JJMIE

Volume 13, Number 4, December. 2019
ISSN 1995-6665
Pages 221 - 230

Jordan Journal of Mechanical and Industrial Engineering

Numerical Simulation of Forced Convection Flows over a Pair of Circular Cylinders in Tandem Arrangement

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Received 17 October. 2019

Abstract

Numerical simulation of time-dependent 2-dimensional forced convection flow over a pair of tandem circular cylinders in a rectangular channel has been carried out. An air stream of Prandtl number (Pr) of 0.702 flows over the cylinders with heated walls. The influence of spacing ratios S/D at 1.1, 1.8, 2.0, 2.2, 2.4, 3.0, 4.0 and 5.0 at Reynolds number based on cylinder diameter (ReD) = 23 500 on the heat transfer and flow parameters, such as isothermal contours, Nusselt number, vortices, drag and lift coefficients over the cylinders were determined using the finite-element based software (COSMOL Multiphysics), taking into considerations the governing equations (Continuity, Momentum and Energy) and the boundary conditions. The results show that the dynamics of the flow is altered by the spacing ratios. Further, the temperature between the cylinders drops significantly as S/D increases. The local Nusselt number on the four portions of the cylinders is found to increase as S/D increases. The shear layers and the vortices about the cylinders at S/D = 2, 3 and 4 present different flow structures. At S/D = 2, the shear layers shed from the upstream cylinder re-attach to the downstream cylinder, but no vortex shedding takes place. However, small vortices are formed between the cylinders and behind the downstream cylinder at S/D = 3. At S/D = 4, the vortices from the two cylinders combine and progress in vortex street behind the downstream cylinder. This work suggests that in minimizing the vibration of the tubes and enhancing effective heat transfer by the heat exchangers, the aforementioned parameters and conditions should be taken into consideration.

Keywords: Nusselt number, vortex shedding, shear layers, forced convection, tandem;

Nomenclature

C_D is drag coefficient; C_L is lift coefficient.
D is diameter of the cylinder in m.
h_θ is local convective heat transfer coefficient in W/m^{2}K.
k is the turbulent kinetic energy.
$Nu_\theta$is local Nusselt number.
P is pressure in Pa.
Pr is Prandtl number.
q'' is heat flux in W/m^2.
q''' is heat generation per unit volume.
Re is Reynolds number while Re_D is the Reynolds number based on diameter.
S is centre-to-centre distance between the two cylinders.
S/D is centre-to-centre spacing to diameter ratio.
S_{ij} is strain deformation.
t is time in s.
T is temperature in K.
T_m is the bulk temperature of the fluid.
T_w is the temperature at the cylinder wall.
U_∞ is free stream velocity in m/s.
V means velocity.
δ_{ij}means kronecker delta.
ε is the rate of dissipation of kinetic energy
λ is thermal conductivity of the fluid in W/mK.
μ is molecular dynamic viscosity in Pa.s
μ_t is the turbulent (eddy) viscosity.
ρ is density in kg/m^3.
τ_{ij} is Reynolds shear stress in Pa

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1. Introduction

Flow over cylinders has been extensively studied from both experimental and numerical points of view (Ishigai and Nishikawa, 1975; King and Johns, 1976; Zdravkovich, 1977; Kim and Durbin, 1988; Zhao *et al.,* 2015) due to its vast applications. For instance, Zdravkovich (1977) reviewed different arrangements of cylinders as well as the influence of spacing between them on the force coefficients and Strouhal number, and highlighted different applications which include chimney stacks in wind and jetties; offshore structures in high seas; vibration of two conductor transmission lines; vibration of heat exchanger tubes etc. It should be mentioned that thermal effects of flow about pair of cylinders at different gap spaces and Reynolds numbers (Mohsenzadeh *et al*., 2010; Moshkin and Sompong, 2009) has also received some considerable attentions. This is not surprising since this field of research is useful in accounting for loss of heat from high-rise buildings, cooling towers, offshore risers, nuclear reactor rods, cooling of electrical components, etc.

To understand the fluid-structure interaction, Zdravkovich (1985) classified fluid-static responses in a discontinuous change of flow regimes, where there are large amplitude oscillations of cylinders. He noted that there were instabilities that built up to extremely large amplitude mainly in the streamwise direction, instability which developed to certain level in the streamwise direction and instability which built up gradually predominantly in the transverse direction. For better understanding of the interaction between cylinders in tandem arrangement, Xu and Zhou (2004) in their experimental investigation at Re = 800 – 4.2 × 10^4observed that formation of shears layers separate from upstream cylinder and do not re-attach to the downstream cylinder to form vortex street at 1< S/D < 2. They also found that the shear layers separating from the upstream cylinder reattach on the upstream side of the downstream cylinder and then separate at 2< S/D < 5; and that both cylinders generate vortices at S/D=4. Further, Zhou and Yiu (2006) conducted experimental study of two cylinders in tandem arrangement with wake spacing ratios of 1.3, 2.5, 4.0 and 6.0, and uncovered two distinct flow structures which depend upon re-attachment of shear layers from the upstream cylinder to the downstream cylinder.

In order to have a clear picture of the interaction and overcome experimental limitations, numerical approaches have been used to study the flow interference as well as heat transfer between two cylinders. For example, Liu *et al.* (1998) developed a numerical approach which addressed the major issues of spatial resolution and temporal resolution to ensure computational efficiency in unsteady flow simulation. Interestingly, they were able to obtain solution for 2D unsteady laminar flows. Similarly, Farrant *et al.* (2000) adopted the cell boundary element method based on unstructured mesh to solve two-dimensional Navier-Stokes equations for laminar flow over four equispaced circular cylinders for various gap spaces of 2.0 and 4.0. For their arrangement of the four circular cylinders in square configuration at gap spacing of 2.0, in-phase vortex shedding was observed as against dominant anti-phase vortex shedding for gap spacing of 4.0. Lacovides *et al.* (2014) assessed the results of Large Eddy Simulation (LES) and four Unsteady Reynold-Averaged Navier Stokes equations (URANS), at Re_D = 41 000, of flow and heat transfer through in-line tube banks. They reported that for close-pitched large in-line tube banks, where pitch size-to-diameter ratio ≤ 1.6, flow would seek the path of least resistance and travel in a diagonal manner, but the URANS except for k-ε Linear Production model showed that straight-through flow would be returned.

Furthermore, Juncu (2007a) examined flow over two tandem cylinders at Reynolds number ranging from 1 to 30, using compact higher order finite difference method to solve the Navier-Stokes equations on bipolar cylindrical coordinates. They reported that interference effects are higher on the trailing cylinder than that on the leading cylinder, with marked increase as the Reynolds number was increased. In addition, they found that drag on each of the two cylinders is smaller than that of an isolated cylinder. Moreover, Eswaram *et al.* (2013) concluded that Strouhal numbers for two tandem cylinders are smaller than that of a single cylinder at Reynolds numbers of 200 and 15000 with spacing of 2 and 4. Kitagawa and Ohta (2008) adopted Large Eddy Simulation (LES) in the examination of two tandem cylinders at Re = 2.2 × 10^4 and S/D ranging from 2 to 5. They reported how the shear layers and vortices are formed and shed around the cylinders, as well as re-attachment of shear layers on the downstream cylinder. Dehkordi *et al.* (2011) simply repeated the simulation of the work by Kitagawa and Ohta (2008) using cartesian-based finite volume method and it was found that the two authors' results were in good agreement.

Mahir and Altac (2008) used FLUENT to determine the effects of centre-to-centre spacing ratio ranging from 2 to10 on unsteady flow past two tandem cylinders at Re_D = 100 and 200. They reported that Nusselt number of the upstream cylinder approaches that of a single isothermal cylinder at S/D > 4, and the mean Nusselt number of the downstream cylinder is about 80% of the upstream cylinder. Similarly, Juncu (2007b) reported that the Nusselt number of the upstream cylinder was greater than that of the downstream cylinder, and that it increased as the Reynolds number increased for all the Prandtl numbers of 0.1, 1, 10, and 100 considered. Also, Hamiri and Saghafian (2012) studied the effect of spacing ratio at 2, 3, 4, 5, 7 and 10 for Prandtl numbers of 0.7 and 7 with Reynolds numbers of 100 and 200 on three tandem circular cylinders. They reported that Nusselt number over the cylinders increased with increasing spacing ratio. In addition, it was observed that Nusselt number was higher at Reynolds number of 200 and Prandtl number of 7 than at Reynolds number of 100 and Prandtl number of 0.7. In their study of a mixed convection flow over two tandem cylinders, Salcedo *et al.* (2016) reported that overall Nusselt number of the downstream cylinder decreased monotonically within $-1 \leq Ri \leq 1$, but increased monotonically within $1.5 \leq Ri \leq 4$, where Ri is Richardson number.

Some engineering applications, such as nuclear reactors which involve internal heat generation may require investigation of heat transfer over them. For instance, Wang and Georgiadis (1996) reported that for an array of cylinders volumetrically heated at constant rate in a laminar flow, the temperature difference in the cylinder blocks monotonically decreases with increasing Re, whereas it increases as the fluid-to-solid conductivity increases. Furthermore, Buyruk *et al.* (1998) reported that the heat transfer from a cylindrical tube in cross flow at Re = 120 and Re = 390 increases when blockage ratio and Reynolds number increases. Buyruk (2002) extended the work of Buyruk *et al.* (1998) by examined the influence of gap spaces on tandem cylinders at Re = 300, Pr = 0.7 and S/D = 1.3 and 6. He concluded that low temperature drop occurred for the two cylinders with S/D = 1.3 while higher temperature drop occurred for the case of S/D = 6.

It is evidenced in the literature that flow-induced vibrations and interferences in the wake of two circular cylinders have great effect on the flow structure. Although a lot of studies has been conducted on two-cylinder wakes at various low and high Reynolds numbers in order to achieve a certain target, this present work examines the influence of spacing ratios S/D = 1.1, 1.8, 2.0, 2.2, 2.4, 3.0, 4.0 and 5.0 on the hydrodynamic force coefficients and heat transfer characteristics across two tandem circular cylinders at intermediate Reynolds number of 23 500. This work will complement existing work and improve our understanding on the flow-structure interaction taking into considerations the hydrodynamic force, heat transfer phenomenon and Reynolds number involve. This work will be relevant especially in heat exchanger application and other related systems. The governing equations with appropriate boundary conditions were implemented using a Finite Element based Software (COSMOL Multiphysics 5.0)

2. Problem Description, Geometry and Physics

Stream of air of Pr = 0.702 flows at free stream velocity U_∞ (4.139m/s) and free stream temperature T_∞ (25^{o}C) into a rectangular channel. Two circular cylinders are placed in-line along the centerline of the channel at some points away from the inlet. They are separated from each other by a distance of S. The centerline of the channel coincides with the centre of the cylinders. The two cylinders are heated to surface temperature T_w (70 ^{o}C) such that $T_w > T_\infty$. The bulk temperature of the fluid T_m, given by $(T_w + T_\infty)/2$ is 47.5 ^{o}C. The fluid properties such as viscosity and density of the dry air are taken at the bulk temperature. At the surface or wall of the cylinders, velocity is zero (no-slip condition) for laminar flow consideration. However, wall function is applied to the boundary condition at the solid wall for a turbulent flow. Initially, temperature of the bulk of the flow domain is assumed to be the same before heating up the cylinders. The reason for heating up the cylinders is to investigate the thermal effects around them as flow takes place over them.

Spacing ratios (S/D) of 1.1, 1.8, 2.0, 2.2, 2.4, 3.0, 4.0 and 5.0 were used for the in-line arrangement of the cylinders as shown in Fig.1. The fluid domain is rectangular, and the length of the domain is 22D (where D is diameter = 0.1 m) and the height is 4D. The first cylinder is placed at distance 2D downstream of the inlet where a parabolic velocity profile is established, while the other is placed at distance S behind it. Computational studies were carried out for the different spacing ratios at the Reynolds number Re_D = 23500. The effects of the S/D ratios on isotherms, temperature field, average Nusselt number, vortices, streamlines, drag and lift coefficients around the two cylinders were studied.

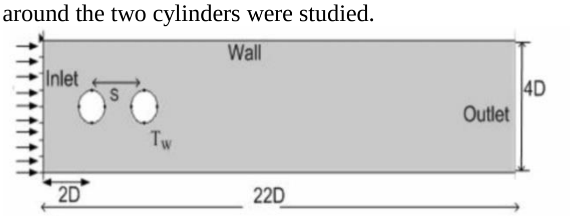

Figure 1. Tandem arrangement.

3. Governing Equations and Boundary Conditions

3.1. Governing Equations

The differential equations that govern unsteady incompressible turbulent flow are given by the continuity and momentum equations (Versteeg and Malalasekera, 2007), expressed as

$$\frac{\partial(\bar{v}_i)}{\partial x_i} = 0 \quad (1)$$

$$\frac{\partial(\bar{v}_i)}{\partial t} + \frac{\partial(\bar{v}_i\bar{v}_j)}{\partial x_j} = -\frac{1}{\bar{\rho}}\frac{\partial \bar{P}}{\partial x_i} + \frac{1}{\bar{\rho}}\frac{\partial}{\partial x_j}\left((\mu + \mu_t)\frac{\partial v_i}{\partial x_j} + \tau_{ij}\right) \quad (2)$$

where i =1, 2; $\tau_{ij} = -\overline{\rho v_i v_j} = 2\mu_t S_{ij} - \frac{2}{3}\bar{\rho}k\delta_{ij}$; $S_{ij} = \frac{1}{2}\left(\frac{\partial \bar{v}_i}{\partial x_j} + \frac{\partial \bar{v}_j}{\partial x_i}\right)$; k is the turbulent kinetic energy per unit mass; μ_t is the turbulent (eddy) viscosity; μ is molecular dynamic viscosity; and t is time. The overhead bar signifies average.

The energy equation is given as

$$\frac{\partial(\bar{T})}{\partial t} + \frac{\partial(\bar{v}_i\bar{T})}{\partial x_i} = \frac{\partial}{\partial x_i}\left\{\alpha\frac{\partial \bar{T}}{\partial x_i} + q_i\right\} + q''' \quad (3)$$

where $q_i = -\overline{v_i T}$, α is thermal diffusivity and q''' is heat generation per unit volume.

The turbulence is modelled using the standard k-ε model represented by the following equations (Versteeg and Malalasekera, 2007).

$$\frac{\partial(\rho k)}{\partial t} + \nabla.(\rho k\vec{v}) = \nabla.\left[\frac{\mu_t}{\sigma_k}\nabla k\right] + 2\mu_t S_{ij}.S_{ij} - \rho\varepsilon \quad (4)$$

$$\frac{\partial(\rho\varepsilon)}{\partial t} + \nabla.(\rho\varepsilon\vec{v}) = \nabla.\left[\frac{\mu_t}{\sigma_\varepsilon}\nabla\varepsilon\right] + c_{1\varepsilon}\frac{\varepsilon}{k}2\mu_t S_{ij}.S_{ij} - c_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \quad (5)$$

Turbulence viscosity is given by $\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$; $c_{1\varepsilon} = 1.44$, and $c_{2\varepsilon} = 1.92$; $c_\mu = 0.09$; $\sigma_k = 1.00$; $\sigma_\varepsilon = 1.30$ are constants and $\vec{v}$ is the velocity vector.

The lift and drag coefficients are evaluated using the following equations (Cengel and Cimbala, 2006).

$$C_D = \frac{F_D}{\frac{1}{2}\rho U_\infty^2 D} \quad (6)$$

$$C_L = \frac{F_L}{\frac{1}{2}\rho U_\infty^2 D} \quad (7)$$

The heat transfer q″ over the cylinders is evaluated as follows (Lienhard and Lienhard, 2006).

$$q'' = -\lambda_{fluid}\frac{dT}{dx}\Big|_{x=0} \quad (8)$$

$$q'' = h_\theta(T_w - T_\infty) \quad (9)$$

$$h_\theta = \frac{-\lambda_{fluid}\frac{dT}{dx}\big|_{x=0}}{(T_w - T_\infty)} \quad (10)$$

where λ is thermal conductivity of the fluid.

In order to measure the convective heat transfer at a solid surface of the cylinder, a dimensionless number called Nusselt number is defined based on diameter. It is given as

$$Nu_\theta = \frac{h_\theta D}{\lambda} \quad (11)$$

3.2. Choice of standard k-ε model over k- ω turbulence model

Both standard k-ε model and k- ω turbulence model are available for simulating fluid flow in COMSOL Multiphysics. COMSOL User Manual (2014) compares the strength and weakness of the two models. Standard k-ε model is suitable for flows over bodies (external flow), has good convergence rate and requires low computational memory. However, k- ω turbulence model is more suited to internal flow and cases of strong curvature or separated flows, but it requires more time for convergence.

Furthermore, dependent of k- ω turbulence model on the assumed free stream value of ω makes it less suitable for external flows (Menter, 1992). This current study is about external flow problem, thus the choice of standard k-ε model for the simulation.

3.3. Boundary conditions

The boundary conditions adopted for this study are described as follows.

1. Inlet: $u = U_\infty$, $T = T_\infty$. The inlet is 2D away from the cylinders
2. Outlet: $\frac{\partial u}{\partial x} = 0$; $\frac{\partial v}{\partial x} = 0$; $\frac{\partial T}{\partial x} = 0$; and pressure p=0.
3. The top and bottom sides as well as the cylinder walls use the slip wall boundary condition.

The slip wall boundary condition ensures that fluid does not leave the domain or does not penetrate through the cylinder. Mathematically, the constraint is defined in COMSOL 5.0 (2014b) as:

$$\vec{\boldsymbol{v}} \cdot \boldsymbol{n} = 0 \tag{12}$$

and

$$\boldsymbol{K} - (\boldsymbol{K}.\boldsymbol{n})\boldsymbol{n} = 0 \tag{13}$$

where

$$\boldsymbol{K} = \mu(\nabla\vec{\mathbf{v}} + (\nabla\vec{\mathbf{v}})^T)\boldsymbol{n} \tag{14}$$

4. The temperature at the upper and bottom surfaces is $T = T\infty$, while the cylinder wall temperature is set to $T=Tw$.

COMSOL Multiphysics 5.0 is used to solve the governing differential equations along with the defined boundary conditions.

4. Mesh Generation

4.1. Geometry Discretization

The governing equation and boundary conditions are solved by first dividing the problem geometry into smaller elements. Mesh generation is the step-in numerical simulation with which a set of elements are obtained. The equations are then solved for each of the elements in the mesh generated to obtain pressure, velocity and temperature at every node in the computational domain.

The mesh size determines the accuracy of the solution that will be obtained. COMSOL Multiphysics has pre-defined meshes which include, 'coarse', 'normal', 'fine', 'finer', and allows for users to define mesh sizes according to the desirable level of accuracy expected. It is important to state that the memory of the computational facility may limit the user from defining very small and fine mesh sizes. More so for very complex problems, the use of very fine mesh imposes more constraints on the requirement of the computational facility.

4.2. 4.2 Mesh Dependency

Few simulations were run for a forced convection flow over a single cylinder, using different COMSOL physics-defined mesh sizes known as 'normal', 'fine' and 'finer'. Table 1 shows and compares that the Nusselt number values obtained from the three mesh sizes are close to each other, with only less than 5% discrepancy between normal size and fine size, and between fine size and finer size. Normal mesh size is just enough for the single cylinder, but we select fine mesh size for the present work as it offers results most stable and which are in good agreement with literature (Table 3 in section 5.1). Moreover, fine mesh uses much less average computational time and memory than finer mesh.

Table 1. Impact of quality of mesh on Nusselt number for a single cylinder

	Normal Mesh Size (6522 elements)		Fine Mesh Size (12690 elements)		Finer Mesh Size (29178 elements)	
Re_D	Nu	Time (s)	Nu	Time (s)	Nu	Time (s)
200	8.006	1314	8.008	1947	7.990	3249
300	9.758	1460	9.940	2102	9.7415	3226
400	11.119	1562	11.520	5340	11.397	5104
500	12.208	1593	12.804	4860	12.789	5516

A typical meshing of the two tandem cylinders in rectangular domain at S/D = 5.0 is shown in Fig. 2. The mesh statistics including domain elements, boundary elements, number of degree of freedoms (DOFs) and solution time are shown in Table 2, as computed using the default physics defined 'fine' mesh.

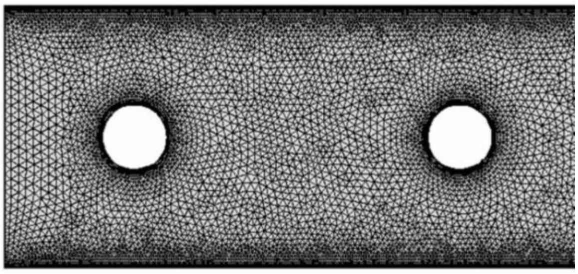

Figure 2. Fine mesh generated at S/D = 5.0.

Table 2. Mesh Statistics for Tandem Arrangement at Re_D =23500

S/D	Mesh Type	Domain Elements	Boundary Elements	†nDOF(plus 1074DOFs)	Solution Time(s)
1.1	Fine	30328	1062	109622	6184
2.0	Fine	30530	1062	110228	4891
2.2	Fine	30564	1062	110330	7038
2.4	Fine	30410	1062	109868	7348
3.0	Fine	30394	1062	109820	7934
4.0	Fine	30620	1062	110498	28979
5.0	Fine	30660	1062	110618	7613

†nDOF means number of degree of freedoms.

5. Results and Discussion

5.1. Validation of solution method

In order to validate the solution obtained in this study, simulation of forced convective heat transfer was carried out for a single cylinder. The results obtained for a single cylinder, using normal mesh size, was compared with the analytical result of Khan *et al.* (2004), and numerical results of Salcedo *et al.* (2016) and Mettu *et al*. (2006) as in Table 3.

Table 3. Comparison of mean Nusselt number for a single cylinder for Re = 200 – 500

Re_D	Nu			
	Present	Salcedo et al (2016)	Khan et al. (2004)	Mettu et al. (2006)
200	8.006	8.003	7.747	7.592
300	9.758	9.563	9.489	9.326
400	11.119	10.795	10.950	10.910
500	12.208	12.108	12.210	12.130

5.2. Heat transfer characteristics

The heat transfer patterns over the cylinders are shown by the means of isotherm contours, surface temperature plot, and Nusselt number. Figure 3 shows the heat transfer characteristics over the cylinders and downstream of the flow channel. Figures 3a and 3c show the isotherm contours over the cylinders at S/D = 1.1 and 2.0. Figures 3b and 3d show the temperature surface plots for the temperature variation between the cylinders and behind the downstream cylinder at the same S/D ratios. There are significant changes in the flow patterns as reflected both in the isotherm contours as well as in the surface plot. It can be inferred that the heat transfer is being caused by the flow. This is also true for S/D = 3 and 4, shown in Fig. 4. Comparing the temperature contour and surface plots in Figs. 3 and 4 at S/D =1.1, 2.0, 3.0 and 4.0, it can be deduced that the temperature drop between the cylinders increases as the spacing ratios increases. This was also reported by Buyruk (2002). This is likely due to turbulent activities being strongly aggravated by the cylinders. The present result suggests that the closer the cylinders the more the turbulent activities

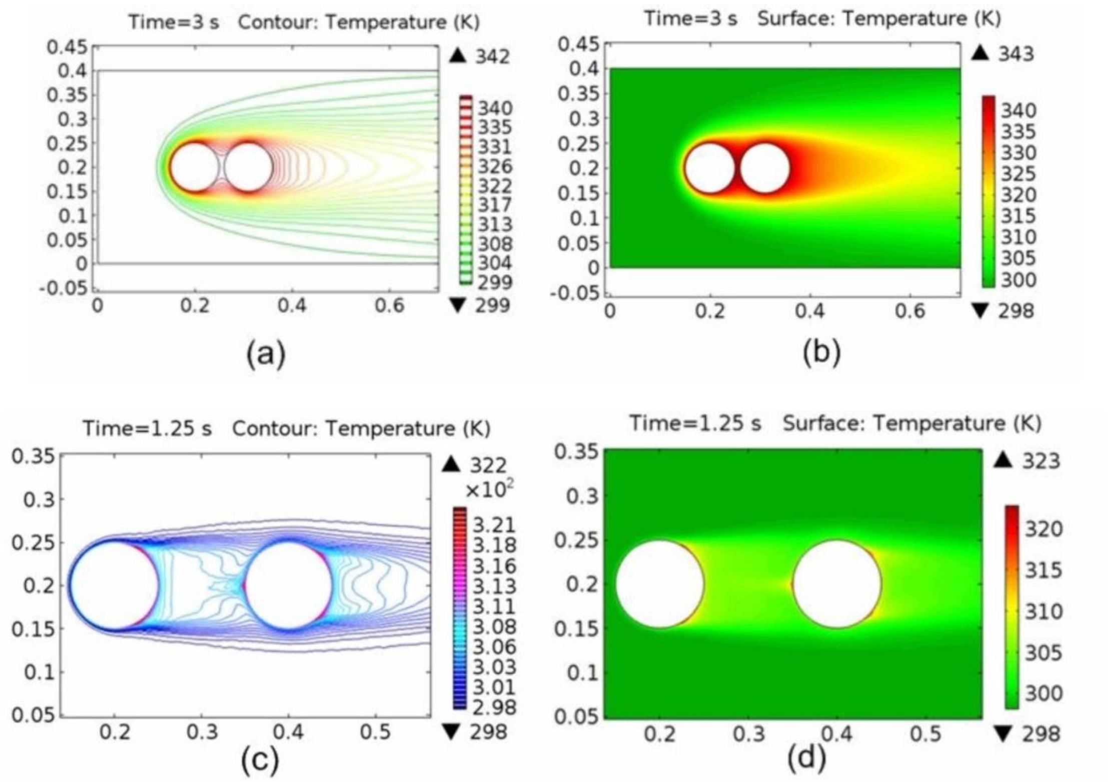

Figure 3. The Isotherms and surface temperature plots around the cylinders at S/D = 1.1 (aand b) and S/D = 2 (c and d).

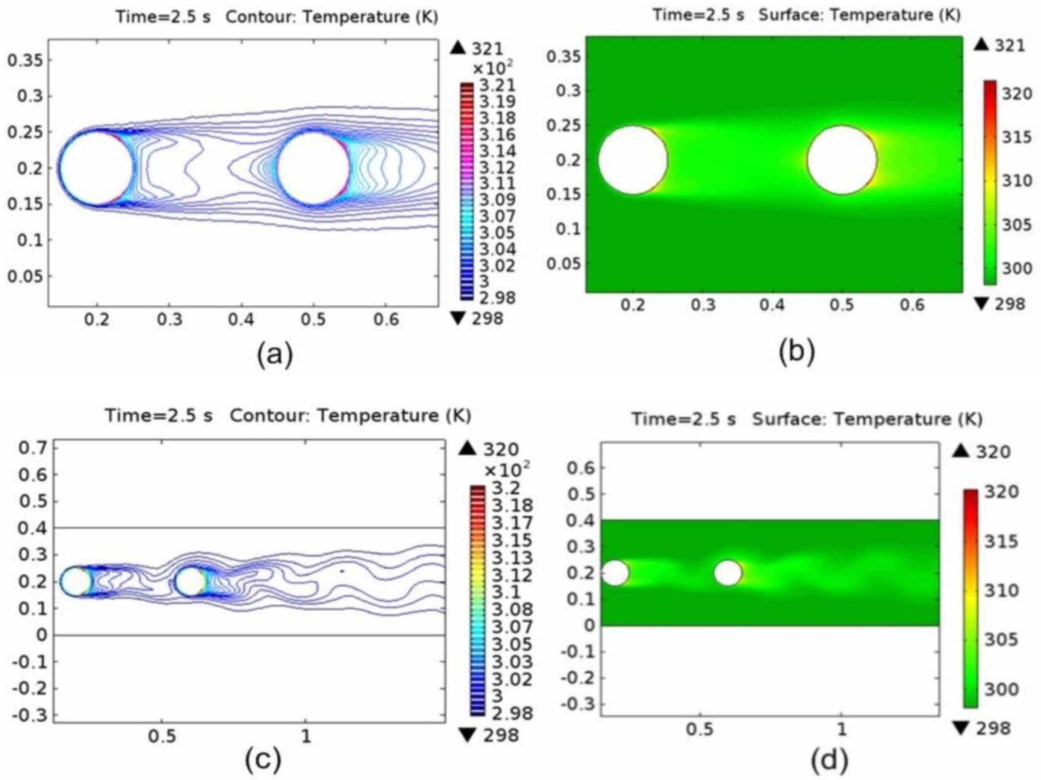

Figure 4. The Isotherms and temperature plot around the cylinders at S/D = 3 (a and b) and S/D = 4 (c and d).

Moreover, since there are different heat characteristics on the various parts of the cylinders, the variation of Nusselt number with S/D ratio should reflect the heat transfer phenomenon between the cylinders. This is presented in Fig. 5. Figure 5a illustrates the relationship of the mean Nusselt number (by considering the two cylinders simultaneously) of each of the four surfaces recognized on the cylinders with the spacing ratio. The Nusselt number increases for increasing S/D ratio within 1<S/D≤4, but sharply decreases for S/D = 5. Similar relationship holds for the mean value of the whole two cylinders. Figure 5b shows the variation of the mean Nusselt number of the two whole cylinders with the spacing ratio. Therefore, it can be deduced from Fig. 5 that the mean Nusselt number is greatest at the front portions of the cylinders. Similarly, Rosales *et al.* (2001) also reported that Nusselt number of the front face of a heated square cylinder at Reynolds number of 500 has the highest surface-averaged Nusselt number compared to other portions of the cylinder

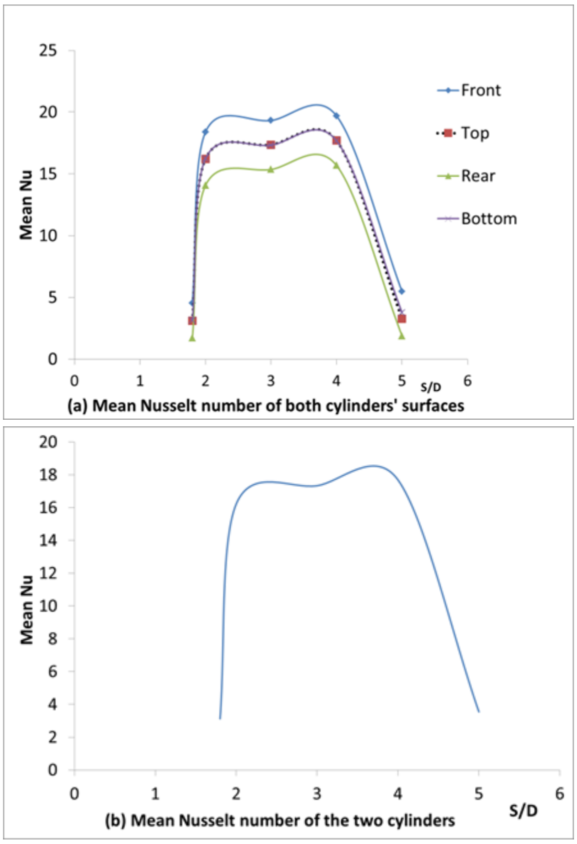

Figure 5. The variation of Mean Nusselt number with spacing ratio (a) for four surfaces; (b) for two cylinders.

5.3. Vortex structure and flow pattern

Figure 6 shows the flow pattern over two cylinders in tandem arrangement for S/D = 2.0 at dimensionless time tU_∞ /D = 55. The shear layers shed from the upstream cylinder reattach to the downstream cylinder. It should be noted that there is no vortex shedding at S/D =2 as shown in Fig. 6a and the streamline plot (Fig. 6b) shows a stable flow pattern. At S/D = 3, the shear layers shed from the upstream cylinder re-attach symmetrically to the downstream cylinder (Fig. 6c). Further, as time progresses, tiny and small vortices are formed between the cylinders, and vortex shedding takes place for a short time behind the downstream cylinder at S/D = 3.

However, the flow phenomenon is entirely different for S/D >3. The streamlines in Fig. 7 shows that there is an unstable flow pattern at S/D = 4 as reflected in the streamlines at tU/D = 75 and 83. This is not surprising, as Fig. 8 vividly confirms this incident. Figure 8 shows that vortices shed from the upstream cylinder reattach to the downstream cylinder. It should be mentioned that these vortices combine, and progress in the vortex street behind the downstream cylinder which eventually results in the strong agitation of the layer. The behavior suggests that these vortices are responsible for the high instability in the flow. The vortex structure and shear layers described for S/D = 2,3 and 4 are consistent with those reported by Xu and Zhou (2004) and Kitagawa and Ohta(2008). However, vortex shedding did not occur at S/D > 4 at Re_D = 23500.

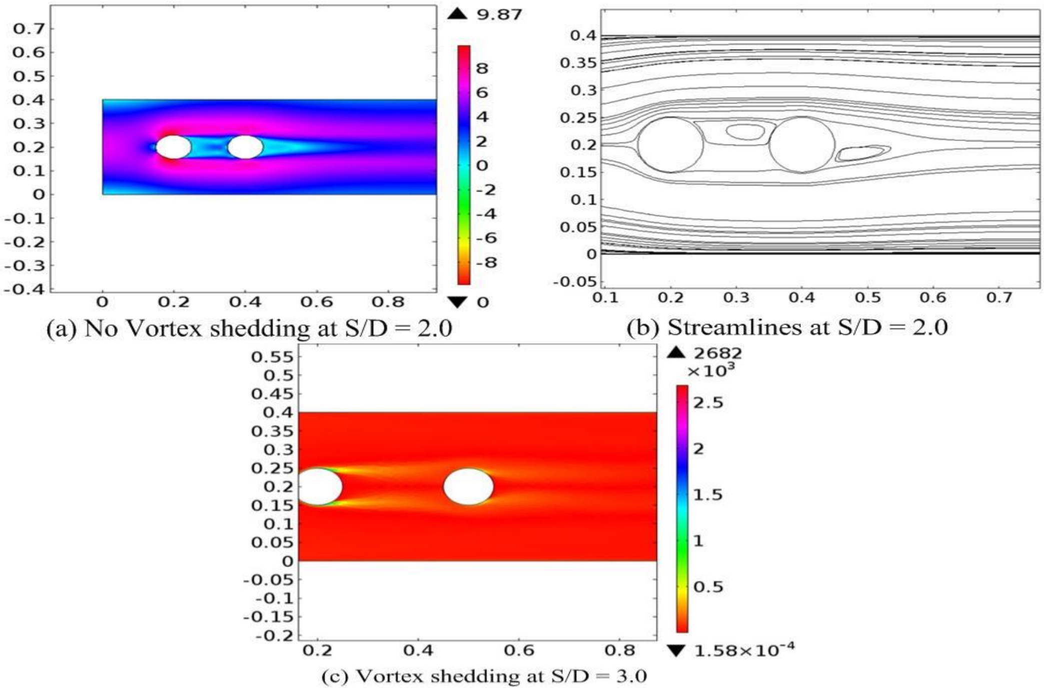

Figure 6. Flow pattern over two cylinders for S/D = 2.0 and 3.0 at tU/D = 56.

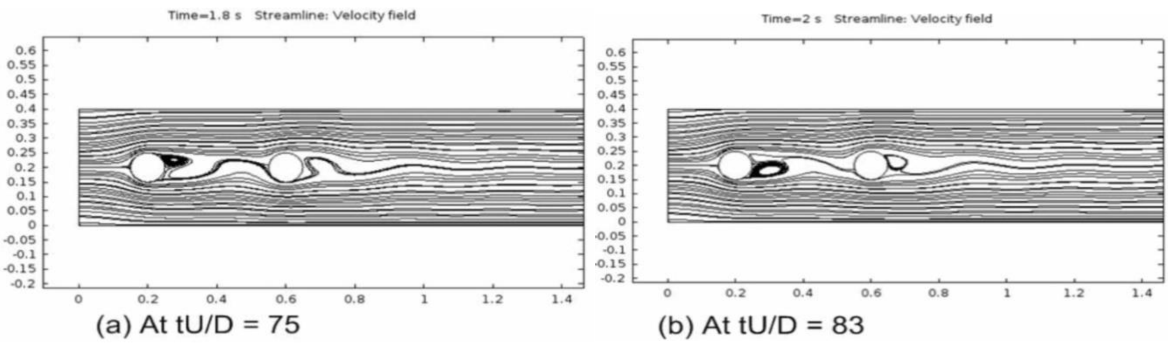

Figure 7. Streamlines over two cylinders in tandem arrangement for S/D= 4.0.

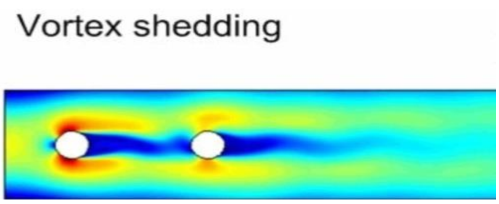

Figure 8. Vortex shedding over two cylinders in tandem arrangement for S/D = 4.0.

5.4. Lift and drag coefficients

The previous results indicated that the dynamics of the layer is influenced by the spacing ratios between the two cylinders. The effect of the spacing ratios on the lift and drag coefficients should shed more light on these changes. At S/D = 4 (Fig. 9), the drag coefficients for the upstream and downstream cylinders are positive but of different magnitudes. Figures 9a and 9b also show that lift coefficients CL_1 and CL_2 oscillate about the mean zero level. It should be noted that the vortices shed from the upstream cylinder at the S/D = 4 cause vortices in the wake of the downstream cylinder and hence, result in high flow instability. This invariably led to high fluctuation in the lift and drag coefficients. Figure 10 shows the variation of mean drag coefficients of the two cylinders with the spacing ratios.

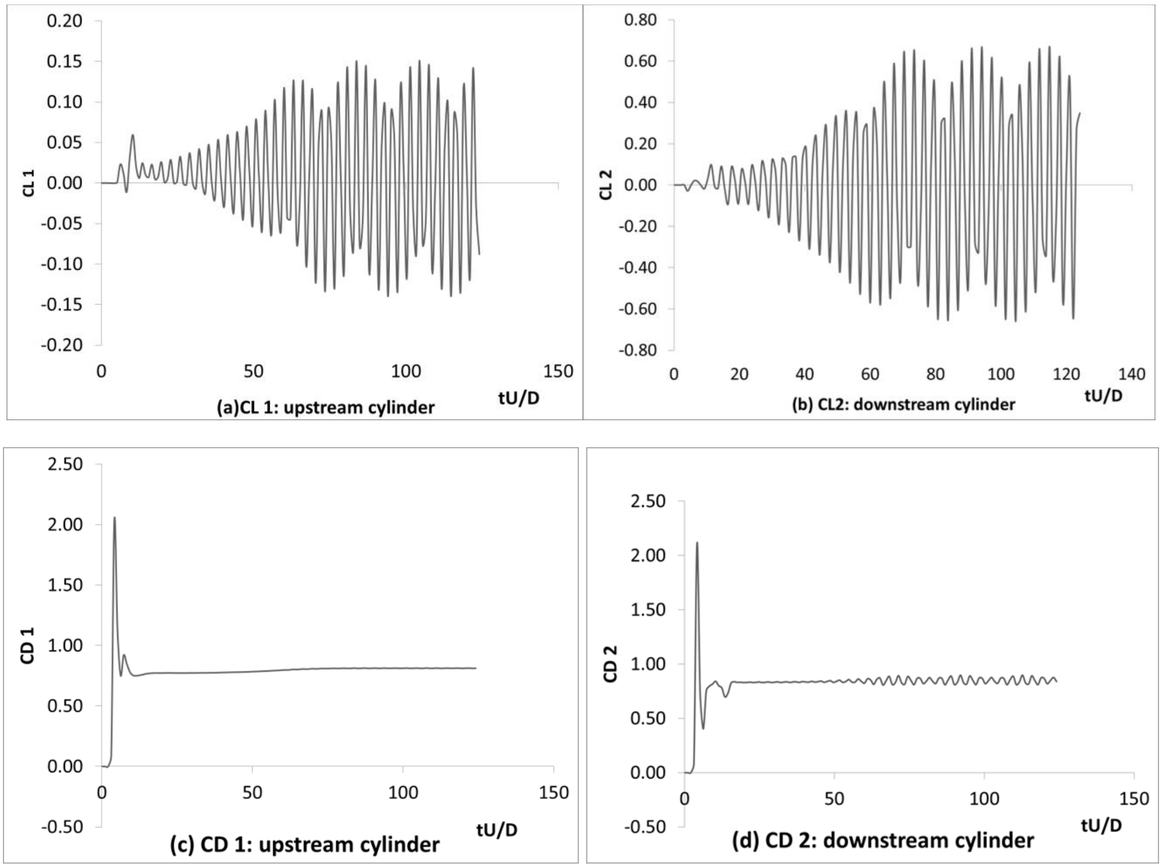

Figure 9. Lift and drag coefficient fluctuations at S/D =4.

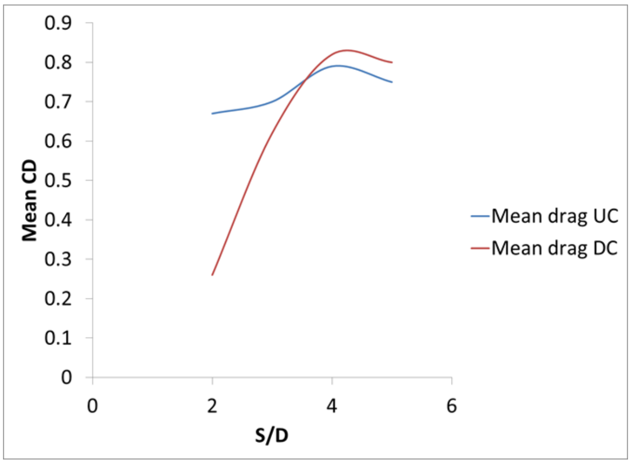

Figure 10. Variation of mean drag coefficient with the spacing ratios.

It can be deduced that the mean drag coefficient of the cylinders increases linearly for 2≤S/D≤4 and later decreases. The mean drag coefficient of the upstream cylinder is higher than those of the downstream cylinder for 2≤S/D<3. Surprisingly, the peak values occur at the same S/D (= 4), where $CL_1 > CL_2$. The result suggests that while the dynamics of the layer alter the magnitude of the mean lift coefficients of the two cylinders in an unequal manner, the oscillations are similar. The drag coefficients and the Strouhal number for these spacing ratios at Re_D = 23500 are presented in Table 4.

Table 4. Drag coefficients and Strouhal number of the cylinders at Re_D = 23500

S/D		C_D	St
2	*UC	0.67	0.193
	**DC	0.26	0.242
3	UC	0.70	0.269
	DC	0.62	0.269
4	UC	0.79	0.338
	DC	0.82	0.338
5	UC	0.75	-
	DC	0.80	-

*UC = Upstream Cylinder; **DC =Downstream Cylinder.

6. Conclusion

The influence of spacing ratios on the heat transfer and flow parameters in a 2-dimensional forced convection flow of air over a pair of tandem circular cylinders in a rectangular channel has been carried out numerically at Re_D= 23 500. The results revealed that the dynamics of the flow depend strongly on the spacing ratios taking into consideration the flow and the initial conditions. It was discovered that the temperature drop within the gap between the cylinders is higher at larger spacing ratios where high vortex-induced vibration also occurred. The mean Nusselt number over the tandem cylinders generally increases (directly) with increasing spacing ratios for 1<S/D≤4. Shear layers shed from the upstream cylinder re-attached to the downstream cylinder, and there is no vortex shedding at S/D = 2. There is re-attachment of the shear layers shed from the upstream cylinder onto the downstream cylinder at S/D = 3. At this ratio, vortex shedding takes place within the gap as well as behind the downstream cylinders. The flow pattern is highly unstable for S/D = 4, and vortices shed from the upstream cylinder impinged on the downstream cylinder, and this progresses in vortex street behind the downstream cylinder. This causes high fluctuations in the lift and drag coefficients for the spacing ratio.

The present work suggests that the above set of parameters and conditions should be taken into considerations in the operation of heat exchangers in order to minimize the vibration of the tubes as well as to ensure efficient and effective heat transfer by the heat exchangers.

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Declaration on Conflict of Interest

The authors declare that there is no conflict of interest among them. All the authors read and approved the article. In addition, they contribute significantly to the article.