NEAR FIELD AND HEAT TRANSFER FOR A TRANSVERSELY OSCILATING CIRCULAR CYLINDER IN UNIFORM FREESTREAM FLOW

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Abstract. *This paper deals with a vortex and heat particle method to investigate the viscous flow around a heated circular cylinder oscillating transversely in an uniform incident flow. In the numerical simulations the Reynolds number is kept in high value. In our preview work, one can observe that the fluid flow develops according to three possible modes or bands. Here, our investigations are concentated at Band III, which is defined by values of high body oscillation frequencies. Within this band, the inertial forces (body oscillation frequency) dominate the phenomena and the frequency of vortex shedding is locked to it. The influence of the frequency and oscillation amplitude on the aerodynamic forces and forced convection heat transfer are discussed.*

Keywords*: heat transfer, Kármán vortex frequency, body oscillation frequency, lock-in, bluff body*

1. INTRODUCTION

In the areas of mechanical, aeronautical and naval engineering, incompressible external flows are very common to find and difficult to analyze. When a bluff body is embedded in a fluid flow it may cause wake to form behind the body. Understanding vortex-shedding is of great importance in the design of a variety of offshore engineering structures, tall buildings, bridges, ground vehicles, submarines, heat exchanger tubes and electricity cables. Cylinders having a two-dimensional structure are very suitable for restricting the complexity and thus observing the fundamental features of the flow.

One of the most interesting features of this flow is the phenomenon of synchronization, in which the frequency of vortex shedding, f, coincides with that of the cylinder oscillation, f_b; this is also known as "lock-in". The literature is plenty of materials on the subject and previous works have reported the lock-in features and vortex shedding patterns in the flow around an in-line oscillating circular cylinder or a transversely oscillating circular cylinder. For instance, vortex shedding frequency f had been found to lock-in to the forcing frequency f_b when f_b is close to the free vortex shedding frequency f_{so} in the transverse oscillating case. But, the in-line vibration lock-in takes place at a number of multiple ratios of f_b/f_{so}, especially, at $f_b/f_{so} = 2.0$, where the lift and drag forces increase greatly. Comprehensive reviews can be found in Koopman (1967), Sarpkaya (1979), Bearman (1984), Williamson and Roshko (1998), Blevins (1990), Griffin and Hall (1991) and Hirata *et al.* (2008).

The development of Lagrangian numerical methods, based on vortex methods, offer a number of advantages over the more traditional Eulerian schemes for the analysis of the external flow that develops in a large domain; the main reasons are [Leonard (1980), Sarpkaya (1989), Lewis (1999), Kamemoto (2004), Alcântara Pereira *et al.* (2004) and Stock (2007)]: (i) as a fully mesh-less scheme, no grid is necessary; (ii) the computational efforts are directed only to the regions with non-zero vorticity and not to all the domain points as is done in the Eulerian formulations; (iii) the far away downstream boundary condition is taken care automatically which is relevant for the simulation of the flow around a bluff body (or an oscillating body) that has a wide viscous wake.

Recicar *et al.* (2006) handled vortex elements to deals with the analysis of a circular cylinder oscillating around a fixed position which is located in an incoming uniform flow with constant velocity; to simplify matters the oscillatory motion was restricted to heave. The numerical experiments were carried out at a higher Reynolds number of 1.0×10^5. Due to the alternate vortex shedding the lift coefficient oscillated, around zero, during the numerical simulation; the amplitude of the lift coefficient oscillation was increased with the cylinder oscillation keeping, however, the mean value almost identically to zero. It is also possible to identify three different types of flow regime as the cylinder oscillation frequency increases. The first type – Type I - was observed for low frequency range of the cylinder oscillation; in this situation the Strouhal number remains almost constant. Type I was followed by an intermediate range of frequency – Type II, the transition regime - where apparently the shedding frequency does not correlate to the frequency of the cylinder oscillation. Finally in Type III – high frequency of cylinder oscillation – the vortex shedding frequency was locked-in with the cylinder oscillation frequency (as will be plotted later in Fig. 2).

While it is evident from a review of the literature that the wake structure is the connection between oscillations and heat transfer, the mechanism of this connection is not understood. In addition, it is not known how the cylinder oscillations determine the wake structure or what other factors, if any, are involved in this process. Pottebaum (2003) presented a series of experiments in order to understand the relationship between wake structure and heat transfer for a transversely oscillating circular cylinder in cross-flow and explored the dynamics of the vortex formation process in the wake. The experiments were carried out in a water tunnel at a Reynolds number of 690. It was found that wake structure and heat transfer both significantly affect one other. The wake mode, a label indicating the number and type of vortices shed in each oscillation period, is directly related to the observed heat transfer enhancement. The cylinder's transverse velocity was shown to influence the heat transfer by affecting the circulation of the wake vortices. For a fixed wake structure, the effectiveness of the wake vortices at enhancing heat transfer depends on their circulation. Also, the cylinder's transverse velocity continually changes the orientation of the wake with respect to the freestream flow, thereby spreading the main source of heat enhancement – the vortices near the cylinder base – over a larger portion of the cylinder surface. Previously observed heat transfer enhancement associated with oscillations at frequencies near the natural shedding frequency and its harmonics were shown to be limited to amplitudes of less than about 0.5 cylinder diameters. A new phenomenon was discovered in which the wake structure switches back and forth between distinct wake modes. Temperature induced variations in the fluid viscosity are believed to be the cause of this mode-switching. It is hypothesized that the viscosity variations change the vorticity and kinetic energy fluxes into the wake, thereby changing the wake mode and the heat transfer coefficient. This discovered underscores the role of viscosity and shear layer fluxes in determining wake mode, potentially leading to improved understanding of wake vortex formation and pinch-off process in general. The heat transfer is also affected by aspect ratio for oscillation conditions characterized by weak synchronization of the wake to the oscillation frequency. Additional research into mode-switching needs to be performed. Identifying the criteria for the occurrence of mode-switching would reveal a great deal about wake formation processes.

On the other hand, there are only a few examples of the simulations of vorticity and heat transport using a particle method. Understanding heat transfer from transversely oscillating circular cylinder in a uniform freestream flow is an important and challenging engineering problem. Vortex-induced vibration is known to occur for long, cylindrical elements in tube-bank heat exchangers. This makes it important to understand how oscillations affect the heat transfer so that equipment can be properly designed. Many areas of fluid mechanics are involved in understanding this type of flow. Convective heat transfer, fluid-structure interactions, separated flows and vortex dynamics are all involved in relating cylinder oscillations to heat transfer.

In a recent paper, Recicar *et al.* (2008) simulate numerically the unsteady flow and heat transfer around an transversely oscillating circular cylinder using a heat particle method (Alcântara Pereira and Hirata, 2003). The two-dimensional aerodynamic characteristics were investigated at a Reynolds number of 1.0×10^5. The main purpose of their numerical study was the investigation of the relationship between near-wake structure and heat transfer for the Type I of flow regime. The generation of vortex due the heat was not analyzed (Ogami, 2001).

The main purpose of present numerical study is the investigation of the relationship between near-wake structure and heat transfer for the type III flow regime (see Fig. 2). The two-dimensional aerodynamic characteristics and heat transfer are investigated at a Reynolds number of 1.0×10^5; due to this fact, even with such a high Reynolds number value, no attempt for turbulence modeling were made once these aspects have a strong three-dimensional component; see Alcântara Pereira *et al.* (2004). In this paper we focus our attention on the case of forced convection heat transfer (Alcântara Pereira and Hirata, 2003).

2. MATHEMATICAL FORMULATION AND FUNDAMENTALS OF VORTEX AND HEAT PARTICLE METHOD

We begin our analysis considering the viscous flow around a heated circular cylinder, immersed in an unbounded region with a uniform flow and freestream speed. We assume the flow to be incompressible and two-dimensional, and the fluid to be newtonian, with constant properties. Figure 1 shows the incident flow, defined by freestream speed U with constant temperature T_∞ and the domain Ω with boundary $S = S_1 \cup S_2$, S_1 being the body surface at constant temperature T_w and S_2 the far away boundary. The cylinder moves to the left with constant velocity; an oscillatory motion with finite amplitude A and constant angular velocity λ is added to body motion. In the Fig. 1 the (x,o,y) is the inertial frame of reference and the (η,O,ξ) is the coordinate system fixed to the cylinder; this coordinate system oscillates around the x-axis as $y_0=A\cos(\lambda t)$, where $\lambda=2\pi f_b$ and f_b is the body oscillation frequency.

The evolution of such a fluid is governed by the following relations for conservation of mass, momentum and energy respectively

$$\mathrm{div}\,\mathbf{u} = 0 \tag{1}$$

$$\frac{\partial \mathbf{u}}{\partial t} + (\mathbf{u} \cdot \nabla)\mathbf{u} + \nabla p = \upsilon \nabla^2 \mathbf{u} + \mathbf{f_c} \tag{2}$$

$$\frac{\partial T}{\partial t} + (\mathbf{u} \cdot \nabla)T = \frac{k}{\rho c_p} \nabla^2 T \,. \tag{3}$$

In the equations above $\mathbf{u}$ is the velocity vector field, p is the pressure, $\mathbf{f_c}$ is body force, υ is the fluid kinematics viscosity coefficient, k is the thermal conductivity and ρc_p is the volumetric heat capacity ($k/\rho c_p$ is the thermal diffusivity). Equation (3) is according Boussinesq's approximation.

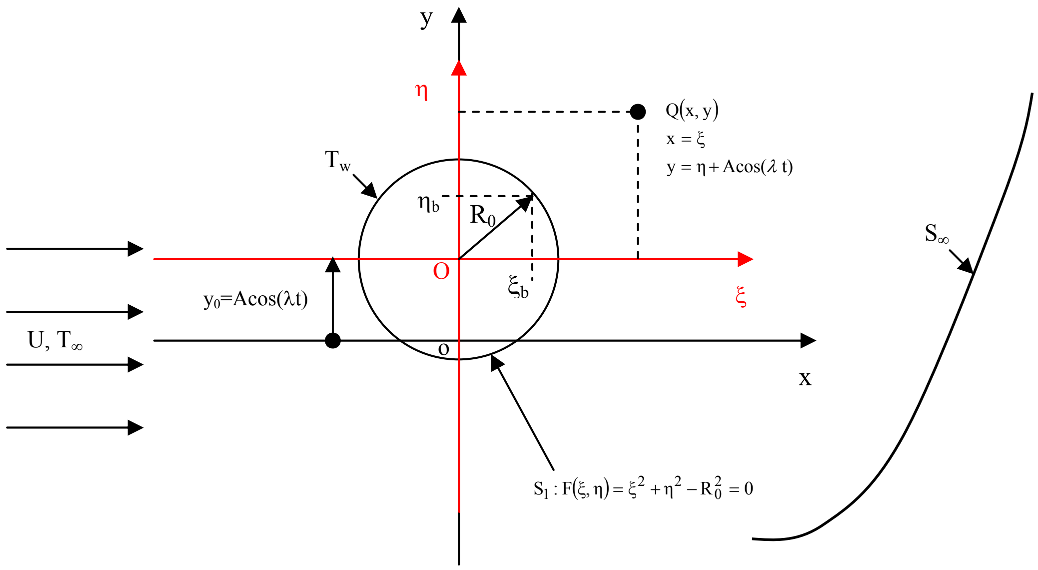


Figure 1. Transversely oscillating heated circular cylinder in uniform freestream flow.

In the present formulation it is necessary to solve for both the pressure, temperature and two velocity components in order to evolve the flow. This can complicate computational approaches and it proves favorable to consider the evolution equation for the curl of the velocity, the vorticity, instead

$$\boldsymbol{\omega} = \nabla \times \mathbf{u} \,. \tag{4}$$

For two-dimensional flow, the vorticity vector has only one component and thus can be expressed as a scalar field ω(x,y).

Taking the curl of Eq. (2) and applying Eq. (1) and the two-dimensionality constraint yields

$$\frac{\partial \omega}{\partial t} + (\mathbf{u} \cdot \nabla)\omega = \upsilon \nabla^2 \omega + \nabla \times f_{\mathbf{c}} \tag{5}$$

which is the evolution of the scalar vorticity field.

In order to determine a flow from the vorticity equation, Eq. (5), it is necessary to find a velocity field in terms of the vorticity field. To do so, consider decomposing the velocity vector into two fields

$$\mathbf{u} = \nabla \times \boldsymbol{\psi} + \nabla \phi \tag{6}$$

where $\boldsymbol{\psi}$ is the vector streamfunction and ϕ is the velocity potential. Now consider the curl of Eq. (6)

$$\boldsymbol{\omega} = \nabla \times \nabla \times \boldsymbol{\psi} + \nabla \times \nabla \phi = -\nabla^2 \boldsymbol{\psi} + \nabla(\nabla \cdot \boldsymbol{\psi}) \tag{7}$$

Note that the velocity potential serves to include components of the velocity field which cannot be represented in the vorticity field. For two-dimensional flow, Eq. (7) simplifies to

$$\omega = -\nabla^2 \psi \ . \tag{8}$$

The Green's function for the two-dimensional Laplacian can be now convolved with the stream-function to give a vorticity-based representation of the streamfunction as follows

$$\psi(\mathbf{x}) = -\frac{1}{2\pi} \iint \ln\left|\mathbf{x} - \mathbf{x}'\right| \omega(\mathbf{x}')d\mathbf{x}' \tag{9}$$

Now Eq. (9) can be used in Eq. (6) to yield the relation commonly known as the Biot-Savart law

$$\mathbf{u}(\mathbf{x}) = -\frac{1}{2\pi} \iint \frac{(\mathbf{x} - \mathbf{x}') \times \omega(\mathbf{x}')\hat{\mathbf{z}}}{\left|\mathbf{x} - \mathbf{x}'\right|^2} d\mathbf{x}' + \nabla\phi \tag{10}$$

The foundation of vortex method rests on the use of Eq. (5) and Eq. (10) to track a fluid based on the evolution of this vorticity field. The vorticity equation is free of the computational instabilities associated with the convective term. In the inviscid case without body forces and frame of reference acceleration

$$\frac{D\omega}{Dt} = 0 \ . \tag{11}$$

Computational simulation requires the discretization in space and time of Eq. (5) and Eq. (10). Particles strengths remain constant to satisfy Eq. (11), so

$$\frac{dx_i}{dt} = u(x_i) \ \text{and} \ \frac{d\Gamma}{dt} = 0 \ , \tag{12}$$

where the amount of vorticity carried by a given particle is termed its circulation and represented by Γ.

The vorticity convection is governed by Eq. (11) and the velocity field is given by (Hirata *et al.*, 2008)

$$\mathbf{u}(\mathbf{x},t) = \mathbf{ui}(\mathbf{x},t) + \mathbf{ub}(\mathbf{x},t) + \mathbf{uv}(\mathbf{x},t) \ . \tag{13}$$

where, $\mathbf{ui} \equiv [1, 0]$ is the velocity vector of uniform flow,

$\mathbf{ub} \equiv [ub^{(i)}, vb^{(i)}]$ is the velocity vector induced by the cylinder at the location of vortex (i),

$\mathbf{uv} \equiv [uv^{(i)}, vv^{(i)}]$ is the velocity vector induced at the vortex (i) due to the vortex cloud.

Once, with the vorticity field the pressure calculation starts with the Bernoulli function, defined by Uhlman (1992) as

$$\overline{Y} = p + \frac{u^2}{2}, \ u = |\mathbf{u}| \ . \tag{14}$$

Kamemoto (1993) used the same function and starting from the Navier-Stokes equations was able to write a Poisson equation for the pressure. This equation was solved using a finite difference scheme. Here the same Poisson equation was derived and its solution was obtained through the following integral formulation (Shintani and Akamatsu, 1994)

$$H\overline{Y}_i - \int_S \overline{Y}\nabla\Xi_i \cdot \mathbf{e}_n dS = \iint_\Omega \nabla\Xi_i \cdot (\mathbf{u} \times \boldsymbol{\omega}) d\Omega - \frac{1}{Re}\int_S (\nabla\Xi_i \times \boldsymbol{\omega}) \cdot \mathbf{e}_n dS \tag{15}$$

where $H = 1$ in the fluid domain, $H = 0.5$ on the boundaries, Ξ is a fundamental solution of the Laplace equation and $\mathbf{e}_n$ is the unit vector normal to the solid surfaces.

The drag and lift coefficients are expressed by

$$C_D = -\sum_{k=1}^{NP} 2(p_k - p_\infty)\Delta S_k \sin\beta_k = -\sum_{k=1}^{NP} C_P \Delta S_k \sin\beta_k \qquad (16)$$

$$C_L = -\sum_{k=1}^{NP} 2(p_k - p_\infty)\Delta S_k \cos\beta_k = -\sum_{k=1}^{NP} C_P \Delta S_k \cos\beta_k \qquad (17)$$

where NP is the total number of flat source panels representing cylinder surface (Katz and Plotkin, 1991). It is assumed that the source strength per length is constant and ΔS_k is the length and β_k is the angle and both of the k-panel.

Note that Eq. (3) gives the law that the temperature distribution, T, moves both with the convection velocity. Vortex elements and discrete heat elements distributed in the flow field are followed during numerical simulation according to the first order Euler scheme. It is clear that the energy equation, Eq. (3), has the similar form to the vorticity transport equation, Eq. (5). This suggests that the energy equation can be solved in an analogous way using the random walk method to the motion of the heat elements to account for diffusion (Chorin, 1973).

In this paper, the temperature T_w is considered constant around the body surface, see Fig. 1. The heat transport from the body surface to the fluid nearby the body surface is determined by the temperature gradient at the surface. The surface heat flux is determined by Fourier's Law

$$\dot{q} = -\lambda \frac{dT}{dn} \qquad (18)$$

where n denotes the normal direction to the surface and λ is the thermal conductivity of fluid. The heat quantity transferred from the surface (j-th panel with length ΔS_j) to the k-th nascent heat element is given by

$$\Delta Q_j = \alpha \Delta t \frac{(T_w - T_j)}{\varepsilon} \Delta S_j \qquad (19)$$

in which $\alpha = \upsilon / Pr$ (Pr is Prandtl number) and ε is the displacement normal to the straight-line panel.

The temperature distribution $T(z)$ results from the contribution of all the heat particles in the field

$$T(z) = \sum_j \frac{\Delta Q_j}{\pi \sigma_T^2} \exp\left[-\frac{(z - z_j)^2}{\sigma_T^2} \right] \qquad (20)$$

where σ_T is the core radius of the heat particles.

3. RESULTS AND DISCUSSION

We preliminary investigate the flow around a fixed circular cylinder to analyze the consistence of the vortex code and to define some numerical parameters; as for example the number of panels used to define the cylinder surface. For this particular configuration, the cylinder surface was represented by NP=100 flat source panels with constant density. The simulation was performed up to 800 time steps with magnitude Δt=0.05 (Mustto *et al*., 1998). During each time step the new vortex elements are shedding into the cloud through a displacement $\varepsilon^* = \sigma_0 = 0.0032d$ normal to the straight-line elements (panels); see Ricci (2002).

Table 1. Mean drag and lift coefficients and Strouhal number for fixed circular cylinder

$Re = 1.0 \times 10^5$	$\overline{C_D}$	$\overline{C_L}$	$\overline{St}$
Blevins (1984)	1.20	-	0.19
Mustto *et al.* (1998)	1.22	-	0.22
Present Simulation	1.22	0.07	0.20

Table 1 shows that the numerical results agree very well with the experimental ones obtained by Blevins (1984), which have an uncertainty of about 10%. The results from Mustto *et al*. (1998) were obtained numerically using a slightly different vortex method from the present implementation. The agreement between the two numerical methods is very good for the Strouhal number, and both results are close to the experimental value. The present drag coefficient shows a higher value as compared to the experimental result. One should observe, that the three-dimensional effects are non-negligible for the Reynolds number used in the present simulation ($Re = 1.0\times10^5$). Here the Reynolds number is defined as $Re = \frac{U\,d}{\upsilon}$, where $d=2R_0$ is the diameter cylinder; the dimensionless time is d/U.

Therefore one can expect that a two-dimensional computation of such a flow must produce higher values for the drag coefficient. On the other hand, the Strouhal number is insensitive to these three-dimensional effects. The mean numerical lift coefficient, although very small, is not zero which is due to numerical approximations. The aerodynamic forces computations were evaluated between t=20 and t=40.

The Strouhal number is defined as

$$St = \frac{f\,d}{U} \tag{21}$$

where f is the detachment frequency of vortices of the lift coefficient. In general, one should observe that the lift coefficient oscillates with a dimensionless frequency (Strouhal number) that is one half the frequency of oscillation of the drag coefficient curve. More details of this preliminary study are discussed in Hirata *et al*. (2008).

In this paper, the body Strouhal number is defined as

$$St_b = \frac{f_b d}{U}\,. \tag{22}$$

Figure 2 shows our plots of the Strouhal number as a function of the body Strouhal number for higher oscillatory motion amplitude. The analysis of this figure shows that, in general, the high Reynolds number simulations agree quite well with the low Reynolds number vortex synchronization regions devised by Williamson and Roshko (1988), except that the synchronization begins at a lower value of St_b which is quite reasonable due to the increasing importance of the inertial effects as the Reynolds number assume higher values.

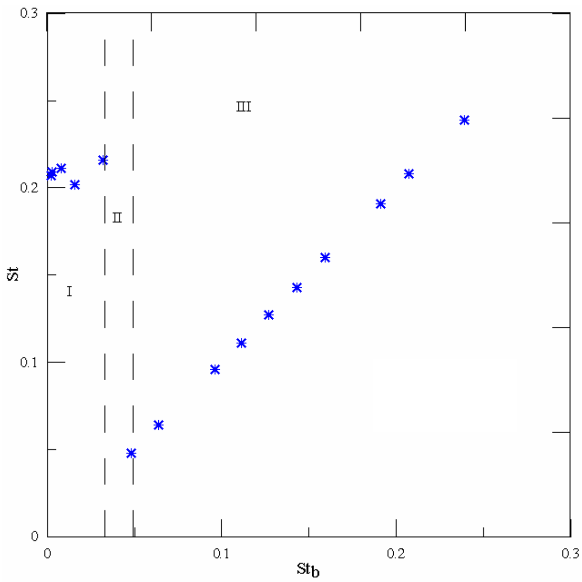
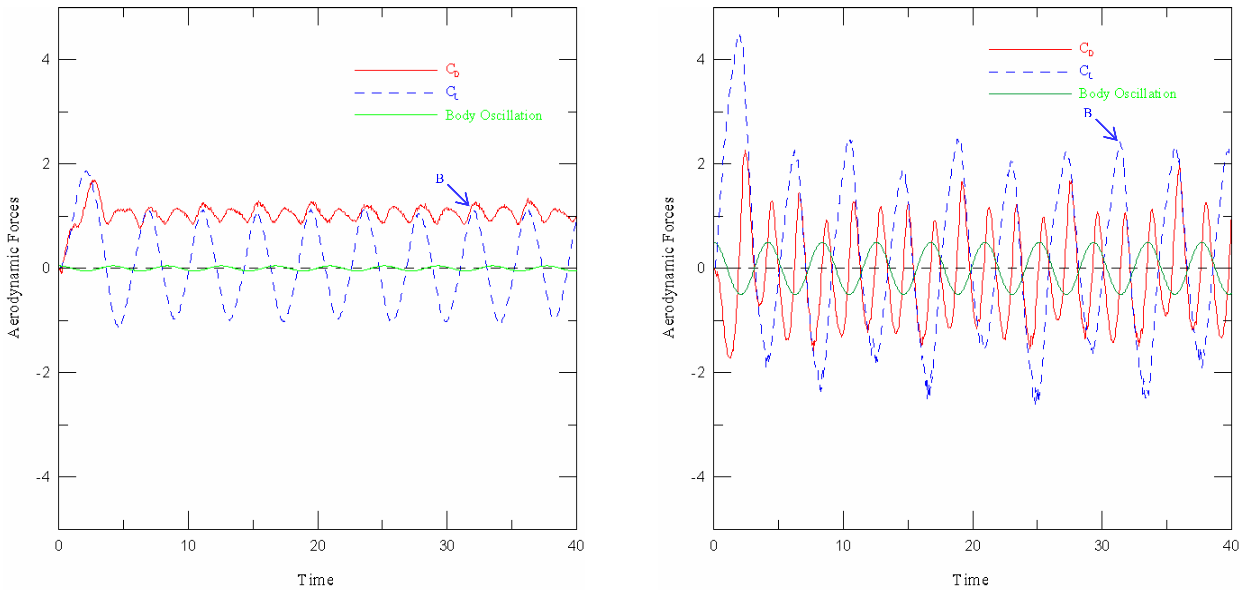


Figure 2. Circular cylinder: frequency of vortex shedding, A/d = 0.50 and Re=1.0x10^5.

When the vortex method is applied to heat-fluid motion, it becomes evident that the procedure is very sensitive to the numerical parameters involved. The main influences are: the non-dimensional time step; distance of release particles from surface; particles blob radius and time increment. Our numerical simulations for flow and heat transfer was performed around a circular cylinder at Re= 1.0×10^5 and Pr= 0.71. A constant temperature T_w= 363 K on the body surface as a boundary condition, and the freestream temperature was set to T_∞= 293 K. The core radius for nascent heat elements was $\varepsilon=\sigma_T$ =0.1d (Alcântara Pereira & Hirata, 2003).

Figure 3 shows time histories of drag coefficient C_D and lift coefficient C_L of transversely oscillating heated circular cylinder at high body oscillation frequency (λ=1.5). Two cylinder configurations are presented: small amplitude (A/d=0.05) and higher amplitude (A/d=0.5). In both configurations, vortex-shedding frequency f had been found to lock on to the forcing frequency f_b when f_b is close to the free vortex shedding frequency f_{so} in the transverse oscillating case.

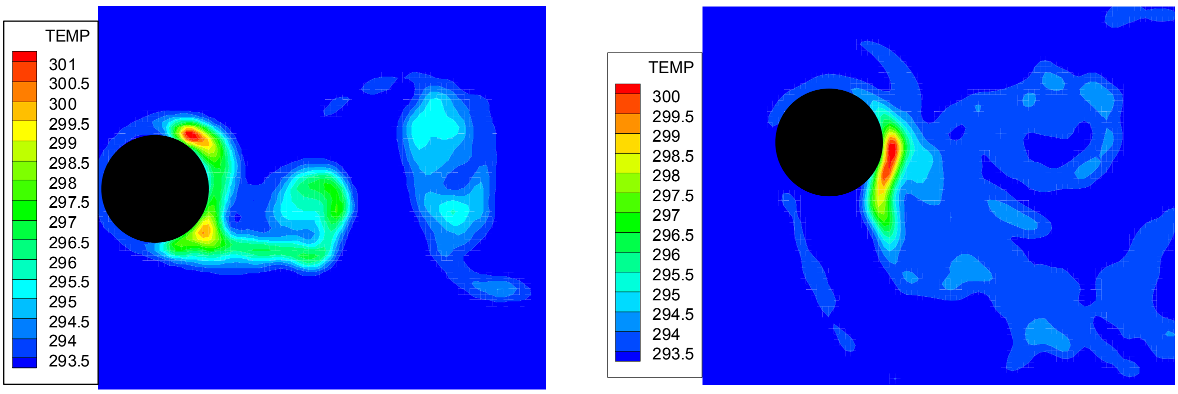
As illustration, consider in Fig. 3(a) and Fig. 3(b) the instant B defined by a maximum value of the lift coefficient; at this moment a large clockwise vortex structure (in fact a cluster of vortices) is detaching from the upper surface and moving toward the viscous wake; the near field for temperature distributions are indicated in Fig. 4. As this structure moves downstream it pushes away an anti-clockwise structure that was stationed behind the cylinder and the drag coefficient increases. For the transversely oscillation case with higher amplitude (A/d=0.5), the temperature distribution in the vicinity of a body give us information of higher transport of heat in the near wake, see Fig. 4(b). Pottebaum (2003) founded that wake structure and heat transfer both significantly affect one another. The wake mode, a label indicating the number and type of vortices shed in each oscillation period, is directly related to the observed heat transfer enhancement. We can observe this situation in Fig. 5 that shows the vorticity field for the instant B. The dynamics of the vortex formation process, including the trajectories of the vortices during roll-up, explain this relationship.



(a) A/d=0.05 (b) A/d=0.5

Figure 3. Variation of C_D and C_L with time, λ=1.5 and Re=1.0x10^5.

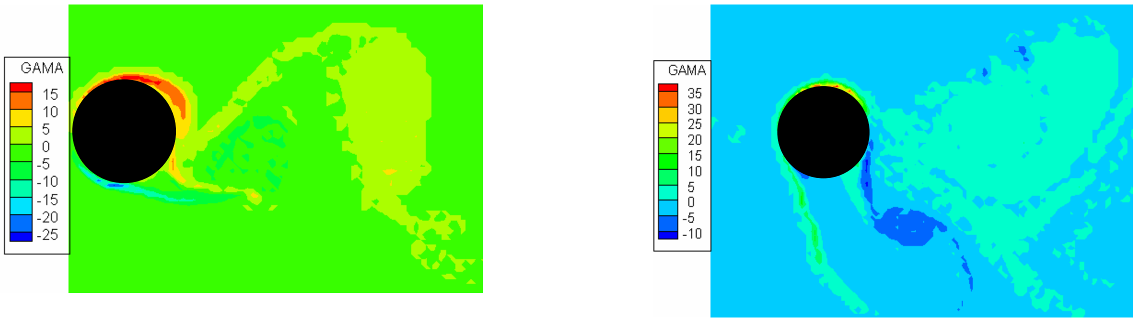
According to the Pottebaum (2003), the cylinder's transverse velocity was shown to influence the heat transfer by affecting the circulation of the wake vortices.



(a) t=31.95, A=0.05 and λ=1.5 (b) t=31.4, A=0.5 and λ=1.5

Figure 4. Temperature distributions at B point for transverse oscillating circular cylinder case, Re=1.0x10^5 and Pr=0.71.

For a fixed wake structure, the effectiveness of the wake vortices at enhancing heat transfer depends on their circulation. Also, the cylinder's transverse velocity continually changes the orientation of the wake with respect to the freestream flow, thereby spreading the main source of heat transfer enhancement – the vortices near the cylinders base – over a large portion of the cylinder surface.



(a) t=31.95, A=0.05 and λ=1.5

(b) t=31.4, A=0.5 and λ=1.5

Figure 5. Vorticity field at B point for transverse oscillating circular cylinder case, Re=1.0×10^5 and Pr=0.71.

Figure 6 shows the temperature distributions for fixed case at instant represented by high pressure distribution on the rear part of the cylinder when the drag curve assumes a maximum value.

Our simulation for the fixed cylinder case provided a very estimate of the temperature when compared to the transverse oscillating circular cylinder case. We conclude that besides the Reynolds number, two additional aspects must be considered for future analysis: the influence of Prandtl number and the buoyancy effect.

The present methodology, therefore, is able to provide good estimates for Strouhal number, lock-in, lift and drag coefficients and pressure distribution. Future analysis will be carried out to compute time-averaged Nusselt number (Alcântara Pereira and Hirata, 2003) to predict the fluid flow and heat transfer correctly in a physical sense.

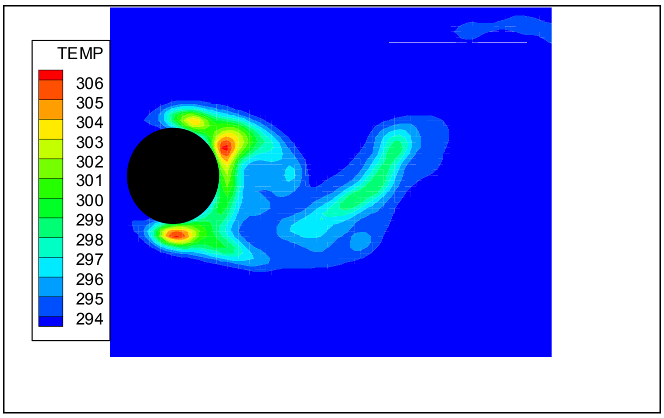


Figure 6. Temperature distributions at B point for fixed circular cylinder case, Re=1.0×10^5 and Pr=0.71.

Finally, numerical studies on the flow around an in-line vibration cylinder or transversely vibration cylinder had mainly been restricted to small Reynolds number. It can still be said that the existing experimental work with higher Reynolds number is very difficult to find.

Our computational code provides us of opportunity to handle fluid flow and heat transfer for regimes of vortex shedding from an oscillating body.

4. CONCLUSIONS

In order to handle fluid flow and heat transfer by the vortex method, the following models are required (Ogami, 2001): (i) the modeling of discretization of heat distribution into the heat particles; (ii) the modeling of the process in which the vortex is generated by the effect of a heat; (iii) the modeling of the diffusion process of heat and vortex.

This work demonstrated a heat particle method, with a wholly mesh-less formulation. Applications have been demonstrated in problems of vortex-heat interaction on the lock-in phenomenon of Kármán vortex shedding on the circular cylinder, with encouraging results.

It is of practical importance to investigate the Kármán vortex excitation, because it is a non-linear self-excitation oscillation caused by the interaction between the body motion and the flow around it. Of particular interest here is to include heat transfer on the phenomenon of synchronization, in which the frequency of vortex shedding coincides with the cylinder oscillation.

The main objective of the work with the implementation of a vortex and heat particles method for the analysis of unsteady and forced-convective heat transfer in a flow around a stationary and oscillating body has been achieved.

In the present paper, vorticity is not created from heat and as a future work the effect of buoyancy will be carried out. Ogami (2001) presented two models for creating vortices from temperature particles. First, he modeled the vorticity equation as it is, and it is regarded as a natural extension of the method of Ghoniem and Sherman (1985). Second, a vortex pair (one positive and one negative) was generated from one temperature particle.

Our new scheme involving natural convection and interaction of temperature and vorticity will be modeled by creating vorticity from temperature by changing the strength of vortices, according to the vorticity equation. The vorticity field showed in Fig. 5, therefore, will be modified by presence of the temperature particles. A new method to simulated diffusion will be carried out (Rossi, 2006).

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