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FORCED CONVECTION HEAT TRANSFER AROUND A CONSTANT SURFACE HEAT FLUX HORIZONTAL CYLINDER WITH BAFFLES. REYNOLDS NUMBER RE<40 CASE

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ABSTRACT

This paper presents the heat transfer management around a horizontal circular cylinder that has a constant surface heat flux using uniformly distributed baffles. The dimensionless boundary conditions are solved in the computational domain using the finite differences methods, in a hybrid scheme, and the results are emphasizing the Nusselt number variation as a function of Reynolds number, number of baffles and baffle height. The analysis shows that for a constant surface heat flux horizontal cylinder and for a Reynolds number less than 40, a reduction as high as 52% can be obtained using uniformly distributed baffles. For small values of baffle height, an increase of the Nusselt number is registered and it becomes more pronounced as the Reynolds number decreases and the number of baffles increases.

KEYWORDS: forced convection, heat flux, horizontal cylinder, baffles

1. INTRODUCTION

The convective heat transfer from a constant heat flux horizontal cylinder to the fluid surrounding has been considered a subject for theoretical and analytical studies for many decades, an interest that is determined by both the industrial and the theoretical applications of this fundamental problem. These studies clarified the velocity and temperature fields of the fluid surrounding the cylinder and determined parameters with practical importance regarding the intensity of heat transfer from the cylinder to the surrounding (the Nusselt number), the drag coefficient, the separation angle, the wake length. The dependance of these variables on the problem parameters, the Peclet number, Pe, the Reynolds number, Re, or with the power-law index of the fluid, n, (for the non-Newtonian fluids) was analyzed.

The drag coefficients and the forced convective heat transfer around a horizontal cylinder with constant surface temperature/heat flux were determined numerically by Soares et al. [16], using *the finite difference method*, for the case of non-newtonian fluids with a wide range of power-law indices $(0.5 \le n \le 1.4)$, Reynolds numbers $(5 \le \mathrm{Re} \le 40)$ and Prandtl numbers $(1 \le \mathrm{Pr} \le 100)$.

The same problem was analyzed in greater detail by Bharti et al. [7], using *the finite volume method*, for the same range of Reynolds number $(5 \le \mathrm{Re} \le 40)$ but for wider ranges of power-law indices $(0.6 \le n \le 2.0)$ and Prandtl numbers $(1 \le \mathrm{Pr} \le 1000)$. The dependence of the Nusselt number at the stagnation points on the dimensionless parameter of the process $(\mathrm{Re}, \mathrm{Pr}, n)$ and the boundary condition at the surface of the cylinder help us to understand this heat transfer process. The numerical results determined the correlation of the average Nusselt number on the dimensionless process parameters $(\mathrm{Re}, \mathrm{Pr}, n)$. This correlation is the developement of the previous dependence determined earlier [8] for the case of Newtonian fluids, a Reynolds number range of $(10 \le \mathrm{Re} \le 45)$ and a Prandtl number in the range $(0.7 \le \mathrm{Pr} \le 400)$.

Khan et al. [11] used an *integral approach* to analyze the fluid flow and heat transfer from a horizontal cylinder with contant surface temperature/heat flux. They detemined closed form expressions for the drag coefficients and the average Nusselt number. They continued their research for the case of power-law fluids [12] and determined the correlation between the Nusselt number and both the Reynolds and Prandtl numbers.

Nazar et al. [13] established the *non-similar solution* of the boundary layer equations for the case of mixed convection from a horizontal circular cylinder with a constant surface heat flux immersed in a viscous and incompressible fluid. The influence of the mixed convection parameter on flow and heat transfer characteristics are studied for two values of Prantl numbers: $\Pr = 1$ and $\Pr = 7$.

The mixed convection around a circular cilinder with a constant temperature/heat flux surface was studied in greater detail for the parallel and contra flow regimes by Soares et al. [17] using the finite difference method.

The case of a rotating cylinder dissipating heat flux in a 2D laminar regime was analyzed by Paramane et al. [15] for the Reynolds range of $(20 \le \mathrm{Re} \le 160)$ and a Prandtl number of 0.7, using the finite volume method.

The research activity concerning the fluid flow and heat transfer around a horizontal circular cylinder is not constrained to analyzing the flow and temperature fields or the heat transfer coefficients, but it tries to find out solutions to increase/decrease the heat transfer from the cylinder to the surrounding medium [1–6, 9, 14]. The method of using external fins or buffles uniformly distributed around a horizontal circular cylinder in order to modify the heat transfer coefficients in the cases of natural, forced or mixed convection is not new. Promissing results were obtained for the case of a constant temperature horizontal cylinder by Abu-Hijleh [1–6]:

- for the case of a *natural convection* heat transfer from a constant surface temperature cylinder with baffles, a reduction of the average Nusselt number up to 72% was determined numerically [1]; for the same case, an increase of Nusselt number can be obtained by using permeable fins with a maximum of 84% increase (at Rayleigh number of 10^6, a dimensionless baffle height of 2.0 and 11 fins uniformly distributed around the cylinder surface) [3]. The optimum placement of the baffles on the cylinder surface was analyzed [5] for the case of one and two baffles, respectivelly;

- for the case of *forced convection* heat transfer, Abu-Hijleh [6] analyzed the case of high conductivity radial fins in cross flow determined the existence of an optimum number of fins that is dependent on the Reynolds number and fin height. An even higher increase of the average Nusselt number can be obtained using permeable fins. A comparative study [4] of the permeable/solid fins influence revealed a relative ratio of 4.5 between the Nusselt number determined by permeable vs. solid fins for a Rayleigh number of 150, a single fin and a dimensionless fin height of 3.0;

- the *mixed convection* heat transfer case analysis performed for the case of low conductivity baffles in cross-flow [2] determined a reduction of the average Nusselt number as high as 75% for an analysis that considered the following range of parameters: $(10 \le \mathrm{Re} \le 200)$, $(0 \le N \le 18)$, $(0 \le H \le 3.0)$ and $(0 \le \gamma \le 5.0)$, where γ is buoyancy parameter. An optimum baffle height, Reynolds number dependent, was revealed by this numerical analysis.

Campo and Cortés [9] used the *1D lumped analysis* to demonstrate that using external baffles on the surface of a cylinder reduces the natural convective heat loss from the internal hot fluid to the ambient air. A reduction as high as 60% was calculated for an in-tube laminar oil flow and 11 baffles uniformly distributed on the exterior surface of the cylinder.

The case of a horizontal cylinder with a constant surface heat flux that is provided with external uniformly distributed baffles and that loses heat by natural convection was analyzed by Neagu [14].

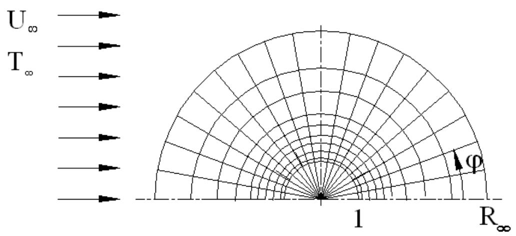
This work adds a new chapter to the theoretical knowledge we have regarding the modification of the forced convective heat transfer from a constant surface heat flux horizontal cylinder using exterior uniformly distributed baffles. For a Prandtl of 0.72 and a Reynolds number in the 1-40 range, this paper determines the Nusselt number for a number of baffles between 1 and 18 and a baffle height between 0.15 and 3.0. These results are normalized with the value of the Nusselt number for the case of a cylinder without baffles, in similar conditions, presenting, in this way, a clearer image of the baffles' influence. This study shows that, generally, the baffles tend to decrease the heat transfer from a constant surface heat flux horizontal cylinder to the fluid. This decrease is more significant when the number of baffles and/or the baffle height increases. For small values of the baffle height, an increase of the Nusselt number is registered and this increase becomes even more important for smaller values of Reynolds number and for bigger number of baffles.

2. MATHEMATICAL MODELING

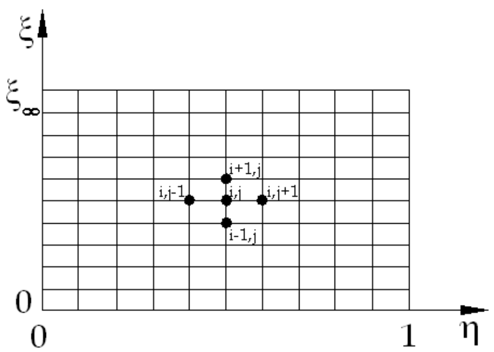
The horizontal cylinder with a constant surface heat flux and uniformly distributed baffles on its outer surface is presented by fig. 1(a) in dimensionless coordinates: (R, φ). The stream function – vorticity formulation of the dimensionless steady state governing equations for the fluid surrounding the cylinder up to a sufficiently far distance, R_∞, were derived elsewhere [12-14]:

$$\omega = -\left(\frac{\partial^2 \psi}{\partial R^2} + \frac{1}{R} \cdot \frac{\partial \psi}{\partial R} + \frac{1}{R^2} \cdot \frac{\partial^2 \psi}{\partial \varphi^2} \right) \qquad (1)$$

$$U\frac{\partial\omega}{\partial R}+\frac{V}{R}\cdot\frac{\partial\omega}{\partial\varphi}=$$
$$\frac{1}{Re}\left(\frac{\partial^2\omega}{\partial R^2}+\frac{1}{R}\cdot\frac{\partial\omega}{\partial R}+\frac{1}{R^2}\cdot\frac{\partial^2\omega}{\partial\varphi^2}\right) \quad (2)$$



(a)



(b)

Fig. 1. *(a) Dimensionless physical domain of the horizontal cylinder with baffles; (b) computational domain.*

$$U\frac{\partial\theta}{\partial R}+\frac{V}{R}\frac{\partial\theta}{\partial\varphi}=$$
$$\frac{1}{Re\cdot Pr}\left(\frac{\partial^2\theta}{\partial R^2}+\frac{1}{R}\cdot\frac{\partial\theta}{\partial R}+\frac{1}{R^2}\cdot\frac{\partial^2\theta}{\partial\varphi^2}\right) \quad (3)$$

Here, the potential function, ψ, is defined through the following two definitions: $U=(1/R)(\partial\psi/\partial\varphi)$ and $V=-(\partial\psi/\partial R)$, while the vorticity, ω, is defined as:

$$\omega=-(1/R)(\partial U/\partial\varphi)+(1/R)(\partial(RV))/\partial R).$$

The boundary conditions applied to the system of Eqs. (1)–(3) are:

- at the cylinder surface, $R=1$: no slip and no penetration conditions imply $\psi=\partial\psi/\partial R=0$, while the constant heat flux condition requires $\partial\theta/\partial R=-1$;

- at infinity, $R=R_\infty$: $(1/R)(\partial\psi/\partial\varphi)=\cos(\varphi)$ and $(\partial\psi/\partial R)=\sin(\varphi)$, while $\theta=0$ in the inflow region $(\pi/2\le\varphi\le\pi)$ and $\partial\theta/\partial R=0$ in the outflow region $(0\le\varphi\le\pi/2)$;

- on the symmetry plane, $\varphi=0$ and $\varphi=\pi$, the symmetry of velocity and temperature fields demands $\psi=\omega=\partial\theta/\partial\varphi=0$;

- on the baffle surface: $\psi=0$, while the temperature is the average of the radial neighbor points temperature.

For a better representation of the phenomena that take place around the cylinder, a coordinate transformation, $R=e^{\pi\xi}$; $\varphi=\pi\eta$, that transforms the stretched grid presented by Fig. 1a into a uniform grid in the computational domain (see fig. 1(b)) is used. The governing equations in the (ξ,η) system of coordinates become [12-14]:

$$\omega=-\frac{1}{\pi e^{\pi\xi}}\left[\frac{\partial^2\psi}{\partial\xi^2}+\frac{\partial^2\psi}{\partial\eta^2}\right] \quad (4)$$

$$\frac{\partial\psi}{\partial\eta}\frac{\partial\omega}{\partial\xi}-\frac{\partial\psi}{\partial\xi}\frac{\partial\omega}{\partial\eta}=\frac{1}{Re}\left(\frac{\partial^2\omega}{\partial\xi^2}+\frac{\partial^2\omega}{\partial\eta^2}\right) \quad (5)$$

$$\frac{\partial\psi}{\partial\eta}\frac{\partial\theta}{\partial\xi}-\frac{\partial\psi}{\partial\xi}\frac{\partial\theta}{\partial\eta}=\frac{1}{Re\cdot Pr}\left(\frac{\partial^2\theta}{\partial\xi^2}+\frac{\partial^2\theta}{\partial\eta^2}\right) \quad (6)$$

The new boundary conditions are:

- at the cylinder surface, $\xi=0$: $\psi=\partial\psi/\partial\xi=0$, $\partial\theta/\partial\xi=-\pi$;

- at infinity, $\xi=\xi_\infty$: $(\partial\psi/\partial\xi)=\pi e^{\pi\xi}\sin(\varphi)$, while $\omega=\theta=0$ in the inflow region $(1/2\le\eta\le1)$ and $\partial\omega/\partial\xi=\partial\theta/\partial\xi=0$ in the outflow region $(0\le\eta\le1/2)$;

- on the symmetry plane, $\eta=0$ and $\eta=1$: $\psi=\omega=\partial\theta/\partial\varphi=0$;

- on the baffle surface: $\psi=0$, $\theta_{i,j}=(\theta_{i,j+1}+\theta_{i,j-1})/2$.

The system of Eqs. (4)–(6) with the corresponding boundary conditions were solved using the finite differences method [18], the hybrid scheme. The computer program was tested for the case of a horizontal circular cylinder with a constant surface heat flux, without baffles, for different Reynolds and Prandtl numbers and the results were compared with the data found in the literature. Table 1 shows a very good agreement.

A number of 91-96 points in the tangential direction, a number of 80 points in the radial direction and an infinite radius of 20 were enough to compute the Nusselt number within 1% variation error and a relative error of the iterative procedure of 10^{-4}.

A set of 1, 3, 5, 8, 12 and 18 baffles with baffle heights of 0.15, 0.35, 0.75, 1.5, 2.0 and 3.0 were considered. Reynolds number, Re_D, varied between 1 and 40.

Table 1. *Comparison of the average Nusselt number with the literature values*

Pr=1					
Re_D	Bharti et al. [7],	Soares [16] as reported by Bharti et al. [7]	Khan et al. [11]		This work
5	1.700	1.693	1.413		1.696
10	2.309	2.259	1.998		2.823
20	3.147	3.048	2.826		3.092
30	3.763	3.622	3.461		3,675
40	4.274	4.104	3.997		4.153
Pr=0.7					
Re_D	Bharti et al. [8]	Parama-ne et al. [15]	Dhiman at al. [10] as calculated by Bharti et al. [7]	Khan et al. [11]	This work
10	2.040		2.146	1.774	2.02
20	2.778	2.695	2.863	2.509	2.722
40	3.775	3.6815	3.793	3.549	3.705

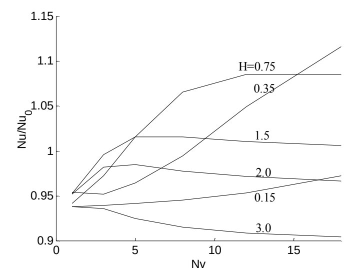
3. RESULTS AND DISCUSSIONS

The influence that the exterior baffles have on the heat transfer from a constant surface heat flux cylinder to the surrounding medium is analyzed using the dimesionless heat transfer coefficient, the average Nusselt number. The number of baffles, the baffle height and the Reynolds number are the parameters whose influence on the average Nusselt number is analyzed.

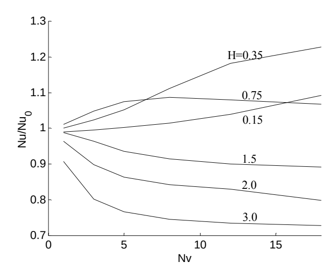
Figure 2 presents the Nusselt number variation as a function of number of baffles for different baffle heights (H= 0.15, 0.35, 0.75, 1.5, 2.0 and 3.0) and for three values of Reynolds number, Re_D: 1, 5, 20. As we can notice, the increase of the number of baffles increases the average Nusselt number for a baffle height as small as 0.15 and for the Reynolds number in the range $(1 \le Re_D \le 40)$. This can be explained by the "disturbance" [1] the baffle is creating on the boundary layer formed at the cylinder surface, a disturbance that increases as we add more and more baffles. As Reynolds number increases, the boundary layer thickness decreases and the baffle thickness that determines an increase of the Nusselt number decreases, an aspect noticed by comparing fig 2(a), fig. 2(b) and fig. 2(c).

For the situations where the normalized Nusselt number decreases as the number of baffles increases we can notice that there is an optimum number of baffles beyond which the Nusselt number does not decrease significantly. The reduction is higher for a higher Reynolds number and it can become as high as 52% for a Reynolds number of 40, 18 baffles with a dimensionless height of 3.0.

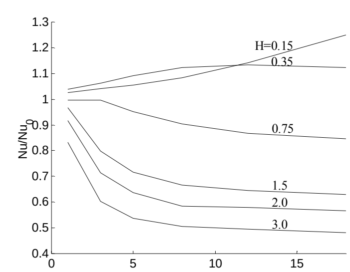
The influence the baffle height has on the average Nusselt number is analyzed by fig. 3. It presents the normalized Nusselt number variation as a function of baffle height for different numbers of baffles (Nv=1, 3, 5, 8, 12 and 18) and for the same values of Reynolds number: 1, 5 and 20. Analyzing fig. 3 we conclude that increase of the baffle height as high as 3.0 continues to decrease the average Nusselt number, i.e., the heat transfer from the horizontal constant surface heat flux cylinder to the surroundings. This dependence is reversed for smaller Reynolds number and baffle height as already indicated by fig. 2.



a) Re_D=1

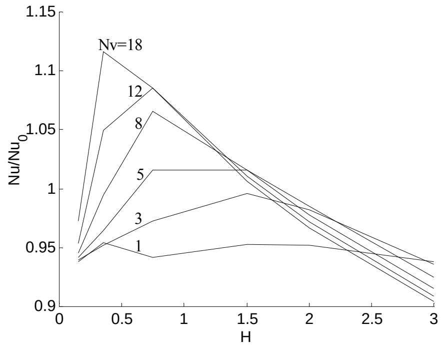


b) Re_D=5

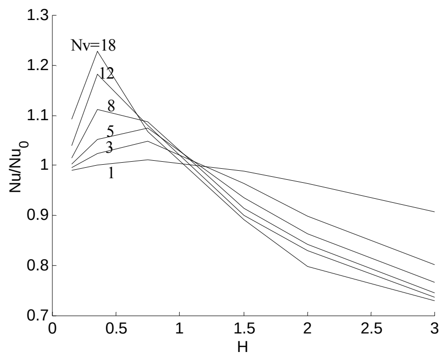


c) Re_D=20

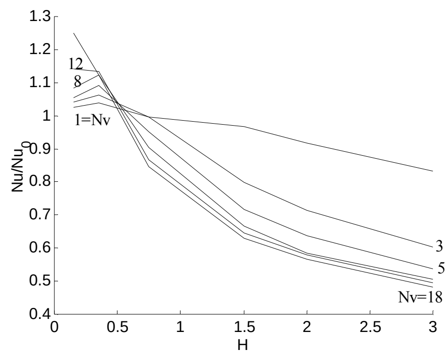
Fig.2. *Normalized Nusselt number variation as a function of number of baffles for different baffle height (H= 0.15, 0.35, 0.75, 1.5, 2.0 and 3.0) and for a Reynolds number Re_D of (a) 1, (b) 5 and (c) 20.*



a) Re_D=1



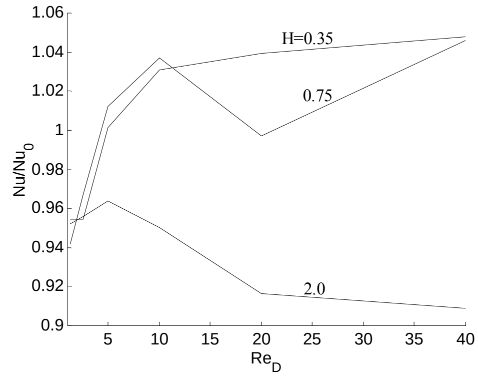
b) Re_D=5



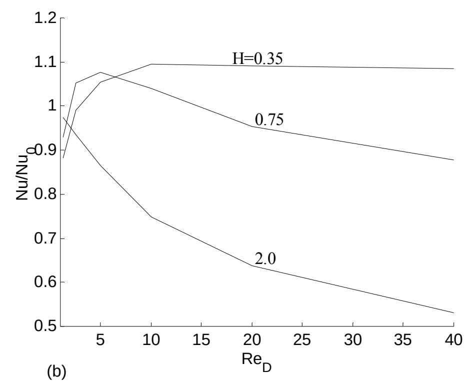
c) ReD=20

Fig. 3. *Normalized Nusselt number variation as a function of baffle height, for different numbers of baffles (Nv=1, 3, 5, 8 and 18) and for Reynolds number, Re_D: (a) 1, (b) 5, (c) 20*

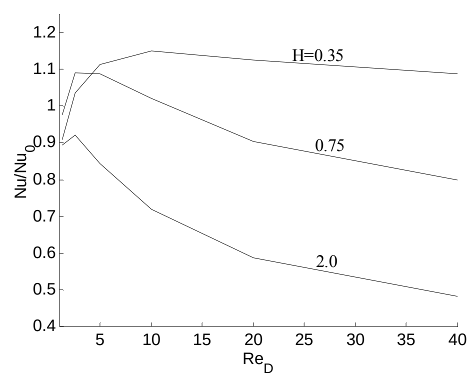
The Reynolds number's influence on the normalized Nusselt number is analyzed by fig. 4 for three cases: 1, 5 and 8 baffles, respectively, with three baffle heights: 0.35, 0.75 and 2.0. We can notice that the increase of the Reynolds number (except the small values of Reynolds number) decreases the heat transfer from the cylinder. This decrease is more pronounced for a greater number of baffles and a higher baffle height.



a) Nv=1



b) Nv=5



c) Nv=8

Fig. 4. *Normalized Nusselt number variation as a function of Reynolds number for different baffle height (H=0.35, 0.75 and 2.0) and for a number of baffles, Nv, of (a) 1. (b) 5 and (d) 8*

5. CONCLUSIONS

This work completes the knowledge we have regarding the modification of heat transfer from a horizontal constant surface heat flux and the surrounding medium using baffles and fins. It shows that in order to decrease the heat transfer from the cylinder, i.e. the Nusselt number, a bigger number of baffles have to be added to the cylinder surface; the increase of the baffle height decreases further the heat transfer.

For small values of baffle height, an increase of the Nusselt number is noticed. This trend becomes more pronounced as both the number of baffles increases and the Reynolds number decreases.

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NOMENCLATURE

D	cylinder diameter
H_b	dimensionless baffle height, $H = h_b / r_0$
h	convective heat transfer coefficient
h_b	baffle height
k	fluid thermal conductivity
N_v	number of exterior equally spaced baffles
Nu	average Nusselt number with baffles, $Nu = \frac{1}{\pi}\int_0^{\pi} hD/k \, d\varphi$
Nu_0	average Nusselt number without baffles
Pr	Prandtl number, $Pr = \nu / \alpha$
q	constant surface heat flux
R	dimensionless radius, $R = r / r_0$
Re	Reynolds number based on cylinder radius, $Re = U_\infty r_0 / \nu$
Re_D	Reynolds number based on cylinder diameter, $Re_D = U_\infty D / \nu$
R_∞	dimensionless radius defining the boundary domain, $R_\infty = r_\infty / r_0$
r	radius
r_∞	radius defining the boundary domain
r_0	exterior cylinder radius
T	fluid temperature
T_∞	free stream temperature
U	dimensionless radial velocity, $U = u / U_\infty$
u	radial velocity
U_∞	free stream velocity
V	dimensionless tangential velocity, $V = v / U_\infty$
v	tangential velocity
X	horizontal axis
Y	vertical axis
α	fluid thermal diffusivity
η	calculation coordinate, $\varphi = \pi\eta$
θ	dimensionless temperature, $\theta = (T - T_\infty)/(qD/k)$
μ	dynamic viscosity of the fluid
ν	kinematic viscosity
ξ	calculation coordinate, $R = exp(\pi\xi)$
ξ_∞	dimensionless radius defining the boundary domain, $R_\infty = exp(\pi\xi_\infty)$
φ	angle
ψ	dimensionless stream function
ω	dimensionless vorticity