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Numerical Investigation of Convective Heat Transfer from a Horizontal Plate Due to the Oscillation of a Vertically Oriented Blade

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ABSTRACT

Convective heat transfer from a flat plate, which is enhanced by an oscillating blade, was numerically investigated at the present study. It was assumed that the blade is made of a rigid and thin plate and it is vertically oriented at the top of the target plate. Numerical analysis was performed using commercial software ANSYS Fluent 6.3 and the periodic oscillation of the blade was modeled by the moving mesh method. Conservation equations of mass, momentum and energy was solved in 2-D and transient form for the laminar airflow with constant physical properties. Constant temperature was considered for the plate and the details of both the flow and thermal fields were determined. The distribution of convective heat transfer coefficient was then calculated for the target plate. The effect of various parameters including the amplitude and frequency of the blade oscillation as well as the geometrical parameters was investigated on the convective heat transfer from the target plate. The results indicated that a wider area of the plate was affected by increasing the oscillation amplitude of the blade. Convective heat transfer was also enhanced over the entire target plate as the rotational Reynolds number was increased.

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1. Introduction

Temperature difference potential transfers heat between solid surfaces and adjacent fluid in many engineering applications and industrial equipments. In most of the applications such as electronic cooling and heat exchangers, convective heat transfer enhancement is desirable. Effort is made to perform heat transfer enhancement with less energy consumption also in a simple and efficient manner. The enhancement techniques are categorized as passive and active methods where the former one consumes no external energy. Due to the variety both in applications and in the improvement techniques, convective heat transfer enhancement is still an open and important research area in the field of heat transfer research.

Passive techniques in heat transfer augmentation have been employed in various manners. Both extended surfaces and surfaces with artificial roughness have generally been used to enhance convective heat transfer in many investigations, based on which several review papers have also been published [1, 2]. Sheikholeslami et al. [3] provided another review of heat transfer enhancement in passive methods focusing on swirl flow devices. The effect of various turbulators including; coiled and corrugated tubes, twisted tape, conical ring and coiled wire on heat transfer augmentation has briefly been discussed in that review paper. Shi et al. [4] numerically investigated the effect of vortex-induced vibration in disrupting thermal boundary layer and hence increasing heat transfer rate. A cylinder with a flexible plate at the back was used as the vortex-generating device inside a plane duct in their research.

Moving a cylinder within a two-dimensional channel has been used as an active method to increase convective heat transfer from the channel walls in

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several studies. Oscillatory flows are known to cause higher heat and mass transfer. Oscillating movement is achieved either by the fluid vibration around a fixed object or by vibration of a solid body within a fluid. Although fluid vibration around a fixed object requires more energy, the same goal is achieved in both approaches. Oscillating flows are widely used in compact high-performance heat exchangers, piston engines, chemical reactors, pulsating burners, high-performance Stirling engines, cryogenic refrigeration, and in various applications in the aerospace industry and military fields [5–9]

Celik et al. [10] numerically investigated heat transfer enhancement in a plane duct by vortices shed from a transversely oscillating circular cylinder. They found that the placement of a transversely oscillating cylinder enhances convective heat transfer within thermally developing flow region considerably. In a similar study, Fu and Tong [11] numerically examined the influence of a transversely oscillating cylinder on the heat transfer from heated blocks in a plane duct flow. Beskok et al. [12] numerically investigated convective heat transfer from heated walls of a plane duct in presence of a rotationally oscillating cylinder. Pourgholam et al. [13] conducted a numerical research to investigate the effect of a rotating and oscillating blade on heat transfer enhancement from the channel walls. Jahangiri and Delbari [14] numerically studied the details of the flow and heat transfer in a mixing tank equipped with a helical single-blade mixer. They suggested a heat transfer correlation for two-phase flow developed within the mixing tank.

Convective heat transfer enhancement by the movement of a solid body as a heat source is another branch of the active methods, which has been considered in several studies. Rahman and Tafti [15] numerically investigated convection heat transfer augmentation in a system composed of an infinitesimally thin plate-fin with forced oscillation in the presence of an approaching flow. They found that combined effect of the oscillation frequency and amplitude on heat transfer augmentation could be represented by a single parameter called ‘plunge velocity’. Rahimi and Soran [16] numerically investigated the effect of a plate movement on the heat transfer distribution over the plate impinging by an air jet. Sarhan [17] experimentally investigated the vibration effect of a rectangular flat plate on convective heat transfer from the plate in both horizontal and inclined positions. Gomaa and Al Taweel [18] analytically investigated the effect of oscillatory motion on heat transfer from a vertical surface. They concluded that higher temperature gradient in the axial direction causes to much higher heat transfer in the presence of an oscillatory motion. Akcay et al. [19] experimentally investigated the convection heat transfer from an oscillating vertical plate. They found that the heat transfer performance increases with the increase of both oscillation frequency and amplitude.

Piezoelectric fan has also been used in convective heat transfer augmentation from solid surfaces. Chen et al. [20] experimented the flapping dynamics of a piezoelectric fan and heat transfer enhancement from the wall in a turbulent channel flow. In a similar study, Li et al. [21] studied the channel flow structure and heat transfer enhancement by a vertically oriented piezoelectric fan. They concluded that the presence of larger channel flow velocities has significant influence on the fan vibration amplitude and hence on heat transfer augmentation. Ebrahimi et al. [22] conducted an experimental and modeling study to investigate power dissipation when using piezoelectric fan. Li et al. [23] experimentally investigated the effect of blade shape and geometry on convective heat transfer induced by a piezoelectric vibrating fan. More studies concerning the flow structure and heat transfer enhancement induced by piezoelectric fans are found in several review papers [24, 25].

Convective heat transfer enhancement from large horizontal surfaces to the surrounding air is another challenging problem. This phenomenon is encountered in several practical applications such as; textiles, glass and paper production and tempering of metallic plates. Heat transfer by impinging jets as an effective approach has widely been used in such applications. Augmentation of heat transfer by a vertically oriented vibrating plate over the target surface could be an efficient and alternative method. While this approach has been employed traditionally as a hand fan, but no published study was found in the literature concerning the specifications of this approach. It is worth noting that this heat transfer mechanism is geometrically quite similar to the piezoelectric fan. However, it can be employed in larger scales and with lower frequencies. Therefore, the details of the developed flow field and heat transfer augmentation from a flat plate caused by a vertically oriented oscillating blade are investigated at the present study.

2. Problem description and the numerical approach

In order to investigate convective heat transfer from a flat plate due to the oscillation of a vertical blade, a solid surface with 0.084 m length was considered horizontally. Stagnant air at 100 kPa and 300 K was assumed as the surrounding fluid. Assuming 0.084 m height over the solid surface, a square computational domain was formed for the numerical analysis. A thin rigid blade having 0.038 m length and 1 mm thickness was supposed to be vertically pivoted at the center of the domain, so that the lower end of the blade was 4 mm away from the horizontal plate. The period and amplitude of the oscillating blade represented by τ and

θ_{max}, respectively. Schematic from the considered domain is shown in Fig. 1.

The flat plate with no slip velocity components and constant temperature of 320 K was the lower boundary condition. Boundary condition for the three remaining boundaries was zero gauge pressure at the outlet of the boundaries. The boundary conditions for the blade surfaces were no-slip velocity components with adiabatic boundary condition. In order to model the oscillation of the blade, a concentric circular section with 0.04 m radius was specified within the computational domain. This subdomain including the blade geometry was defined as the moving (oscillating) section in the numerical analysis. The circle connecting two subdomains was the interface boundary condition in the computations. Specified boundary conditions were presented in equation form at the end of the governing equations.

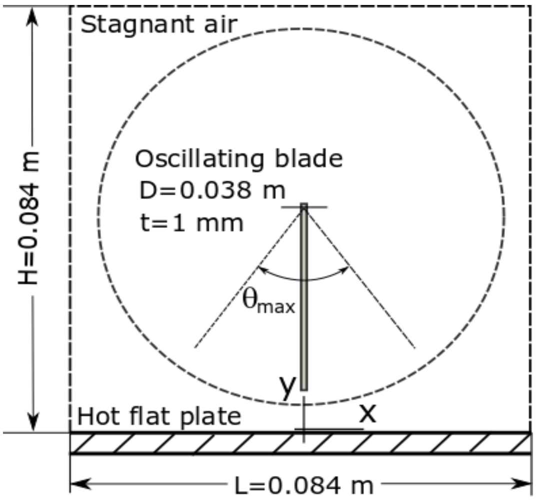


Figure 1. Schematic of the computational domain

Conservation equations of mass, momentum and energy in two dimensional and transient form were solved numerically. For a fluid with constant physical properties and in the case of laminar flow, these equations can be written as:

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

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\nu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\nu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \tag{3}$$

$$\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

$$x=\pm\frac{L}{2}\rightarrow\frac{\partial p}{\partial x}=0,$$
$$T=T_\infty\ (for\ the\ backflow\ air)$$

$$y=H\rightarrow\frac{\partial p}{\partial y}=0,\qquad y=0\rightarrow u=v=0,\quad T=T_w \tag{5}$$

$$at\ the\ balde\ surface:\quad u=v=0,\ \frac{\partial T}{\partial n_1}=0$$

$$at\ the\ interface:\ \frac{\partial u}{\partial n_2}=\frac{\partial v}{\partial n_2}=\frac{\partial T}{\partial n_2}=\frac{\partial p}{\partial n_2}=0$$

where n_1 and n_2 are unit vectors perpendicular to the blade and the interface surface, respectively.

Numerical analysis was performed using ANSYS Fluent 6.3 commercial software. Velocity and temperature gradients were evaluated using Green-Gauss cell-based procedure. Pressure-based solver with second-order implicit formulation were used in the analysis. Pressure-velocity coupling was accomplished based on SIMPLE algorithm and the residuals were less than 10^{-3} in the conversation equations as the convergence criteria.

The oscillating motion of the blade was represented by the following sinusoidal equation:

$$\theta(t)=\theta_{max}sin\left(\frac{2\pi}{\tau}t\right) \tag{6}$$

Based on which, the angular velocity of the blade was:

$$\omega(t)=\frac{2\pi}{\tau}\theta_{max}cos\left(\frac{2\pi}{\tau}t\right) \tag{7}$$

The blade velocity and its angular position are shown in Fig. 2 for a period with periodicity and amplitude of τ = 0.1 Sec and θ_{max} = 30°, respectively. Discretized values of the angular velocity are also shown in the figure that is required in the numerical calculations. Time increment value is 0.05τ at this sample representation. Discretized angular velocity values were sequentially applied in the transient numerical analysis as the blade rotational velocity over the specified time interval.

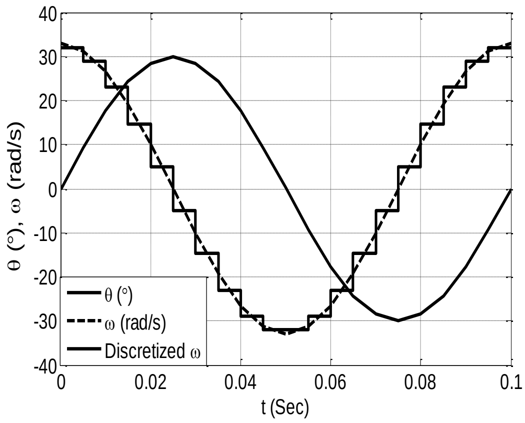


Figure 2. Variations of the angular velocity and position of the oscillating blade

3. Grid independency, time step sufficiency and the result validation

In order to use a reasonable mesh size in the numerical analysis, a systematic grid independency procedure was accomplished. While the first mesh line was 0.25 mm away from the plate surface, the mesh size was increased in the normal direction by a factor of 1.2. After several layers with increasing dimensions, mesh with constant size of almost equal to that of the last layer was generated in the remaining part of the domain. Fig. 3 indicates general characteristics of the generated mesh, with quite fewer points for the sake of simplicity.

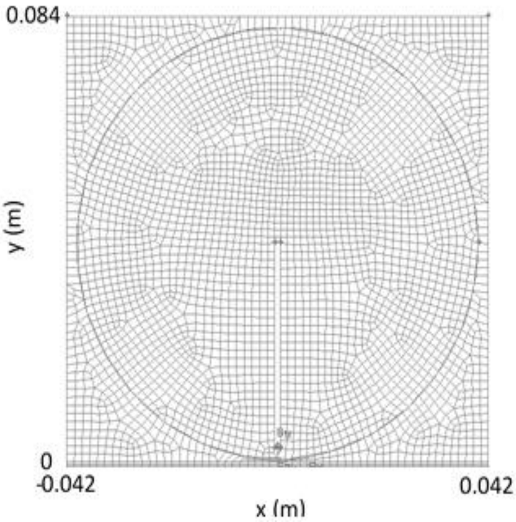


Figure 3. Schematic of generated grid with a fewer grid points

Several values were examined for the interior mesh size and each time computational analysis was conducted to obtain the flow field and the temperature distribution. Convective heat transfer coefficient over the flat plate was calculated using the temperature distribution. These results which are shown in Fig. 4 are related to the end of the fourth cycle of oscillation and when the blade is moving to the right from its upright position.

Based on the results, mesh size of 1.5 mm for the interior section of the domain predicts smaller average heat transfer coefficient. It slightly increases when using mesh size of 1 mm for the main part of the domain. However, very close values are obtained when using the mesh sizes of 0.5 mm and 0.25 mm for the main section of the computational domain. This is while, the number of grid points are approximately four times greater in the latter case.

Therefore, boundary layer mesh with 0.25 mm size at the vicinity of the plate and mesh size of 0.5 mm for the interior section of the domain were used in the main analysis. It should be mentioned that no significant change observed in the results by further reduction in the boundary layer mesh size.

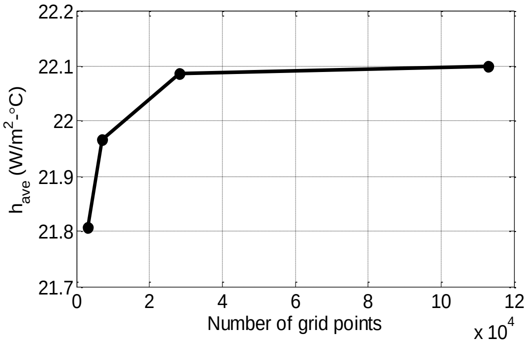


Figure 4. Average convective heat transfer coefficient over the plate for different grids, θ_{max}= π/6, Re$_D$ =6240, d/D = 0.105

As indicated in Fig. 2 and based on the selected time interval, discretized angular velocity values were employed in the analysis. To specify a suitable size for the time step, the numerical analysis was conducted for several values such as; 0.1τ, 0.05τ, 0.025τ and 0.0125τ. The variation of the convective heat transfer coefficient over the plate was obtained for each time step size as shown in Fig. 5. This figure indicates that the results are significantly affected by the selected time step size. The largest time step size being 0.1τ, predicts a very different heat transfer distribution over the plate. Time step size being equal to five percent of the periodicity, which was also used in the evaluation of the appropriate mesh size, predicts a different result compared to the previous one. However, consistent results are obtained for the two smaller time step sizes so that very negligible differences exist between these two heat transfer distributions. Therefore, time step size of two and half percent of the time-period was suitable to be used in the main numerical analysis.

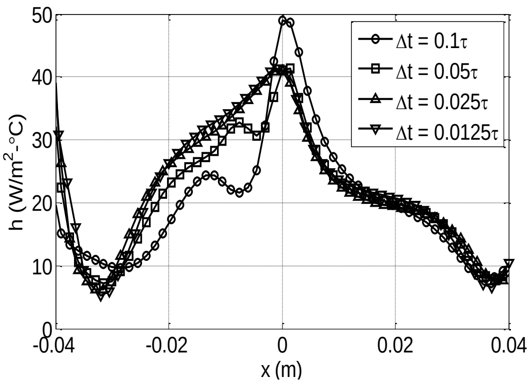


Figure 5. Convective heat transfer coefficient distributions using different time steps, θ_{max} = π/6, Re$_D$ =6240, d/D = 0.105, τ = 0.1 Sec.

At the present transient analysis, the initial condition was zero velocity at all grid points and the flow field was gradually developed by the blade oscillation. Based on the results and due to the depreciation of the initial conditions, a periodic flow field was established within the domain after about four periods of the blade oscillation. Also, no significant change was detected in convective heat transfer distribution over the plate when its length was further increased. Except that the sharp rise in the convective heat transfer coefficient occurring at the left end of the plate was shifted again to a similar position. This rise was essentially the direct effect of the backflow entering into the domain from the left boundary. Also, by increasing the plate length, the number of oscillation periods required for the depreciation of the initial condition was slightly increased.

In order to evaluate the accuracy of the numerical analysis employed at the present study, no similar study was found in the published literature. Therefore, some of the results from the study of Pourgholam et al. [13] were selected for comparison. That study concerns to a laminar and steady flow between two parallel plates in which the effect of a rotating blade positioned at the center of the channel on the convective heat transfer from the walls is examined. Reynolds number of the undisturbed flow was 50 based on the hydraulic diameter of the channel. The length of the thin blade was one-sixth of the channel hydraulic diameter and its rotational velocity was 300 rpm. Assuming thermal conductivity of air constant and equal to 0.024 W/m-K and based on the channel hydraulic diameter, Nusselt number was defined using convective heat transfer coefficient. The present numerical method was employed to obtain similar results under quite the same circumstances. The results obtained from both the studies are presented in Fig. 6, which shows a reasonable consistency between these results. Therefore, the accuracy of the present numerical method could be reliable.

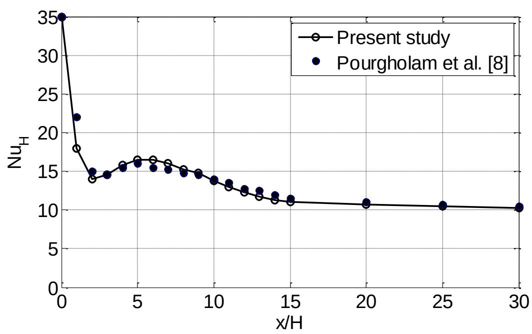


Figure 6. Comparison of the Nusselt number distributions.

4. Results and discussion

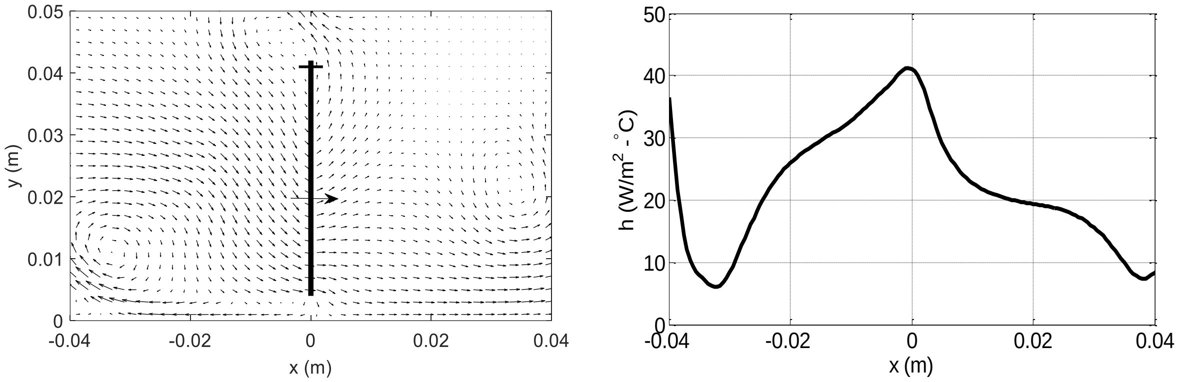
Convective heat transfer from the plate under the effect of the oscillating blade has various distributions at different instances of the blade oscillation period. In order to have a sense of the transient heat transfer distribution, the details of the flow field could also be useful. Therefore, the flow field and convective heat transfer over the plate is considered in details at the present section. Then, the effect of different parameters on the average heat transfer distribution is discussed.

4.1. Transient flow field and heat transfer distribution

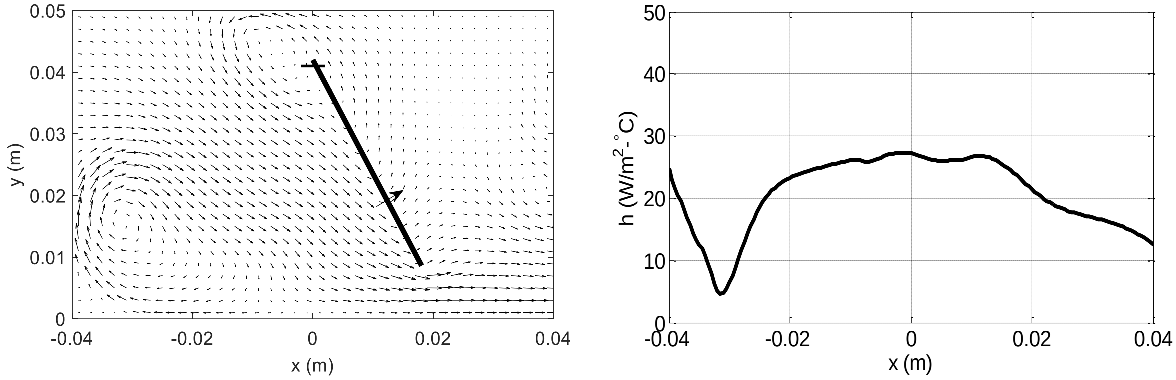
As mentioned in the preceding section, the effect of the initial conditions depreciates after a few periods of oscillation and a periodic flow field is established within the computational domain. The velocity vectors at a very limited number of points from the domain and at different time instants of a half period of the blade oscillation are shown in Fig. 7. According to this figure, the time instant in which the blade crosses the upright position in its counter-clockwise rotation is considered as the starting point of the period and it is denoted by $t = 0$. At this time instant, a large amount of fluid moves behind of the blade meanwhile, a high-speed stream forms in the opposite direction at the vicinity of the plate. As a result of these two streams with opposite directions a large circulating region is formed at the left and lower corner of the domain. At the other side and in front of the blade a moderate flow moves parallel to the plate and exits from the right outlet boundary. Angular velocity of the blade decreases as it rotates in counter-clockwise direction.

As a result, the flow moving behind of the blade tends to pass through the gap formed between the tip of the blade and the target plate. The flow passing through this gap intensifies as the blade decelerates and finally stops at the end of its counter-clockwise rotation. At time instant of 0.4τ when the blade rotates in clockwise direction, consistent flows develop both at the back and front of the blade. Meanwhile, the opposite flow passing through the gap intensifies progressively as the gap size is decreased. Finally, at time instant of 0.5τ the blade is again at upright position but in its clockwise rotation and the established flow field is inversely the same with that of the starting point.

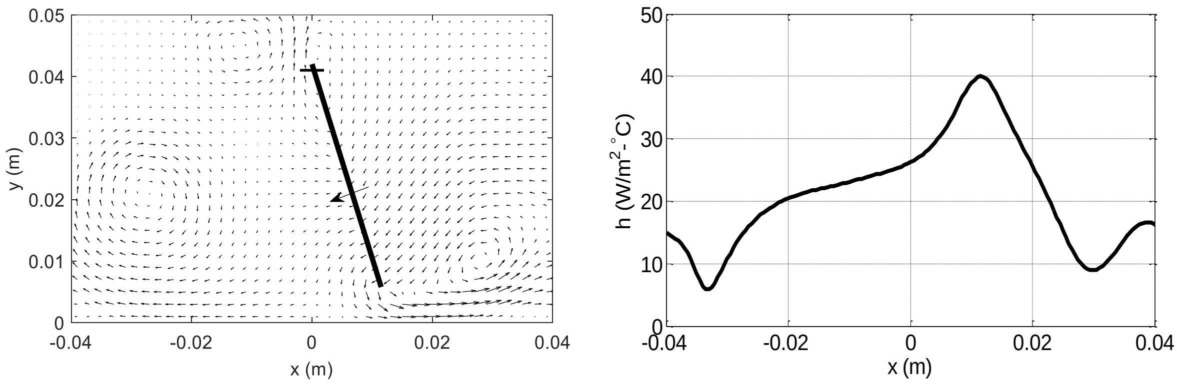
Consistent with the described flow field, convective heat transfer from the plate at different time instants is also shown in Fig. 7. This figure indicates that the convection is initially higher over the central region especially at the backside of the blade at time instant of $t = 0$. A sharp rise is also seen in the heat transfer distribution at the left end of the plate. This rise is due to the circulating flow pattern existing at that region and hence the entering flow into the domain. Convection intensity over the central region decreases rapidly as the blade rotates in counterclockwise direction and keeps out from this region. A relatively uniform distribution is seen for the convection over the central area when the blade is far from its upright position. At time instant of t = 0.4τ when the blade approaches to the central position in its clockwise rotation, convection rate intensifies again over the central region of the plate and at the backside of the blade. Finally, heat transfer distribution which is inversely the same with that of $t = 0$ is obtained for time instant of 0.5τ.



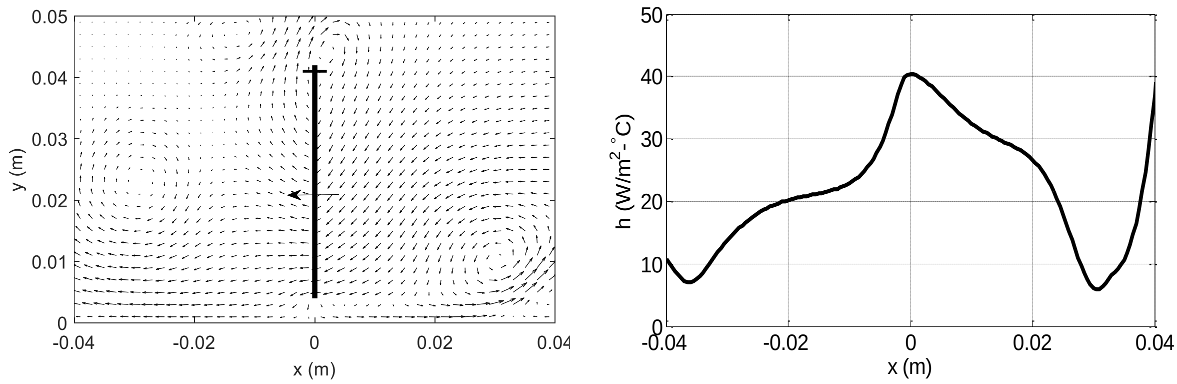
a) t = 0.00



b) t = 0.2τ



c) t = 0.4τ



d) t = 0.5τ

Figure 7. Velocity vectors and convective heat transfer coefficient at some points and different time instants of a half period of oscillation, θ_{max} = π/6, Re_D =6240, d/D = 0.105, τ = 0.1 Sec.

4.2. Effect of different parameters on convective heat transfer distribution

Convective heat transfer distribution induced by an oscillating vertical blade depends on various parameters. Heat transfer coefficient can be regarded as a function of several important parameters such as;

$$h = f(v, k, D, d, x, \theta_{max}, \omega) \quad (8)$$

This expression may be rewritten using dimensional analysis in the following form

$$Nu_D = f'\left(Re_D, \frac{d}{D}, \frac{x}{D}, \theta_{max}\right) \quad (9)$$

where Nu_D and Re_D is the dimensionless Nusselt and Reynolds numbers which are defined by the following expressions:

$$Nu_D = \frac{hD}{k}, Re_D = \frac{D^2\omega}{v} \quad (10)$$

As seen from Fig. 7, convective heat transfer coefficient has various distributions at different time instants. Average convection rate over a full cycle of oscillation in addition to the transient distribution could be practically meaningful and reflects heat transfer augmentation from the target plate. Therefore, average convective heat transfer coefficient was calculated and used in the presenting of the main results.

Nusselt number calculated based on the average heat transfer coefficient had generally a bell-shaped distribution over the target plate. Due to the recirculating flow pattern and also incoming flow from both the right and left boundaries of the domain, the minimum of the bell-shaped distribution is followed by an increase in the Nusselt number distribution. This rising effect which was also symmetrical at both ends, moved away from the central bell-shaped distribution when the length of the plate was further increased.

Fig. 8 indicates the effect of oscillation amplitude on the Nusselt number distribution. Based on this figure, Nusselt number is higher only over a limited section of the central part of the plate for the smallest oscillation amplitude. Average Nusselt number is significantly increased at both sides of the centerline by increasing the oscillation amplitude, while it rises slightly over the central section. Further increase in the oscillation amplitude gives rise to a uniform distribution of the Nusselt number over a large central area of the plate.

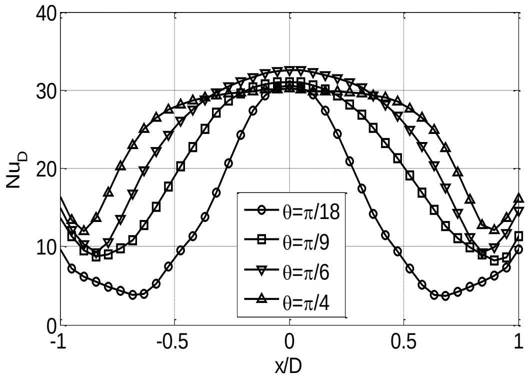


Figure 8. Average Nusselt number distribution, d/D = 0.105, Re_D = 3120

The effect of the rotational Reynolds number on the Nusselt number distribution is shown in Fig. 9. This figure indicates that the Nusselt number increases continuously over the entire plate as the Reynolds number is increased where, the increment is more pronounced over the central region. For air as the surrounding fluid with constant physical properties, rotational Reynolds number is evaluated based on the length and the rotational velocity of the oscillating blade according to Eq. 10. Increasing each of these two parameters results in a higher rotational Reynolds number. Existence of a higher rotational Reynolds number means that a strong flow field is being developed within the domain which results in a higher convective coefficient and hence Nusselt number distribution over the target plate.

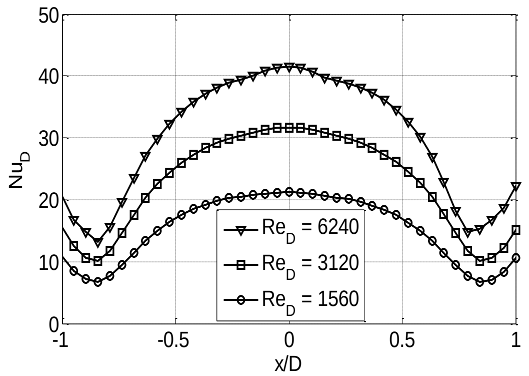


Figure 9. Average Nusselt number distribution, d/D = 0.105, θ_{max} = π/6

Finally, the effect of the basic geometrical parameters on the Nusselt number distribution is presented in Fig. 10. Unlike to the previous parameters, it is seen from this figure that the effect of d/D is not very significant on the Nusselt number distribution. For a constant length of the oscillating blade, Nusselt number decreases slightly over the entire plate as the blade tip-to-plate is increased.

This is because the flow intensity passing through tip-to-plate gap decreases in this case and hence convective heat transfer is slightly reduced. In a similar way, when the distance between the blade tip and the plate is constant, convective heat transfer rate is slightly intensified as the length of the blade is increased.

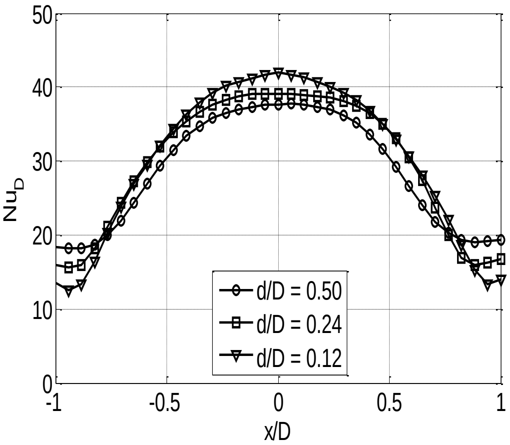


Figure 10. Average Nusselt number distribution, Re_D = 5000, θ_{max} = π/6

Conclusion

Convective heat transfer from a flat plate under the effect of an oscillating vertical blade has been investigated numerically. Based on the results, convective heat transfer has various distributions over the plate at different time instants of an oscillation period. Heat transfer rate intensifies over the central region of the plate when the oscillating blade passes from its upright position in both directions. Average Nusselt number distribution over a full cycle of oscillation has a bell shape distribution whose maximum is placed at the center of the plate. More surface area of the plate is affected when the amplitude of the oscillation is increased. Meanwhile, the convection rate remains almost constant over the central region. Average Nusselt number distribution over the plate increases almost linearly as the rotational Reynolds number is increased. The effect of the dimensionless geometrical parameter denoted by d/D is not very significant on the heat transfer rate from the plate and hence on the average Nusselt number distribution.

Nomenclature

d	Blade tip-to-plate separation [m]
D	Oscillating blade length [m]
h	Convective heat transfer coefficient [W m^{-2} K^{-1}]
H	Hydraulic diameter of the channel [m]
k	Thermal conductivity of air [0.0242 W m^{-1} K^{-1}]
Nu_D	Nusselt number [h D k^{-1}]
p	Pressure [N m^{-2}]
Re_D	Oscillation Reynolds number [D^2 ω ν^{-1}]
T	Temperature [K]
t	Time [s]
u	X-velocity component [m s^{-1}]
v	Y-velocity component [m s^{-1}]
x	Cartesian coordinate [m]
y	Cartesian coordinate [m]

Greek Symbols

ρ	Air density [1.225 kg m^{-3}]
τ	Oscillation period [s]
θ	Oscillation amplitude [Degree 0r Rad]
ω	Angular velocity [Rad s^{-1}]
ν	Air kinematic viscosity [1.455 × 10^{-5} m^2 s^{-1}]

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