder and the results show that maximum heat transfer be with the shape (7).

Keyword : cylinder shape, enhancement heat transfer, numerical analysis

تأثير شكل الاسطوانة على انتقال الحرارة وجريان المائع

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الخلاصة

جريان ثنائي البعد حول اسطوانة مفردة بأشكال مختلفة في جريان حر تم فحصه عدديا. تم الاخذ بنظر الاعتبار تاثير سبعة اشكال بنفس مساحة المقطع وعدد رينولز (Re) على انتقال الحرارة وشكل الجريان. تمت الدراسة ضمن مدى لرقم رينولز (200-2000). بينت الدراسة ان انتقال الحرارة والهبوط بالضغط يتاثر بشكل الاسطوانة. تم تحديد العلاقة بين رقم نسلت (Nu) مع الزاوية حول السطوانة المقاسة من نقطة الركود (θ) كذلك العلاقة بين معامل الضغط (Cp) مع الزاوية حول السطوانة المقاسة من نقطة الركود (θ). الاشكال الجديدة للاسطوانة تم مقارنتها مع الشكل الاسطواني واوضحت النتائج ان اعلى انتقال للحرارة يتم بواسطة الشكل (نوع 7).

Nomenclature

A area of cylinder (m^2)

b the width of wake (m)

Cp pressure coefficient

FVM finite volume method

H high of duct (m)

h_θ local heat transfer coefficient

k thermal conductivity (W/(m. K)

L length of wake from the center of cylinder (m)

Nu_θ local Nusselt number

P_o stagnation pressure (N/m^2)

P_θ local pressure (N/m^2)

Re_H Reynolds number

S_Φ source term

T temperature (k)

u velocity in x-direction (m/s)

v velocity in y-direction (m/s)

ρ density (kg/m^3)

μ viscosity (Ns/m)

∝ thermal diffusivity, (W/(m K))

ß angle of corner

θ angle from stagnation point

Introduction

Circular cylinders are commonly used in heat exchanger and the air (in gas cross flow heat exchanger) cross the cylinder [Tiwari 2003]. The compact heat exchanger find application in automobiles, air conditioning, electronic cooling and aircraft. These devices usually operate at moderate Reynolds number. Laminar or nearly laminar flow of air is associated with low heat transfer coefficient and therefore, the efficiency is improved by enhancing the boundary layer conditions- for example with turbulence promoters and by additional enlargement of the heat transferring surface (via fins). Fins can act as turbulence promoters also [Keys 1984]. Heat transfer from cylinder has been investigated experimentally [Ertan 1999] and [Achembach 1975] and numerically [Karniadtis 1988] and [Kurdyuman 1998]. Heat transfer from circular cylinder with using wing and winglet have been investigated [Tiwari 2003], [Sohal 2001] and [Jalal 2006] to control the flow over cylinder and enhanced heat transfer. Many studies deal with cylinder with rectangular shape [Valencia 1995], oval tube [Sohal 2001], single diamond-shaped cylinder [Shuichi 2008] and square cylinder with cutoffs corners [Yoichi 2009]. [Peter 1998] studied the effect of novel tubes on flow characteristic.The aim of this study is to investigate the influence of seven shapes of cylinder **(Figure (1))** on heat transfer and fluid flow placed inside channel.

Mathematical formulation

The conservation equations describing the flow are two-dimension Navier-Stokes equations and energy equation for constant property incompressible fluid [Ferizger 1999].

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

x- momentum:-

$$\left.\begin{aligned} &u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{\mu}{\rho}\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right] \\ &\text{y- momentum:-} \\ &u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\frac{\mu}{\rho}\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right]\end{aligned}\right\} \tag{2}$$

Energy equation

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\frac{1}{\alpha}\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right] \tag{3}$$

The physical configuration, the coordinate system of the flow and the grid of domain for type (4) are shown in **Figure (2).** The figure shows the schematic of two dimension body fitting grid used for present computation, this grid is a body-fitting type. The grid obtained by solved partial differential equation and with clustering near the walls.

Statement of the problem

Rectangular cross section channel with different shapes of cylinder have the same area. Because of the complex geometry equations (mass, momentum and energy) in the general coordinates (ξ, η) were used, an incompressible finite volume code, with collocated grid arrangement was used. The general differential form of the transport equation can be written for dependent variable (Φ) [Ferizger 1999].

$$\frac{\partial(\rho u\Phi)}{\partial x}+\frac{\partial(\rho v\Phi)}{\partial y}=\frac{\partial}{\partial x}\left[\Gamma_\Phi\left(\frac{\partial\Phi}{\partial x}\right)\right]+\frac{\partial}{\partial y}\left[\Gamma_\Phi\left(\frac{\partial\Phi}{\partial y}\right)\right]+S_\Phi \tag{4}$$

The two terms in the left of the general transport equation are convective terms, the first two terms in the right hand of equation (4) are the diffusive term and the last is the source term. Equation (4) serves as starting point for the computational procedure in (FVM). By setting the depending variables (Φ) and (Γ_Φ)according to the table (1) and the term (S_Φ) in equation (4) is the source term of quantity (Φ), When (Φ) stands for velocity components, (S_Φ) contains the appropriate pressure gradient term as well as arising from coordinate curvature. When (Φ) stands for temperature, (S_Φ) contains the term arising from coordinate curvature. The (mass, momentum and energy)equations can obtain respectively. By applied partial differential chain rule to equation (4) and by applied **Table (1)** we can obtain the (mass, momentum and energy) equations in general coordinate with conterevariant velocity components so this is the procedure for transport this equation to general coordinates [Ferizger 1999].

Boundary Conditions

The boundaries of the solution domain (**Figure 2**) include symmetry plan, solid wall (cylinder and duct), inlet and outlet plans. On symmetry plan the velocity normal to the symmetry plan is zero as normal gradient of the temperature and all other variables. For the temperature, the boundary condition is as follows: in the case of cylinder, heat flux is prescribed directly for using to the solution procedure, in the case of wall duct, adiabatic condition is prescribed in the solution procedure. In the case of inlet all quantities have to be prescribed and in the case of outlet, the first derivative of all properties variables in the direction normal to such a boundary equal to the zero.

- **Heat transfer calculation**

local Nusselt number (Nu_θ) can be calculated from the following equation:-

$$Nu_\theta = h_\theta * (H/k_f) \tag{5}$$

- **Local static pressure coefficient**

The local static pressure coefficient (Cp_θ) along the wall surface of cylinders can be calculated as following:-

$$Cp_\theta = \frac{P_\theta - P_\infty}{0.5 \rho u_{in}^2} \tag{6}$$

Results and Discussion

Figure (3) shows the streamline around cylinders at Re=1500. It is clear that the wake area is the different between the types of cylinder and the figures show that minimum area behind the cylinder type 6 and maximum wake area behind the cylinder type 3 we also shows that the vortex extended for long distance behind the cylinder type 2 and there is a small vortex in front of cylinders Types 3 and 4 due to large area in front of flow. Also there is a small vortex on the top side of cylinder type 6 due to long horizontal wall and the flow moves toward the main flow then returns toward the cylinder.

Figure (4) shows the temperature contours around all cylinder shapes. In the case of types 1, 2, 5 and 7, it can be noticed that the region in front of cylinders are cooled because the main flow sweep the hot fluid near the cylinder, the regions behind the cylinder have the highest temperatures because of the separation point between (θ= 70 and 100).

Figure (5) shows the pressure distribution along the cylinder for all shapes at Re=1500. The ordinate shows the pressure coefficient (Cp), and the abscissa shows the angular position (angle of the stagnation point θ). The figure shows the difference in pressure coefficient for different cylinder shapes, at the right figure (θ=0 ~ 70) the pressure is high at stagnation point (θ=0) and become small gradually of the type 5 (circular cylinder), the pressure is high for the face of cylinder type 3 (θ=0 ~ 70), for all shapes of cylinder except type 5(circular cylinder) because the shape of circular cylinder have a curvature but another shapes have an angle ß has different values dependent on the cylinder shape. At the left figure (θ=70 ~ 180) the pressure is low and increases slowly for type 5 and sudden increase for other types because of the separation of flow and the vortex behind the cylinders

Figure (6) shows the heat transfer distribution along cylinder surface for all cylinder types at Re=1500. Local heat transfer along cylinder surfaces for all shapes are different, maximum heat transfer at the front area of cylinder for all shapes and decreases with increase thermal boundary layer thickness. Minimum heat transfer at the rear of cylinder shows at types (1, 2 and 3) because of large wake area (hot area), types (4 and 7) have high heat transfer from the rear area because of thin vortex area behind the cylinders.

Figure (7) shows the variation of non dimensional length (L/H) of wake of cylinders with Reynolds number (Re), type 2 has longer wake because the flow needs long distance to return to the main flow and the distance between the vortex is long. Also we show that the length of wake behind cylinder type 6 is the smallest when compared with the another types because the cylinder has long side distance and the flow returns toward the cylinder.

Figure (8) shows the variation of non dimensional width (b/H) of wake of cylinder with Reynolds number. It is clear that the width became small with increasing Reynolds number because of high momentum of main flow. Cylinder type 6 has small width and cylinder type 3 has large width due to large area of cylinder in front of flow.

Conclusions

The results of numerical analysis around cylinder with different shapes lead to following conclusion:-

1- The shape of cylinder effects on the heat transfer and pressure drop.
2- The pressure across cylinder type 5 (circular cylinder) changes gradually but in other types has a sudden change due to side corner.
3- Pressure drop increases in the front of area (θ=0 ~90) in type 5 by (42 %- 86 %) but it increases behind area (θ=90 ~180) also for type 5 by (14 % - 53 %).
4- The average pressure drop for type 5 with respect to the other shapes was (32% - 63 %).
5- Minimum pressure drop be for cylinder type 2.
6- Heat transfer increases in the front of area (θ=0 ~90) in type 5 by (0.4 %- 1.6 %) but it increases behind area (θ=90 ~180) for type 7 by (1.3 % - 9.5 %).
7- The average heat transfer enhancing for type 7 with respect to the other shapes by (0.34 % - 4.7 %).
8- The length of wake in type 2 is larger than the other types of cylinder but cylinder type 6 is the smallest.
9- The width of wake in type 3 is larger than the other types of cylinder and type 6 was the smallest width.

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Table (1) variables, diffusion coefficients and source terms

Equation	Φ	Γ_Φ
Continuity	1	0
x-equation	U	μ
y-equation	V	μ
energy	T	μ/p_r

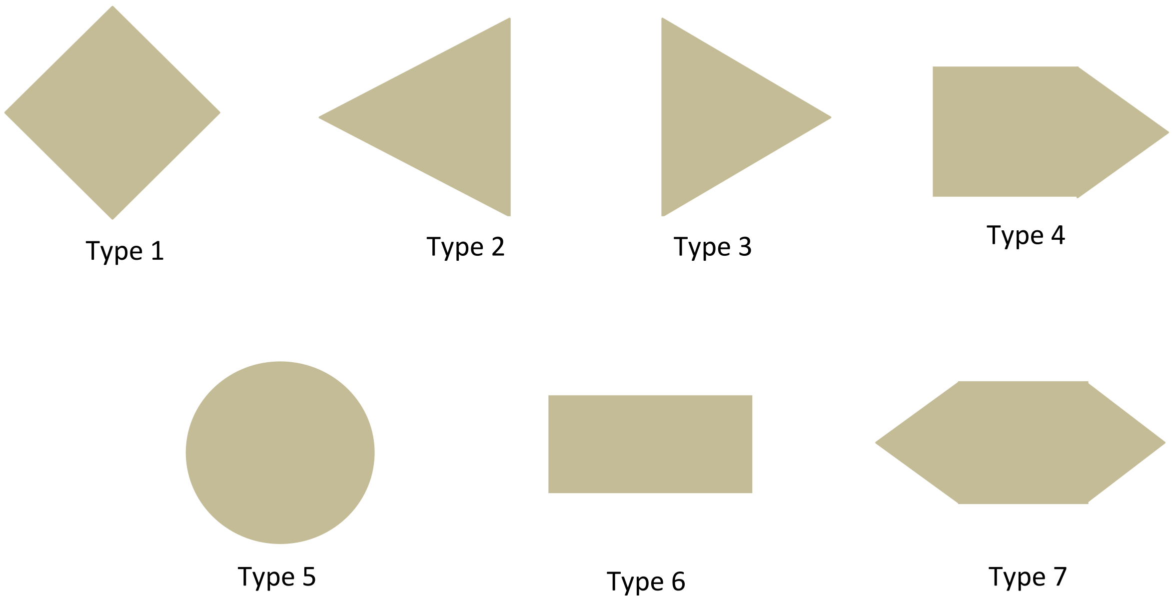


Figure (1) Types of cylinder shapes

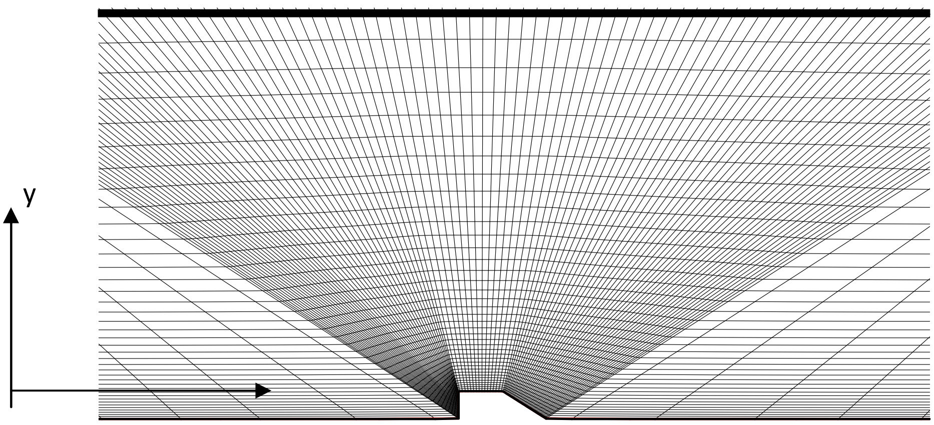


Figure (2):- Grid generation and computational domain for cylinder (type 4)

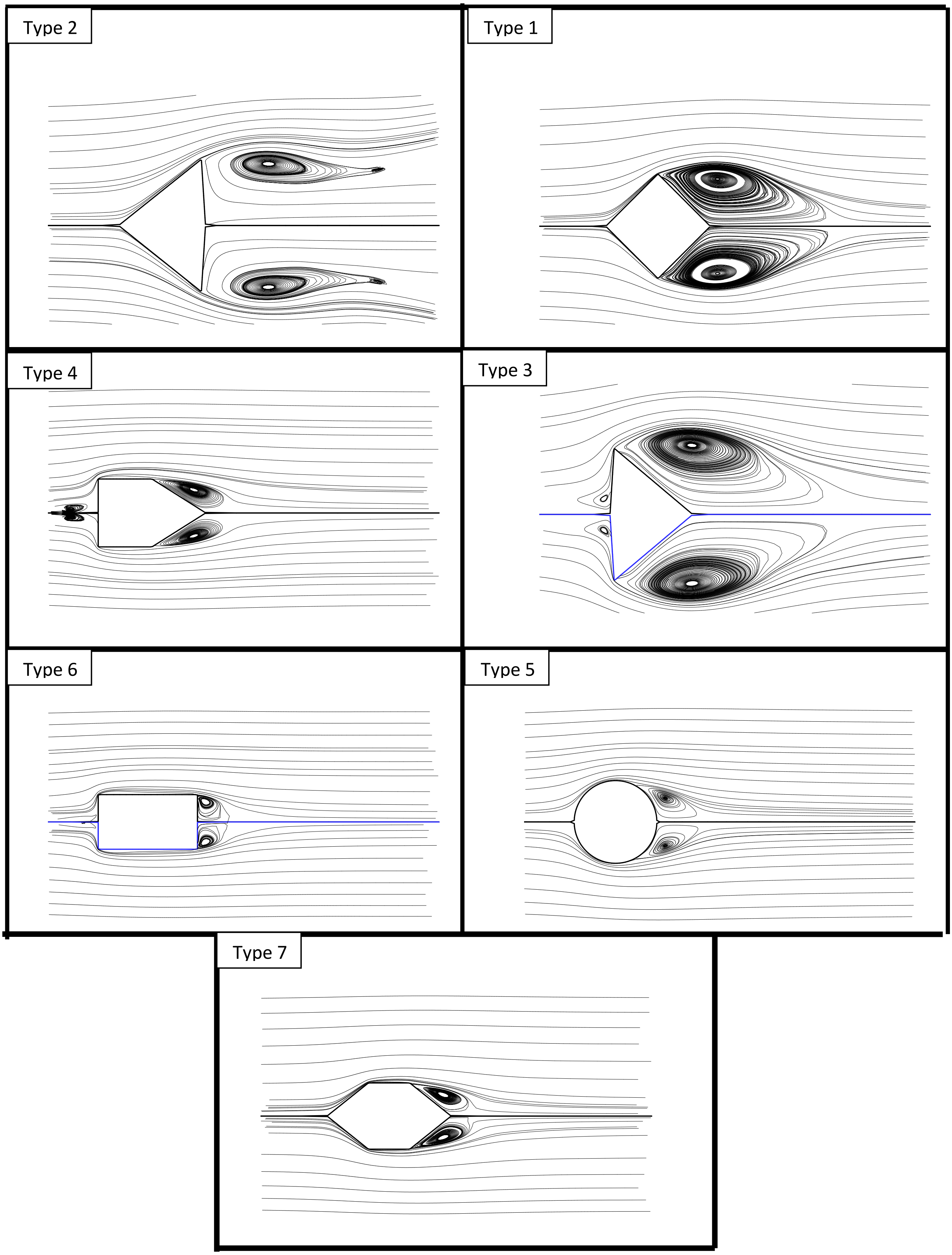
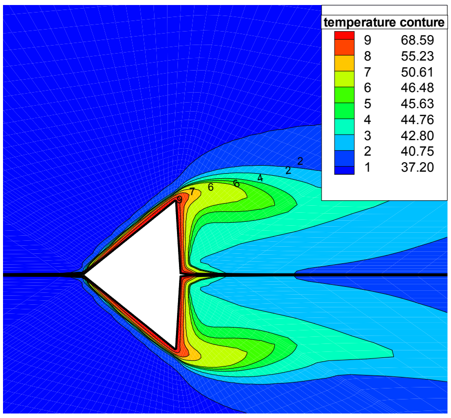
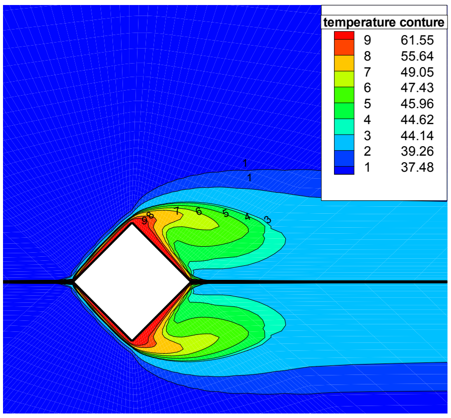
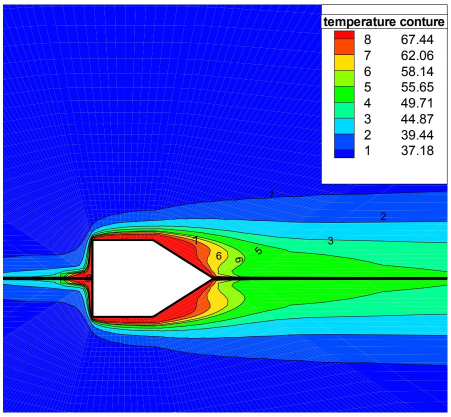
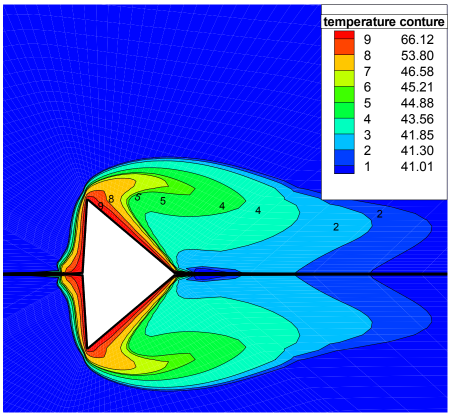


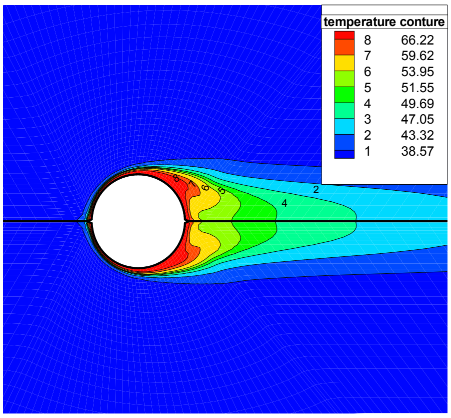
Figure (3) the computational result of the streamline around the cylinders

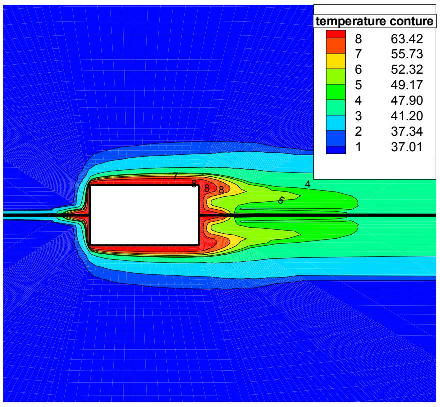












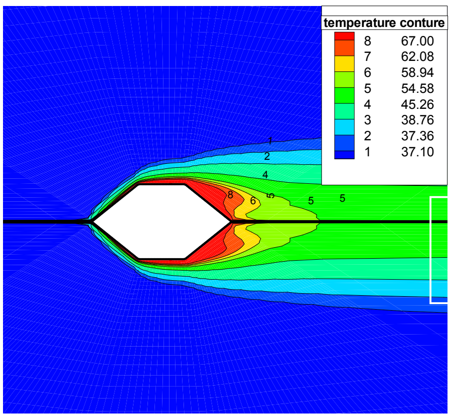


Figure (4) the computational results of temperature contours around cylinders at

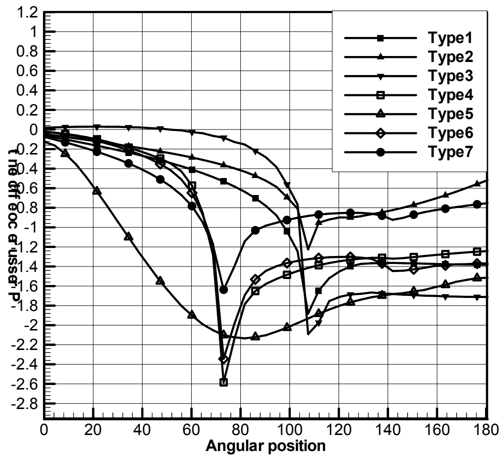


Figure (5) pressure coefficient distribution around cylinders

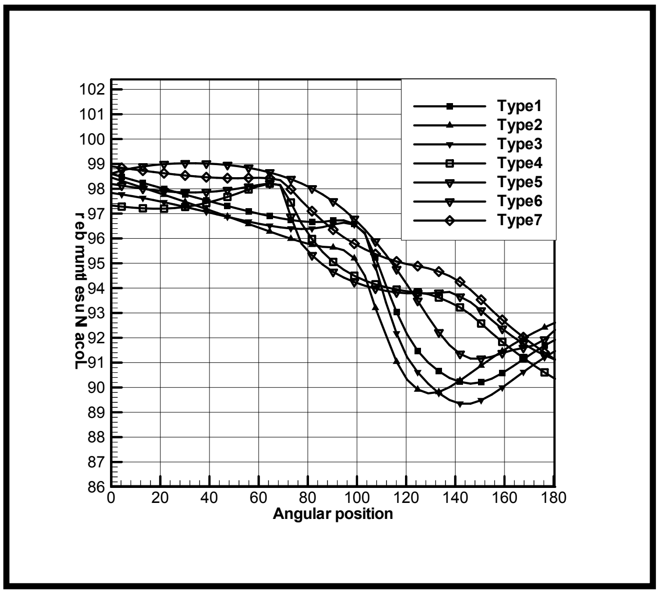


Figure (6) Nusselt number distribution around cylinders

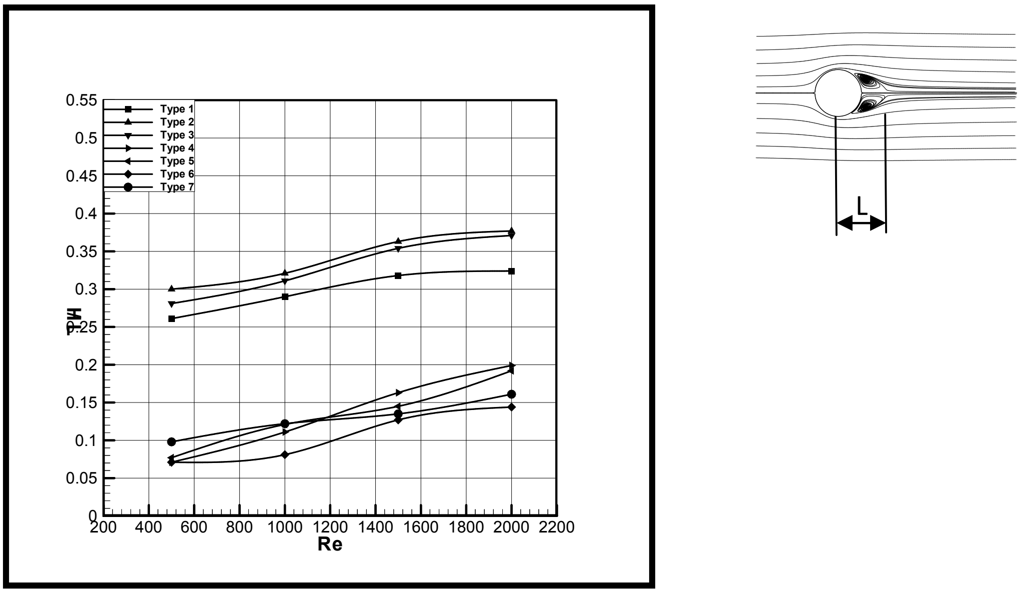


Figure (7) variation of non dimensional length (L/H) of wake of cylinders with Reynolds number

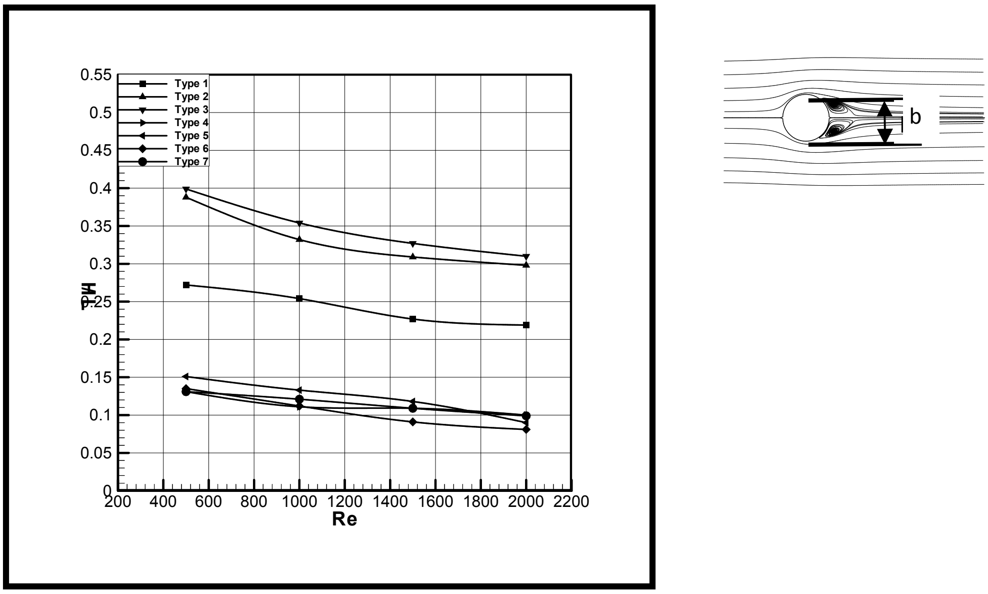


Figure (8) variation of non dimensional width (b/H) of wake of cylinders with Reynolds number