NUMERICAL INVESTIGATION OF FLUID FLOW AND HEAT TRANSFER CHARACTERISTIC AROUND HEATED CYLINDER INSIDE CONVERGEN- DIVERGENT CHANNEL +

دراسة عددية لخصائص الجريان وانتقال الحرارة حول اسطوانة موضوعة داخل مجرى متضيق-متوسع

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Abstract:

Two-dimensions flow over single circular cylinder in inside convergent-divergent channel are investigated. Considerations is given to the effect of Reynolds number, (Re), peak of convergent-divergent channel, (a), and the length of base of peak, (b), on heat transfer and flow phenomena. A finite volume procedure on non-orthogonal collocated grid on SIMPLE algorithm was used to solve the conservation equations (mass, momentum and energy) with constant heat flux under laminar fully developed flow conditions.

Comparisons with the cylinder in straight channel, in the same flow rate and heat transfer conditions, have been performed. Heat transfer from cylinder in convergent-divergent channel is found to be always larger than the value characteristic of a cylinder in straight channel, also pressure drop performance increases. The results show that the maximum of heat transfer be at (a/H= 0.53, b/H= 0.46).

Key wards: numerical solution, heat transfer, circular cylinder, Convergent- divergent channel

المستخلص:

تم دراسة جريان ثنائي الاتجاه على اسطوانة دائرية موضوعة داخل مجرى متضيق – متوسع. تم دراسة تأثير رقم رينولز(Re)، قمة التضيق في المجرى(a) وطول قاعدة القمة(b) على خواص الجريان وانتقال الحرارة من الاسطوانة. تم استخدام طريقة الحجوم المحددة (FVM) على الشبكة غير المتعامدة وباستخدام الطريقة البسيطة (SIMPLE algorithm) لحل المعادلات الحاكمة (الكتلة، العزم والطاقة) تحت ظروف الجريان الطباقي كامل النمو. تمت مقارنة النتائج مع الاسطوانة الموضوعة داخل مجرى مستقيم بنفس الظروف من جريان ودرجة حرارة.

أوضحت النتائج ان انتقال الحرارة من الاسطوانة الموضوعة داخل المجرى المتضيق المتوسع دائما اعلى من حالة الاسطوانة الموضوعة داخل المجرى المستقيم كذلك الهبوط في الضغط فانه يزداد، وان أفضل انتقال حرارة يكون عند (a/H= 0.5053, b/H=0.46).

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Nomenclature

Symbol	Description	Unit
A	Convection-diffusion discritization coefficient	
a_1, a_2 a	Transformation coefficients peak	 m
b	Length of base of peak	m
Cp	Static pressure coefficient	
D	Diffusion term coefficient	
F	Mass flux	
H	Channel height	m
h_{ij}	Local heat transfer coefficient	W/(m^2.K)
J	Jacobean of transformation	
k	Thermal conductivity	W/(m. K)
n	Normal distance	
Nu	Nusselt number	
P	Source term in grid generation system of equation	
P_0	Stagnation pressure	N/m^2
P_{ij}	Local static pressure	N/m^2
Pr	Prandtl number	
q	Heat flux per unit area	W/m^2
Q	Source term in grid generation system of equation	
S	Source term for the dependent variable Φ	
T	Temperature	K
u	Velocity in x-direction	m/s
U, V	Contravariant velocities	
v	Velocity in y- direction	m/s
x, y	Cartesian coordinates	

Greek Symbols

Symbol	Description	Unit
ξ, η	General coordinates	
Φ	General variable	
ρ	Density	Kg/m^3
Γ	Coefficient of Diffusivity	

Abbreviation

Symbol	Description	Unit
2-D	Two dimensions	
FVM	Finite volume method	

Subscripts

Symbol	Description	Unit
b	Bulk	
e, w, n, s	Faces of control volume	
E, W, N, S	Neighbor nodes of node P	
f	fluid	
P	Center of control volume	
ij	Index counter in (x, y) or (ξ, η) directions	
in	Inlet condition	
Φ	Total source of Φ transport property	
w	Wall	

Introduction:

For a long time, efforts have been made to produce more efficient of heat exchangers by employing various methods of heat exchanger augmentation. A wealth various aspects of this flow configuration [1]. The optimum design of heat exchanger can minimize the size of wake (stagnation flow) behind a cylinder tube and also improve the heat transfer [2]. The heat transfer from cylinder can be improved for example by means of boundary layer modification by using wings and winglets.

Flow around circular cylinder and boundary layer separation have been investigated experimentally [2, 3] and numerically [4, 5]. Also, the enhancement of heat transfer by using different shapes of winglets have been investigated experimentally [6] and numerically [7]. Vortex generators leads to enhancing heat transfer because of distribution of the growth of thermal boundary layer and controlling the flow over the cylinder surface with increasing pressure drop. The optimization of heat exchangers always has to be aimed at an increase of heat transfer simultaneously with minimum increase of pressure drop. The subject of this study is to place circular cylinder inside a channel has two convergent-divergent positions, the first position in front of cylinder and the second behind it. This type of channel enhanced heat transfer from cylinder by controlling the flow toward the cylinder and sweeping the heat from it. The channel with converging-diverging cross section is one of several device employed for enhancing the heat and mass transfer efficiency due to turbulent promotion [8]. The effect of channel wall on heat transfer and pressure drop have been investigated numerically with four different waves peak (a/H) (0.013 to 0.053) and five different length of base of peak (b/H) (0.33 to 0.6) for Reynolds number range (1000 - 10000) The geometry considered in this study is shown in figure (1).

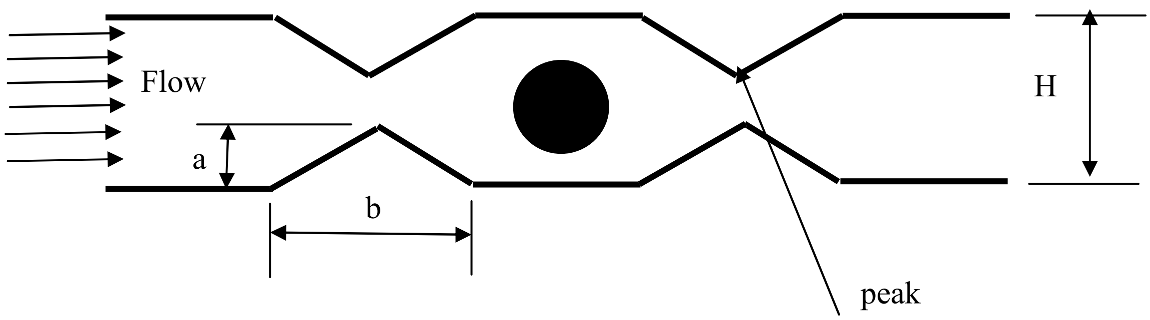


Figure (1) Schematic for the flow past cylinder inside convergent-divergent channel

Problem formulation

1-Grid generation

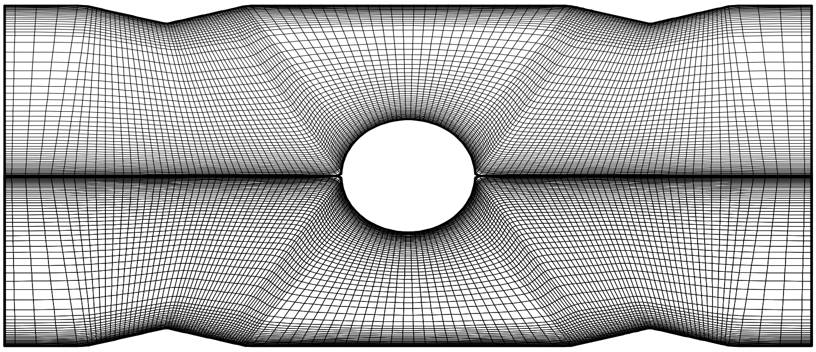
The grid of present problem have been generated by solving non-homogenous 2-D Laplase equation [9].

$\xi_{xx} + \xi_{yy} = P(\xi, \eta)$ --(1)

$\eta_{xx} + \eta_{yy} = Q(\xi, \eta)$ --(2)

P and Q are used to clustering the grid near the walls to sense the velocity gradient..

Figure (2) shows the grid for present solution



Fiqure (2) Grid generation for present case

1- Governing equations

Consider the steady- two dimension flow (2-D) of viscose incompressible Newtonian fluid past circular cylinder inside convergent-divergent channel as shown in figure (1). The cylinder is located at the center of channel and at the same distance from wave's peak. The length of the cylinder in Z-direction is assumed to be sufficiently long to have insignificant end effects.

Under these conditions, the equations of continuity, momentum and energy are[10]:-

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \quad \text{---(3)}$$

x- momentum:-

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{\mu}{\rho}\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right]$$

y- momentum:-

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\frac{\mu}{\rho}\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right] \quad \text{-----------------------------------(4)}$$

Energy equation

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\frac{1}{\alpha}\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right] \quad \text{--(5)}$$

Because the convergent-divergent channel geometry is complex, equations (3, 4 and 5) must transform to general curvilinear coordinates system (ξ, η). The general equation in curvilinear coordinates system can be written as follow:-

$$\frac{\partial}{\partial \xi}\left(\rho U \emptyset-\frac{\alpha_1 \Gamma_\emptyset}{J}\frac{\partial \emptyset}{\partial \xi}\right)+\frac{\partial}{\partial \eta}\left(\rho V \emptyset-\frac{\alpha_2\ \Gamma_\emptyset}{J}\frac{\partial \emptyset}{\partial \eta}\right)=JS_\emptyset+S_N \quad \text{-----------------------(6)}$$

Where continuity, momentum and energy equations in general curvilinear coordinates system can be obtained by setting the dependent variables ($\emptyset, \Gamma_\emptyset$ and $S_\emptyset$) according to table (1).

Table 1 : Variables and sources terms

Equation	$\emptyset$	$\Gamma_{\emptyset}$	$S_{\emptyset}$
Continuity	1	0	0
x-momentum	u	μ	$-\frac{\partial p}{\partial x}$
y-momentum	v	μ	$-\frac{\partial p}{\partial y}$
Energy	T	$\frac{\mu}{Pr}$	0

And S_N is the source term due to non orthogonal characteristic of grid system which disappears under orthogonal grid system. The transformation coefficients (a_1, a_2) are defined as:-

$$\left.\begin{aligned} a_1 &= \xi_x^2 + \xi_y^2 \\ a_2 &= \eta_x^2 + \eta_y^2 \end{aligned}\right\} \text{--- (7)}$$

And

$$\left.\begin{aligned} \xi_x &= \frac{y_\xi}{J} \\ \xi_y &= -\frac{x_\eta}{J} \\ \eta_x &= -\frac{y_\xi}{J} \\ \eta_y &= \frac{x_\eta}{J} \end{aligned}\right\} \text{---(-8)}$$

And for non staggered grid arrangement all properties located at the center of control volume.

After integration of equation (6) over control volume the following equations can be obtain:-

$$\left[\rho U \emptyset - \left(\frac{a_1 \Gamma_{\emptyset} \emptyset_\xi}{J}\right) \Delta\eta\right]_w^e + \left[\rho V \emptyset - \left(\frac{a_2 \Gamma_{\emptyset} \emptyset_\eta}{J}\right) \Delta\xi\right]_s^n = dv\left[J\, S_{\emptyset} + S_N\right] \text{ ----(9)}$$

Where (e, w, n and s) are faces of the control volume. To interpolate convection and diffusion terms, the power law difference scheme can be used [10] to obtain algebraic equation.

$$A_p\, \emptyset_p = A_E \emptyset_E + A_W \emptyset_W + A_N \emptyset_N + A_S \emptyset_S + dv(J\, S_{\emptyset} + S_N) \qquad \text{------------(10)}$$

Boundary conditions:

The boundary conditions for present domain include symmetry plane, solid wall, inlet and exit planes.

No-slip boundary condition for velocities on all solid wall are used, at the channel inlet, a normal component of velocity is assumed to be zero, axial velocity and all other properties have to be prescribed. At the exit, the boundary conditions for the dependent variables are obtained by sitting the first derivative in the axial direction equal to zero. Constant heat flux was prescribed for using in the solution procedure in the case of cylinder wall and in the case of channel wall, an adiabatic condition was prescribed in the solution procedure. On symmetry plane, the normal velocity and normal gradient of other dependent variables assume to be zero.

Numerical Solution:

In this study a (79 * 45) grid mesh is used in the numerical computation. The finite volume method (FVM) is used and the solution stares from the conservation equation in general form. The integral forms of governing equations are discretized using control volume based on finite volume method [11]. The solution domain is first subdivided into finite number of contiguous volume center. At centroid of each control volume lies, a computational node at which all variable values are to be calculated by using collocated arrangement. Iteration is continued until difference between two consecutive field values of variables is less or equal to (10^{-4}). Air has been used as working fluid, hence the Prandtl number was (0.71). Nusselt number and pressure drop are calculated:-

- Heat transfer calculation

To calculate heat transfer (Nu) from heated walls at constant heat flux, first the temperature distribution can be calculated from the following equation[7]:-

$$q_{wall} = -k\frac{\partial T}{\partial n} \quad \text{--} \; (11)$$

Where n is the normal distance

Then calculate local Nusselt number (Nu_{ij}) for the following equations:-

$$q_{conv.}=q_{wall} \quad \text{--}(12)$$

$$q_{conv}=h_{ij}*(T_{ij}-T_b) \quad \text{--} \; (13)$$

$$Nu_{ij}=h_{ij}*(H/k_f) \quad \text{--}(14)$$

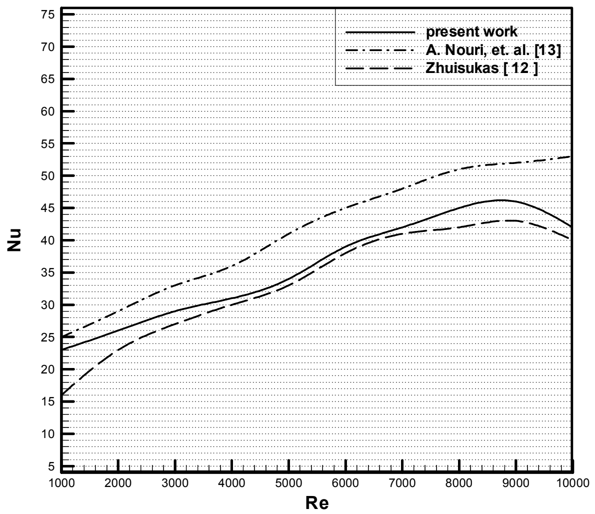
- Local static pressure coefficient

The local static pressure coefficient (Cp_{ij}) along the wall surface of cylinder can be calculated as following[7]:-

$$Cp_{ij} = \frac{P_{ij}-P_o}{0.5\rho u_{in}^2} \quad \text{--}(15)$$

Results and discussion :

Before solving the problem of interest, the code has been validated by using the prior results that are available in the literature to establish the level of accuracy of the results obtained in this work. For flow over cylinder inside channel with the same dimension, the average of heat transfer with Reynolds number and local heat transfer over cylinder show a good accuracy with previous results as in figure (3-a,b).



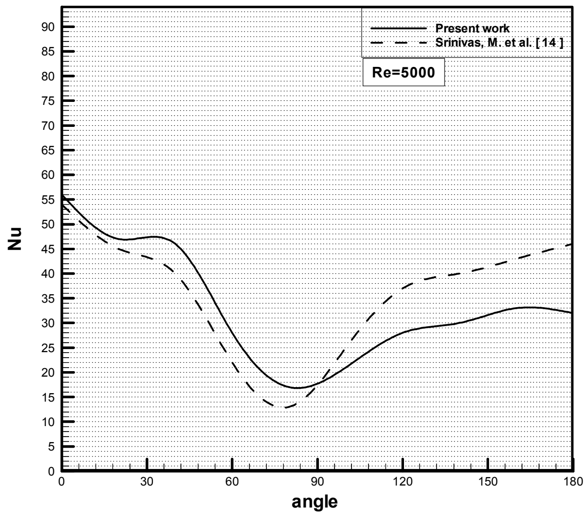


Figure (3-a): Comparison of present average Nusselt number with Churchil [13] and Zhukauskas, A.[12] for flow over cylinder inside straight channel

Figure (3-b): Comparison of present local Nusselt number across cylinder with Srinivas, M., et. Al. [14]

Also figure (4) Show the pressure drop- across cylinder and the result show good agreement with the previous work of [15].

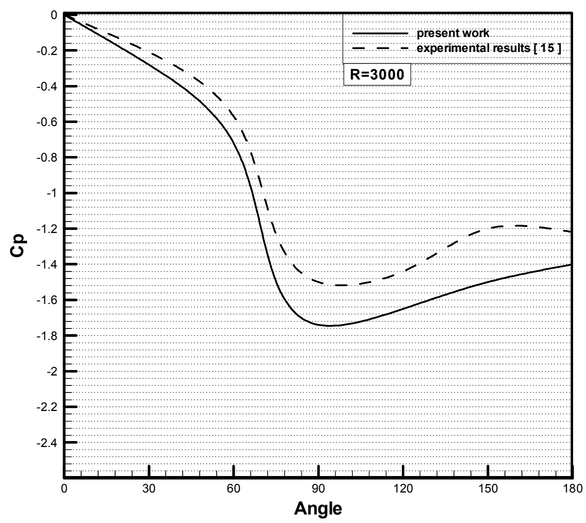


Figure (4): Comparison of pressure drop over cylinder with Norbery, C. [15] inside straight channel

Figure (5) shows the stream line of flow over a cylinder located inside convergent-divergent channel with different (b/H) at (Re= 5000) with (a/H=0.053), also the figure show the stream line over cylinder inside straight channel. The results show that when the cylinder being inside straight channel, the boundary layer separated nearly at (θ=90) and recirculation flow are clearly notes at the rear of cylinder also, the size of the wake (causing poor heat transfer) behind the cylinder is large compared to the region of wakes behind the cylinder inside convergent-divergent channel. The convergent-divergent channel work as a guide to pass more of fluid toward the rear of cylinder and lead to decrease the wake region and the heat transfer will be increasing due to the sweeping of heat from the rear part of cylinder. The figure shows that when the distance (b) is small, a small vortex will be apparent behind the narrow channel due to the recirculation of flow at the rear of peak and disappearing with increasing the length of base (b).

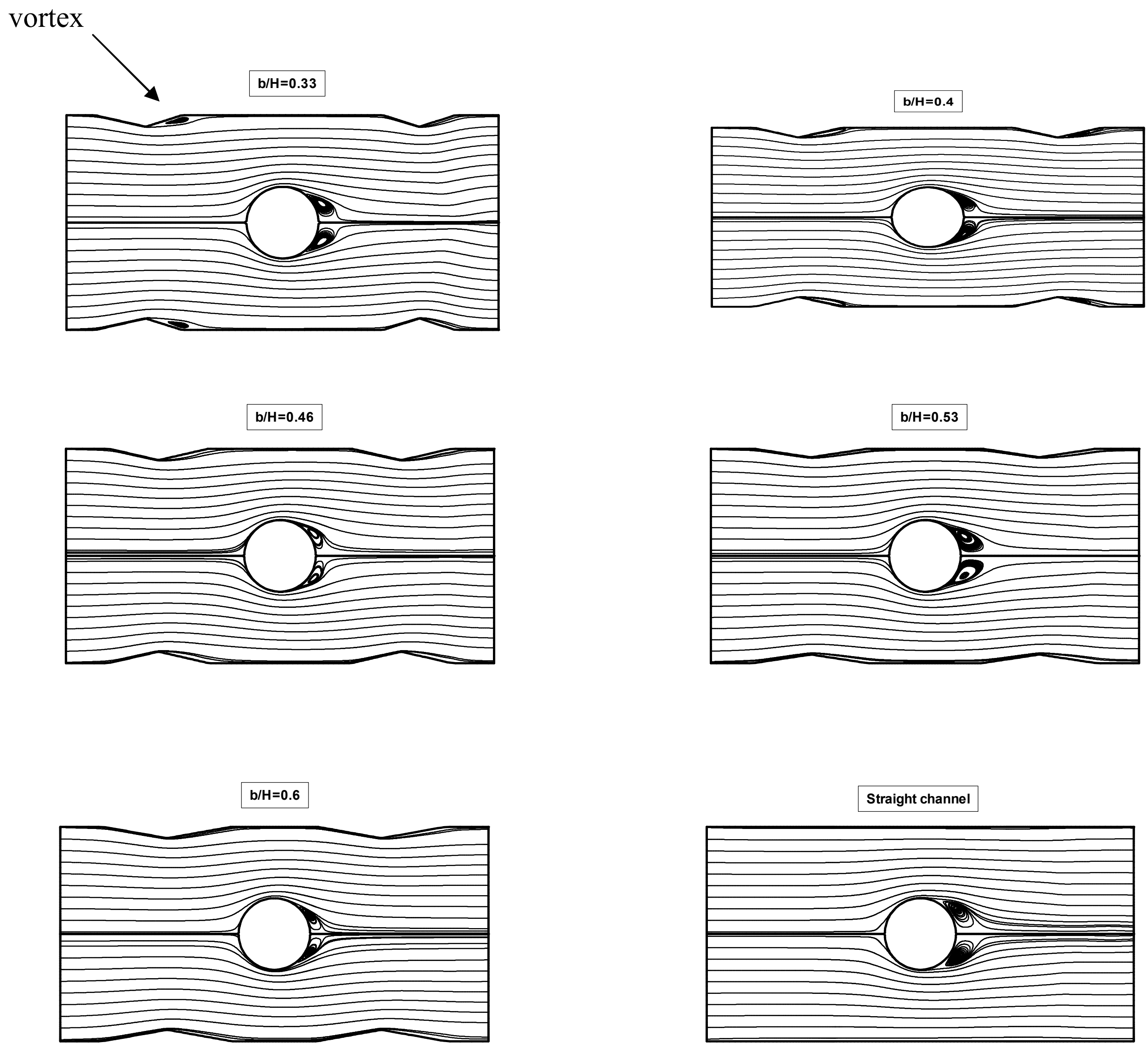


Figure (5): Streamline of flow over cylinder inside convergent-divergent channel at (a/H=0.053) with deferent length of base (b/H)

In figure (6) effects of peak highness is examined at (Re=5000), the wake region behind the cylinder is decreased with the increasing peak highness, this due to pass the fluid flow toward the rear of cylinder and leads to increasing heat transfer from cylinder but recirculation areas will be apparent behind the divergent parts at high value of peak, this lead to increase in pressure drop across the channel.

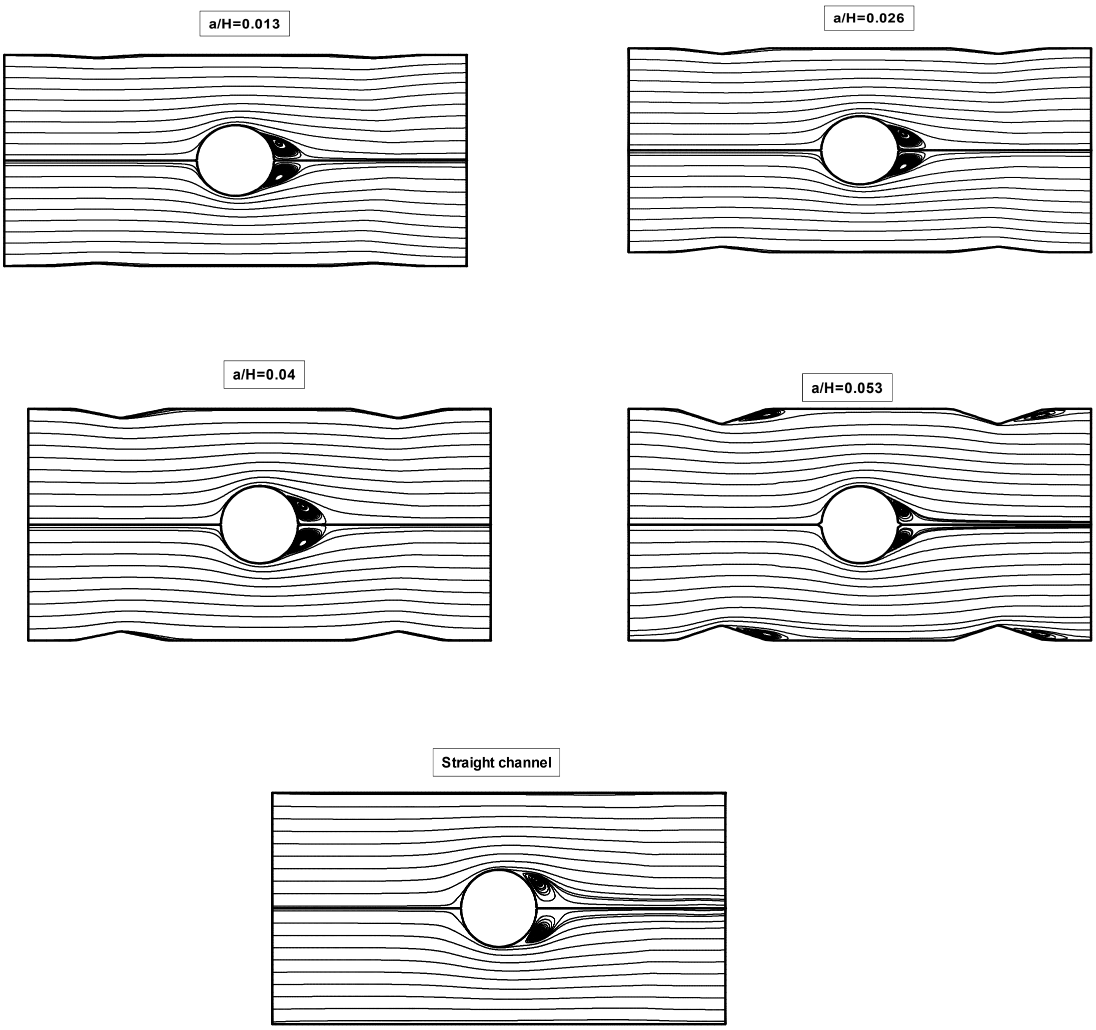


Figure (6): Streamline of flow over cylinder inside converging-diverging channel at (b/H=0.4) with deferent peak (a/H)

Local pressure drop across cylinder inside convergent-divergent channel at (a/H=0.053) with different(b/H) and (b/H=0.4) with different (a/H) is shown in figure (7- a, b), the pressure drop across cylinder is increased in convergent- divergent channel when comparison with cylinder in straight channel and maximum pressure drop be at (b/H=0.46) and increases with the increasing of (a/H).

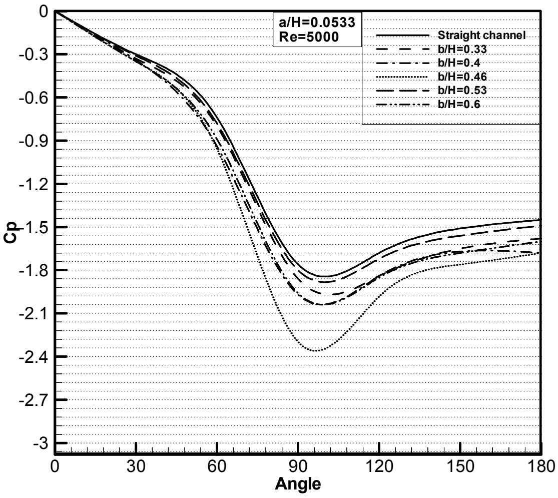


Figure (7-a): Pressure drop across cylinder inside converging-diverging channel (a/H=0.053) with different(b/H)

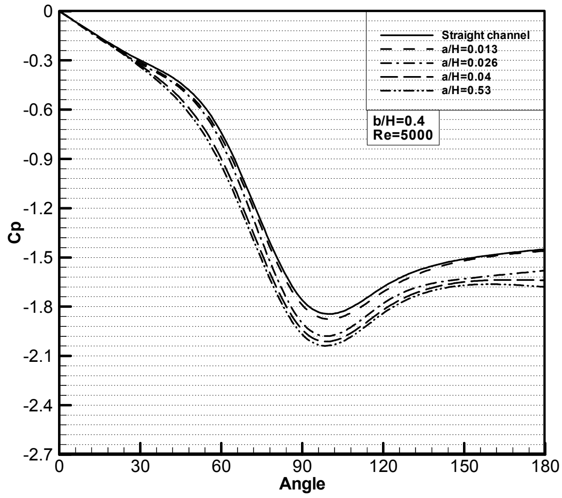


Figure (7-b): Pressure drop across cylinder inside converging-diverging (b/H=0.4) with different (a/H)

Local heat transfer (Nu) across cylinder inside convergent-divergent channel at (a/H=0.053) with different(b/H) and (b/H=0.4) with different (a/H) is shown in figure (8-a, b). Heat transfer from cylinder is increased inside convergent-divergent channel compared with cylinder inside straight channel because of the flow pass toward the cylinder is delay the separation of flow and sweep the heat to the main flow. Maximum heat transfer be at (b/H=0.46) for all (a/H) values and increasing with increase the peak.

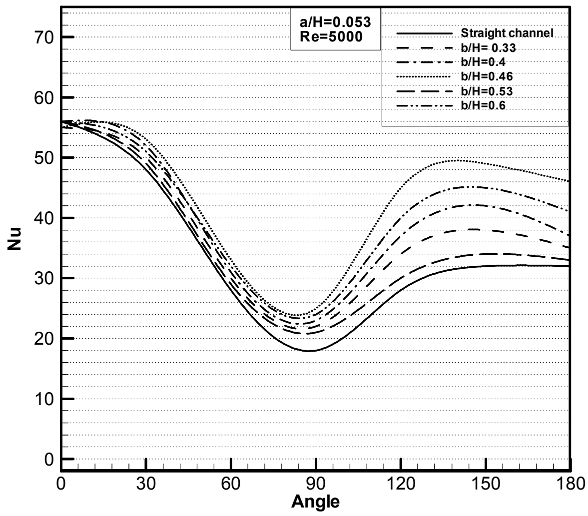


Figure (8-a) Nu across cylinder inside converging-diverging channel (a/H=0.053) with different(b/H)

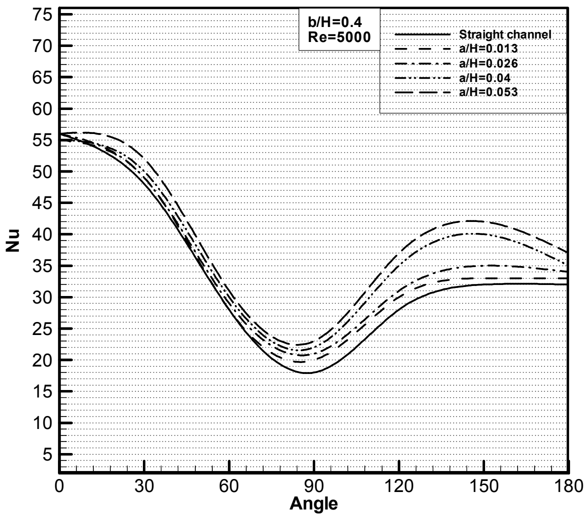


Figure (8- b): Nu across cylinder cylinder inside converging-diverging channel (b/H=0.4) with different (a/H)

Conclusions:

A numerical study has been performed on heated cylinder inside convergent-divergent channel with different Reynolds number (Re) (1000 to 10000). The peak of convergent-divergent channel (a/H) ranging from(0.013 to 0.053) and the length of base of peak (b/H) ranging from (0.33 to 0.6). The most important results can be summarized as follows :-

1- Heat transfer increases from cylinder inside convergent- divergent channel.

2- Heat transfer increases with increasing peak of channel.

3- Maximum heat transfer occurs at (b/H=0.46) for all peak values.

4-Heat transfer enhanced about 14% with increasing pressure drop about 21% at (b/H=0.46, and a/H=0.053) at Re=5000.

5-The size of wake region behind the cylinder will be decreasing when using convergent-divergent channel.

6-A recirculation of flow will be apparent behind the divergent parts of channel at high values of peak (a) and small values of base (b).

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