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Heat Transfer from Two Cam Shaped Cylinders in Side-by-Side Arrangement

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***Abstract*—**Heat transfer from two cam shape cylinder in side-by-side arrangement had been studied numerically.

The transverse gap between the centers of cylinders (T) is allowed to vary to change the pitch ratio (T/D_{eq}). The equivalent diameter of the cylinder (D_{eq}) is 27.6 mm and pitch ratio varies in range of $1 \leq T/D_{eq} \leq 3$. The Reynolds numbers based on equivalent circular cylinder are within $50 \leq Re_{eq} \leq 150$. Results show that Nusselt number of cylinders increases about 1 to 36 percent when pitch ratio increases from 1 to 3.

***Keywords*—**Cam shaped, side-by-side cylinders, numerical, heat Transfer.

I. Introduction

THE convection heat transfer around multiple bluff bodies has wide engineering applications such as heat exchangers, space heating, cooling towers, oil and gas pipelines, electronic cooling and so on.

Spatial arrangement of two cylinders can be classified into three categories, namely, aligned with the direction of the main flow (in tandem), placed side-by-side, and placed in a staggered arrangement. There are many experimental and numerical studies [1]–[3] devoted to the flow and heat transfer over two circular cylinders with different arrangement. As it is clear from the previous studies [4], [5] drag coefficient of circular tube is more than streamline cylinder. There are some studies about flow and heat transfer around side-by-side streamline cylinder.

Flow around a pair of side-by-side square cylinders using the lattice Boltzmann method investigated by Agrawal, Djenidi and Antonia [6]. The effects of the gap ratio s/d (s is the separation between the cylinders and d is the characteristic dimension) on the flow were studied.

It has been found that vortices develop behind the cylinders for $Re \approx 10$ and the flow starts to oscillate at $Re \approx 30$ with two cylinders in the flow at s/d = 2.5. The existence of both synchronized and flip-flop regimes with square cylinders, in agreement with the well known results for circular cylinders. However, by comparison to circular cylinders, the transition between these regimes occurs for a much larger spacing between the cylinders.

Kun et al [7] studied wake pattern of two circular cylinders at low Reynolds number in side-by-side arrangements. Reynolds number varied within 30< Re<100 and gap spacing (T/D) varied from 1.1 to 3. They observed nine kinds of wake patterns, four steady wake pattern and five unsteady wake patterns.

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Their results confirmed that wake patterns significantly depended on gap spacing. However, wake patterns are relatively intensive to the Reynolds number expect for intermediate gap spacing.

Yoon, Seo and Kim [8] studied laminar forced convection heat transfer around two rotating side-by-side circular cylinder at a various range of absolute rotational speeds ($|\alpha| \leq 2$) for four different gap spacing of 3, 1.5, 0.7 and 0.2 at Reynolds number of 100. They found that as $|\alpha|$ increase the thermal field become stabilized and eventually steady beyond critical rotational speed depending on the gap spacing.

The aim of characterizing its features or developing reduced-order models that predict the induced drag forces on it. So, the purpose of this study is to numerically investigate the convection heat transfer characteristics of two cam shaped cylinders of equal equivalent diameter in side-by-side arrangements subject to cross flow of air.

II. Problem Description and Governing Equations

The cross section profile of the cylinder comprised some parts of two circles with two line segments tangent to them. The cylinder have identical diameters equal to d=11 mm and D=22 mm with distance between their centers, l=13 mm, (Fig. 1). Characteristic length for this tube is the diameter of an equivalent circular cylinder, $D_{eq}=P/\pi=27.6$ mm, whose circumferential length is equal to that of the cam-shaped cylinder.

The typical solution domain and the cylinder boundary definition and nomenclature used in this work are shown in Fig. 2 the inlet flow has a uniform velocity U_∞. The velocity range considered only covers laminar flow conditions. The solution domain is bounded by the inlet, the outlet, and by the plane confining walls, AB and CD. These are treated as solid walls, while AC and BD are the flow inlet and outlet planes.

In order to decrease the effect of entrance and outlet regions, the upstream and downstream lengths are $15D_{eq}$ and $50D_{eq}$, respectively and for neglecting the wall effects on cylinders the distance between cylinders and walls is $15D_{eq}$.

Equations are written for conservation of mass, momentum and energy in two dimensions. Cartesian velocity components U and V are used, and it has been assumed that the flow is steady and laminar, while the fluid is incompressible and Newtonian with constant thermal and transport properties. Furthermore, the effects of buoyancy and viscous dissipation are neglected. The governing equations consist of the following four equations for the dependent variables U, V, P and T:

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

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial P}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial P}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \quad (3)$$

$$\frac{\partial}{\partial x}(uT)+\frac{\partial}{\partial y}(vT)=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \quad (4)$$

Equations (1) to (4) are the conservation of mass, x and y direction momentum and energy equations, respectively. The boundary conditions used for the solution domain shown in Fig. 2 are uniform inlet velocity, outflow and no-slip cylinder surface boundary. The Nusselt number for a cam cylinder is respectively as follows:

$$Nu=\frac{q\,D_{eq}}{A\,k\,\Delta T} \quad (5)$$

Where q is the total rate of heat transfer to the fluid and A is the total surface area of tubes. The temperature of the cylinders wall is 400 K and the bulk temperature of the cross-flow air is 300 K and ΔT is the difference between these temperatures.

III. Numerical Method

This problem considers a 2D section of a cam shaped cylinder. For the simulations presented here, depending on the geometry used, fine meshes of 70000 to 100000 elements for T/D_{eq}=1 to 3 were used. The computational grids used in this work were generated using the set of regions shown in Fig. 3.

In this domain quadrilateral cells are used in the regions surrounding the cylinder walls and the rest of the domain. In all simulation, a convergence criterion of 1×10^{-6} was used for all variables.

The governing equations with appropriate boundary conditions are solved using finite volume approach based in Cartesian and coordinate systems.

The second order upwind scheme was chosen for interpolation of the interpolation of the flow variables. The SIMPLEC algorithm [9] has been adapted for the pressure velocity coupling.

IV. Results and Discussion

For the purpose of the validation of the solution procedure, it is essential that numerical simulations be compared with experimental data. Fig. 4 compares the Nusselt number of circular cylinder with the results of Zhukauskas [10]. There is a difference of about 2.5 percent between the present results and the results of Zhukauskas. It can therefore be concluded that the CFD code can be used to solve the flow field for similar geometries and conditions.

Effects of the increasing pitch ratio from 1 to 3 over total heat transfer of the cam shaped cylinders presented in Fig. 5.

The Nusselt number which presented in this figure is average Nusselt number of both cylinders. By comparing these results with heat transfer from single cam shaped cylinder it can be seen that effect of pitch ratio on Nusselt number of cam shaped cylinders at T/Deq>1.5 is inconsiderable and by increasing pitch ratio from 1.5 to 3 heat transfer increase up to 1 percent. However pitch ratio has considerable effect on Nusselt number in small pitch ratio. Results show that by increasing pitch ratio from 1 to 1.5 Nusselt number of cylinders increase about 2 to 35 percent for various rage of Reynolds number.

Fig. 6 shows the relations between Nusselt number with drag coefficient and Reynolds number for various pitch ratio of first cylinder.

V. Conclusion

In this study heat transfer of two cam shaped cylinders in side-by-side arrangement had been investigated. The dependency of the Nusselt number for cam shape cylinders on pitch ratio is quite clear from the results. The heat transfer from the cylinders is similar to single cylinder when pitch ratio is more than $1.5D_{eq}$. However the results show that pitch ratio has more effect on small pitch ratio T/D<1.5, by increasing the pitch ratio from 1 to 1.5 Nusselt number increases about 2 to 35 percent.

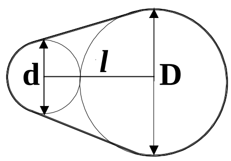

Fig. 1 Schematic of a Cam Shape Tube

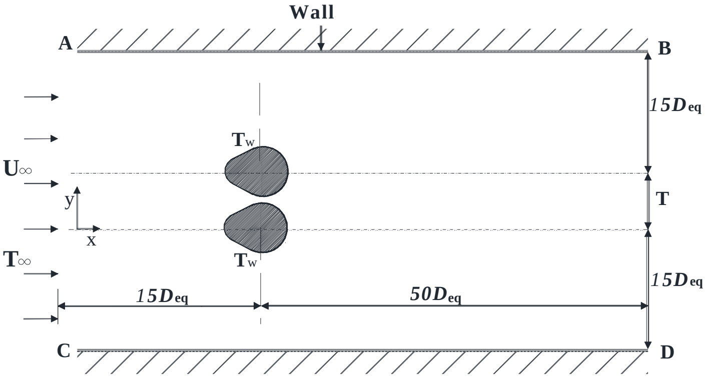

Fig. 2 Solution Domain

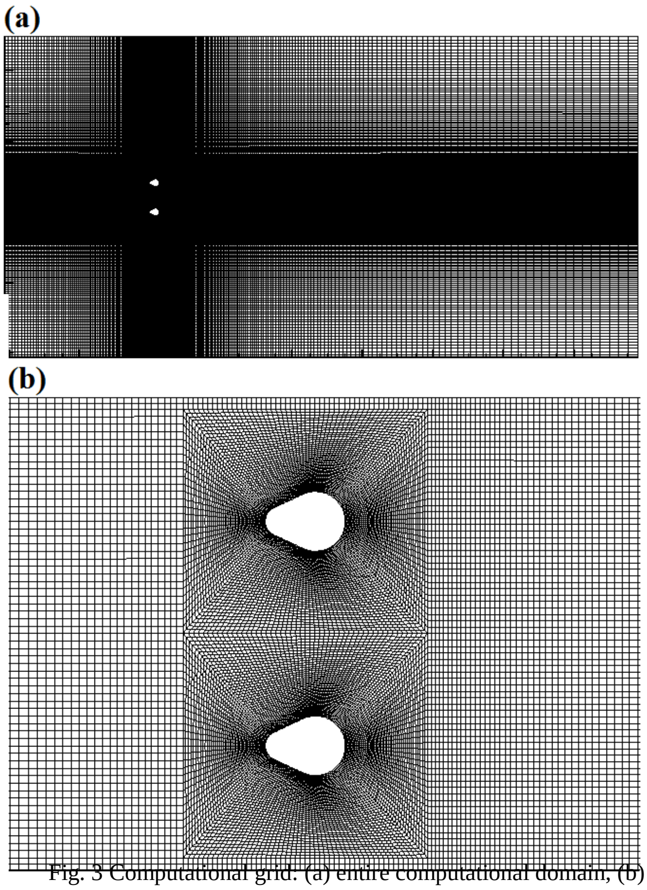

Fig. 3 Computational grid: (a) entire computational domain, (b) closer view around cylinders

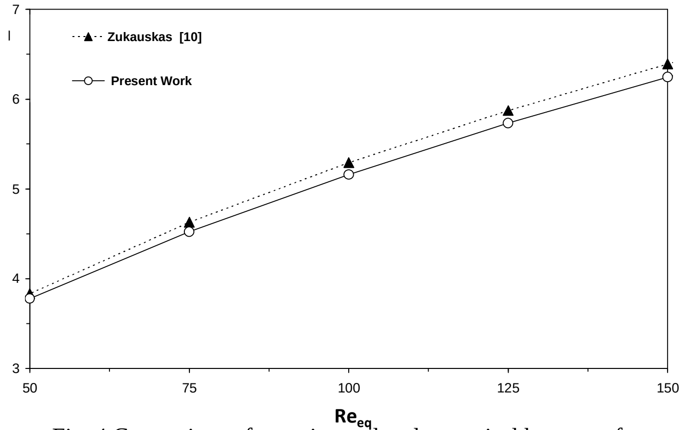

Fig. 4 Comparison of experimental and numerical heat transfer

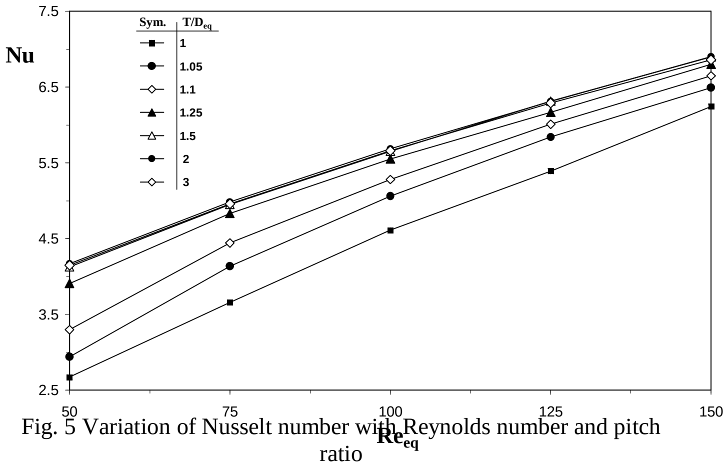

Fig. 5 Variation of Nusselt number with Reynolds number and pitch ratio

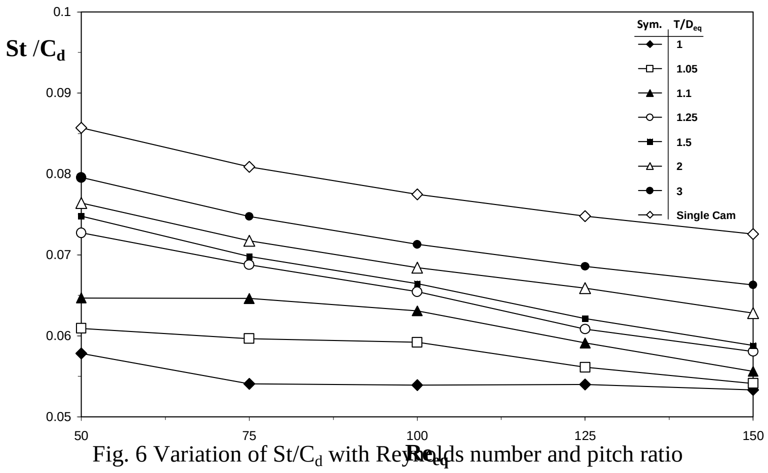

Fig. 6 Variation of St/C_d with Reynolds number and pitch ratio

NOMENCLATURE

d	Small diameter
D	Large diameter
T	Transverse gap between the centers of cylinders
ℓ	Distance between centers of circles
P	Pressure, circumferential length
Re	Reynolds number, $U_\infty D/n$
St	Stanton number, Nu/(Re.Pr)
T	Temperature
U	x-direction velocity
V	y-direction velocity
X	Distance between stagnation point and every Point on circumferential length

(i) Greek

ρ	Density
μ	Kinematic viscosity

(ii) Subscripts

Cam	Cam-shaped cylinder
Cir	Circular cylinder
eq	Equivalent
∞	Free stream

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