

Transient Cooling Of A Cylinder In Cross Flow Bounded By An Adiabatic Wall

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The present study investigates the parameters controlling the cooling process of a cylindrical food in the storage area for a period of time. Transient analysis of the conduction and convection (conjugate) heat transfer from a cylindrical food, or a cylindrical can filled with food is selected for numerical simulations. The cylinder is bounded by an adiabatic wall and the cold air is flowing normal to the cylinder axis (cross flow). The parameters investigated are: Reynolds number, food thermal properties (density, specific heat and thermal conductivity) and the cooling period. The range of the Reynolds number is selected from 50 to 500 to be in laminar flow conditions. Three different materials were selected according their thermal properties. The results are presented to show the cooling process starting from blowing cold air stream on the cylinder for a period of 4 hours. The results show that the food with low thermal inertia is cooled faster than that of high thermal inertia. The present results show also that the cooling process can be shortened by increasing the air velocity and lower its temperature.

Keywords: Computational fluid dynamics, Transient analysis, Conjugate heat transfer, Circular cylindrical, Cross flow, Thermal inertia.

INTRODUCTION

Agricultural products can be preserved with their desirable characteristics by controlling the temperature and humidity of the air inside the storage area for a period of time. The quality of the agricultural products is related to the uniform cooling and humidification, especially during long-term storage. Usually pre-storage treatment includes cooling the food to low temperature in most of food engineering operations. The

cold store is used to store different products and different periods and the uniform air conditions are difficult to attain in the industrial food storage rooms. Non uniform cooling causes either increasing the respiration at high air temperature or chilling (in sometimes freezing) at low air temperature.

The literature review shows a vast use of computational fluid dynamics (CFD) techniques in the agricultural products storage and processing industry. The computational fluid dynamics can be used

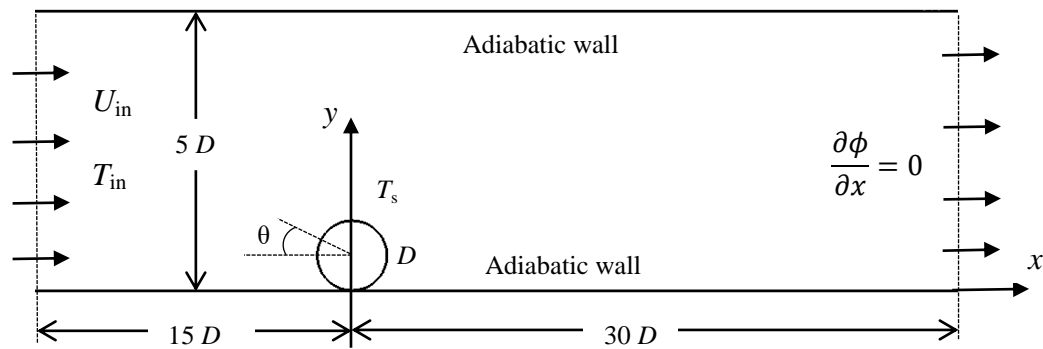


Fig. 1: The physical model and coordinate geometry

to model and numerically simulate the heat and mass transfer in this complicated geometry for the design optimization. CFD has been used in the recent years in many food processing and storage industry. Review articles can be found in the following references (Wang and Sun, 2003, Norton et al, 2007, Dehghannya et al, 2010, Lemus-Mondaca et al (2011), Ambaw et al, 2013).

The heat transfer and flow structure across non-bounded circular cylinders (suspended in a stream of fluid) is studied extensively due to its importance in many industrial applications. Literatures concerning convective heat transfer from circular cylinder in cross flow are available in abundance. Representative studies in this area may be found in the books by Sumer and Fredsoe (1997) and Zdravkovich (1997, 2003) and many review articles, for examples: Coutanceau and Defaye (1991) and Williamson (1996), Norberg, (2003) and Sumner (2010).

It is noticed from the literature review that the heat transfer and flow structure across circular cylinder bounded by a wall is not considered for investigation. There many applications in for example cooling of cylindrical food placed on a surface by

blowing cold are normal to its axis. Motivated by the lack on such important analysis, the objective of the present study is to analyse the parameters effecting the transient conjugate cooling of a cylindrical food bounded by adiabatic surface using the numerical simulation. **Figure 1** show the physical model and the coordinate geometry of the present problem.

MATHEMATICAL MODEL

The mathematical model of a conjugate heat transfer is describing the conduction heat transfer through a solid body and the fluid flowing around or inside it. The heat and mass transfer in the cooling of agriculture products is a conjugate heat transfer phenomenon. It is important to investigate the cooling parameter affecting the quality of the products during the storage period.

In the present study the cooling fluid is assumed to be air as incompressible (constant density) with constant properties, and the buoyancy effect was neglected. It is assumed that the length of the cylinder is much greater than its diameter and therefore the cooling process can be modelled in two dimensions across the

cylinder. The height of the duct is assumed 5 times the diameter of the cylinder so that the wall effects of the duct can be neglected. Similarly, the dimensions of the upstream and downstream distances shown in Figure 1 are kept much larger than the cylinder diameter to justify correct implementation of the boundary conditions at the inlet and outlet. Therefore, the governing equations for the present transient conjugate heat transfer are:

$$\nabla \cdot \vec{V} = 0 \quad (1)$$

$$\frac{\partial(\rho u)}{\partial t} + \nabla \cdot (\rho u \vec{V}) - \nabla \cdot (\mu \nabla u) = -\frac{\partial p}{\partial x} \quad (2)$$

$$\frac{\partial(\rho v)}{\partial t} + \nabla \cdot (\rho v \vec{V}) - \nabla \cdot (\mu \nabla v) = -\frac{\partial p}{\partial y} \quad (3)$$

$$\rho c_p \frac{\partial T}{\partial t} + \rho c_p \nabla \cdot (\vec{V} T) - \nabla \cdot (k \nabla T) = 0 \quad (4)$$

where \vec{V} is the velocity vector which has the components u and v in the x and y directions respectively. The heat diffusion equation for the solid food is:

$$(\rho c_p)_s \frac{\partial T_s}{\partial t} - \nabla \cdot (k_s \nabla T_s) = 0 \quad (5)$$

where subscript s stands for the solid cylinder. No slip (zero velocity) boundary condition is imposed on all solid surfaces. The duct walls are assumed adiabatic (thermally insulated) while coupled thermal boundary condition is imposed on the food surface. Mathematically it is represented as:

$$k_s(n \cdot \nabla T_s) = k(n \cdot \nabla T) \quad (6a)$$

$$T_s = T \quad (6b)$$

where n is the normal direction to the solid-fluid interface. At the inlet, uniform velocity and temperature (U_{in} and T_{in}) of the air are defined at the inlet boundary. At the outlet, zero diffusion flux normal to the outflow boundary for all variables is imposed as indicated in **Figure 1**.

Air is used as working fluid to cool the cylindrical food. Three cylindrical materials are considered in the present study. The physical properties of air and the cylindrical materials are considered constants as listed in **Table 1**.

NUMERICAL SCHEME

The governing equations (equations (1)-(5)) subjected to the boundary conditions, specified in the mathematical model, are

Table 1. Physical properties of air and cylindrical materials

Property	Air	Wood	Cucumber	Meat
		Suleiman et al (1999)	Fasina and Fleming (2001)	Marcotte and Taherian (2005)
density ρ (kg/m ³)	1.225	543	959	1020
specific heat c_p (J/kgK)	1006	1080	4040	3115
thermal conductivity k (W/mK)	0.0242	0.29	0.61	0.37
Viscosity μ (kg/ms)	1.789×10^{-5}	-	-	-

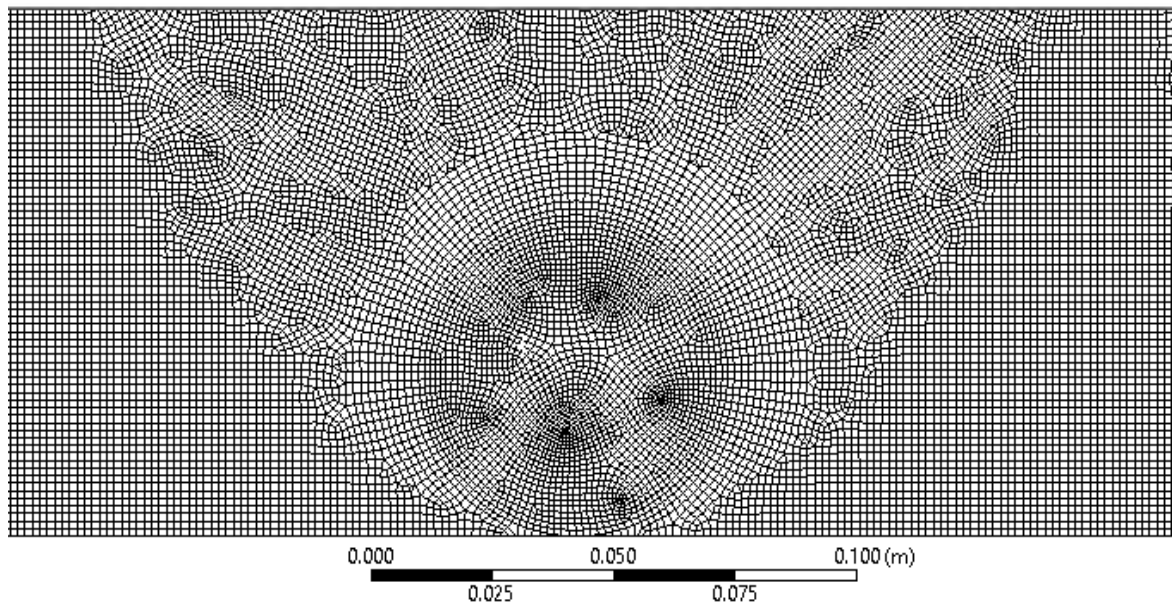


Fig. 2: Mesh geometry in the Vicinity of the Cylinder (total mesh size 417 209 cells)

solved numerically using the finite volume method (Patankar 1980) build in ANSYS (2011) software. This method is based on the discretization of the solution domain into finite number of control volumes (or cells) and integrates the governing equations over each control volume. Therefore the solution domain is discretised (meshed) by dividing it into uniform quadrilateral cells. Unstructured mesh is generated using ANSYS (2011) so that the mesh cells faces are aligned with the solid-fluid interface for accurate implementation of the condition 6(b). The mesh geometry in the vicinity of the cylinder is shown in **Figure 2**.

The time discretization scheme is implicit with first order accuracy with 30 seconds time step. For the spatial discretization, **the least squares cell-based gradient evaluation is used where** the dependent variables are assumed to vary linearly between the cells centres. The central differencing scheme is

used to approximate the diffusion terms of the energy equation as well as for the momentum equations. The second order upwind scheme is used for the convection-diffusion terms formulation of the momentum and energy equations. The discretized equations were solved based on the pressure-based coupled algorithm which solves a coupled system of equations comprising the momentum equations and the pressure-based continuity equation ANSYS (2011). This method is based on the iterative solution and the recommended values of the explicit relaxation factors for momentum and pressure are 0.75 and the flow Courant number is 200 (ANSYS, 2011). The convergence criterion is based on the residual in the governing equations. The maximum residual in the energy equation was 10^{-7} and the residual of other variables were lower than 10^{-6} in the converged solution. In all the computational cases the global heat and mass balance are satisfied in the converged solution within $\pm 10^{-3}\%$.

RESULTS AND DISCUSSION

The velocity and thermal fields generated in the numerical solution are used to evaluate the performance of the cooling of the circular cylinder. The convection process is evaluated based on the Nusselt number. The Nusselt number is calculated using the cylinder diameter D as the characteristic length. Therefore the Nusselt number can be defined as:

$$Nu_{\theta} = \frac{-(\partial T / \partial n)_{\theta} D}{(T_{s\theta} - T_{in})} \quad (7)$$

where $(\partial T / \partial n)_{\theta}$ is the local temperature gradient at the solid-fluid interface and $T_{s\theta}$ is the cylinder surface temperature with angle θ with the horizontal as shown in **Figure 1**. The average Nusselt number at the cylinder wall is also calculated by the area weighted average over the cylinder circumference as follows:

$$\overline{Nu} = \frac{1}{\pi D} \int_0^{\pi D} Nu_{\theta} ds \quad (8)$$

where s is the distance along the cylinder

circumference. To generalize the findings of the present study, the results are presented in terms of the Reynolds number based on the cylinder diameter ($Re = \rho U_{in} D / \mu$).

The effects of the number of the cells in the mesh on the accuracy of the results are tested by employing different mesh sizes and compare the results with the reference values. Due to lack of benchmark results (experimental or numerical) of the current problem, the results of the heat transfer from a non-bounded isothermal circular cylinder are used to validate the present results. **Table 2** listed the present results of the average Nusselt number around the cylinder with referenced values of the same flow across non-bounded circular cylinder.

The results presented in **Table 2** indicate that the values of the average Nusselt number for the non-bounded circular cylinder are higher than the respective values of the present simulations case. **Table 2** shows also that the mesh with 417 209 cells is suitable for the simulation as the differences in the calculated values of Nusselt number are small by approximately doubling the total number of cells in the mesh. Therefore the mesh with 417 209

Table 2. Comparison of the average Nusselt number

Re	Non-bounded circular cylinder				Circular cylinder bounded with adiabatic wall		
	Zukauskas (1972)	Morgan (1975)	Khan et al. (2005)	Haider et al. (2010)	Present 237 813 cells	Present 417 209 cells	Present 789 752 cells
50	3.18	3.68	3.7	3.82	2.055	2.053	2.052
100	4.49	5.10	5.2	5.15	2.741	2.736	2.734
200	6.35	7.07	7.4	7.12	3.682	3.669	3.662
300	7.78	8.56	9.0	8.59	4.383	4.358	4.344
400	8.99	9.80	10.3	9.97	4.973	4.930	4.906
500	10.05	10.89	11.5	11.17	5.503	5.435	5.400

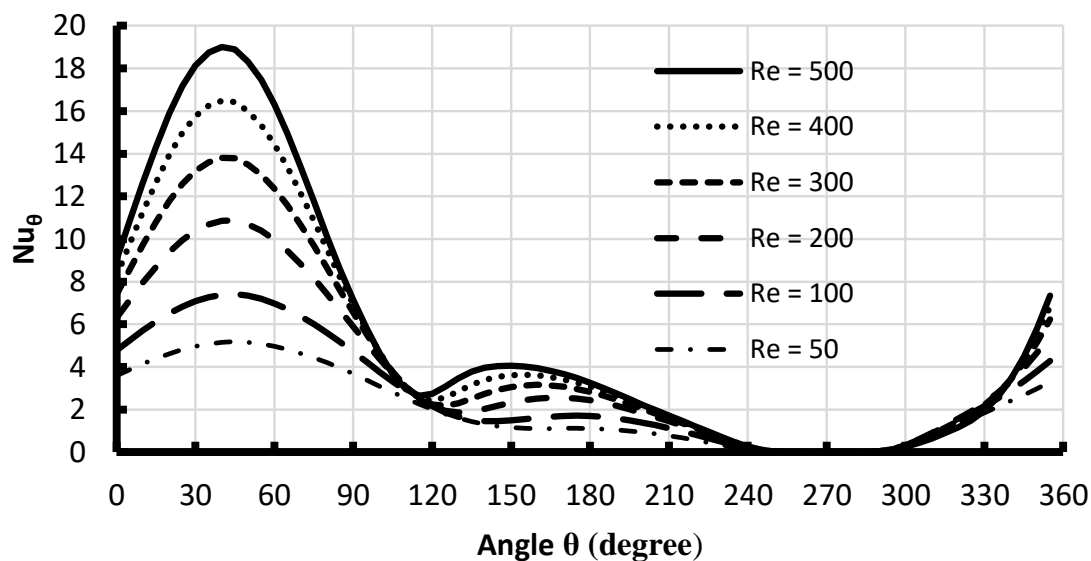


Fig. 3: Variation of Nu_{θ} for steady flow across isothermal horizontal cylinder

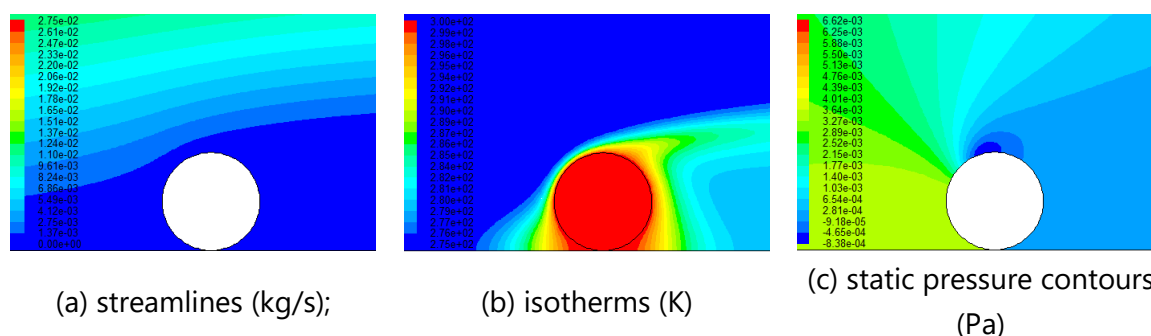


Fig. 4: Contours plot of the flow around the isothermal cylinder at 300K and $Re = 300$

cells will be used to generate the results in the present study.

The variation of the local Nusselt number of the baseline case, steady flow across isothermal horizontal cylinder bonded by an adiabatic wall, is presented in **Figure 3**. The respective average values of the Nusselt number are listed in **Table 2**.

Figure 3 shows the maximum value of the local Nusselt number at $\theta = 40^\circ$ which is unlike the case of the non-bounded circular cylinder where the maximum Nu is happening at $\theta = 0^\circ$ due to the presence of the bounded surface. Figure 3 also shows the minimum value of the local Nusselt number $Nu_{\theta} = 0$ at $\theta = 270^\circ$ which is the

contact point with the adiabatic boundary wall. The results presented in **Figure 3** show similar variation of Nu_{θ} for the cases with different values of Reynolds numbers. Higher values of Nu_{θ} are obtained in the cases of the flow with high Reynolds number.

The details of the flow around the cylinder for the case of maintaining the cylinder at constant temperature of 300K and blowing air with moderate Reynolds number ($Re = 300$) is shown in **Figure 4**. It is clear from the streamlines that the stagnation point is not the front surface of the cylinder ($\theta = 0^\circ$) rather than it is at an angle around $\theta = 40^\circ$. The isotherms also

show that the maximum clustering isotherms are at around $\theta = 40^\circ$ which results in maximum value of Nuessult number. The static pressure contours indicate the high pressure at this point at $\theta = 40^\circ$ and negative pressure behind the cylinder which leads to fluid circulation in the wake region of the cylinder. The results presented in **Figure 3** indicate that similar flow structure is expected at different values of Reynolds number.

To investigate the transient effects on the cooling of the cylinder, the results are generated with constant initial conditions for all the cases as: at time = 0, the cylinder and the air in the duct is at 300K and the air at the inlet is at 275K and the inlet velocity value is calculated from the respective value of the Reynolds number of the considered case. The simulations are carried out on the three solid cylinders

listed in **Table 1**. In these cases the cylinder surface is not isothermal and its temperature depends on the cooling time. **Figure 5** depicted the isotherms near the cylinder made of soft wood at different time steps and Reynolds number. The results presented in **Figure 5** indicate that the cooling period needs to be increased for the low Reynolds number flow. It is obvious as the Reynolds number is increased by increasing the inlet air velocity and keeping other parameters fixed.

It is observed that the cylinder temperature near the stagnation point $\theta = 40^\circ$ is the lowest while the cylinder temperature near the contact point with the adiabatic wall $\theta = 270^\circ$ is the highest. This observation is valid for the other cylinders made of meat and cucumber as shown in **Figure 6** where the isotherms near the cylinders are presented after 4

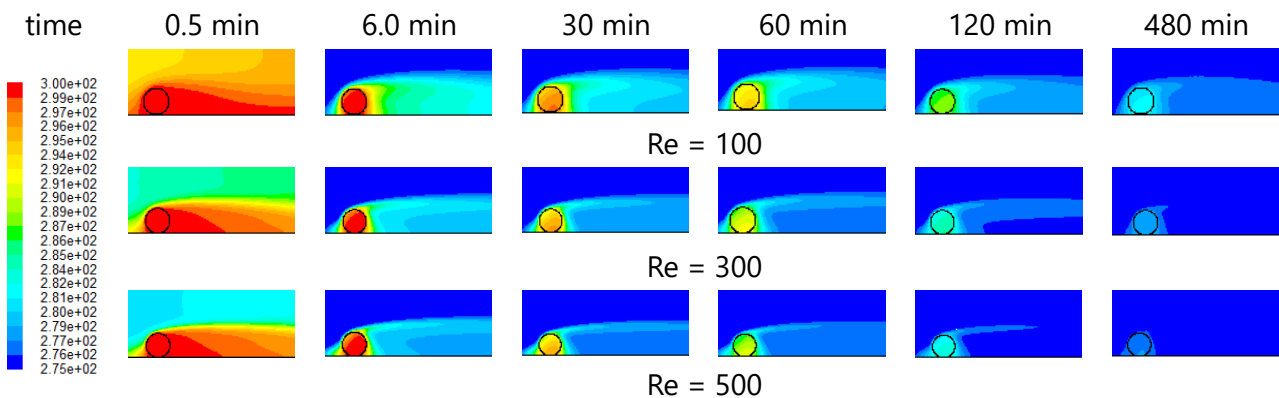


Fig. 5: Isotherms (K) near the cylinder made of wood

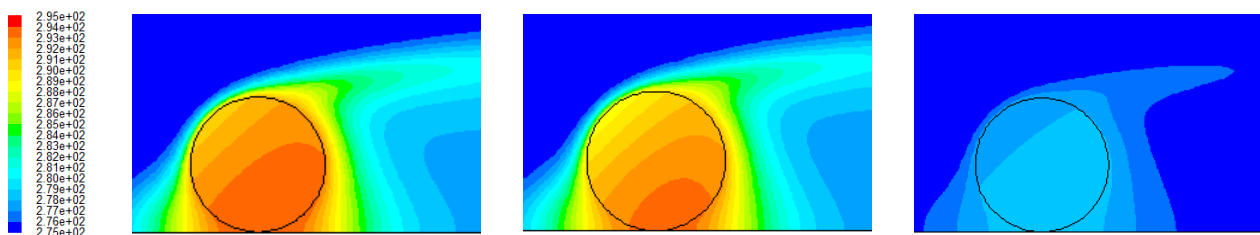


Fig. 6: Isotherms (K) near the cylinders after 4 hours of cooling by air at $Re = 300$. Cucumber (left), meat (middle), wood (right)

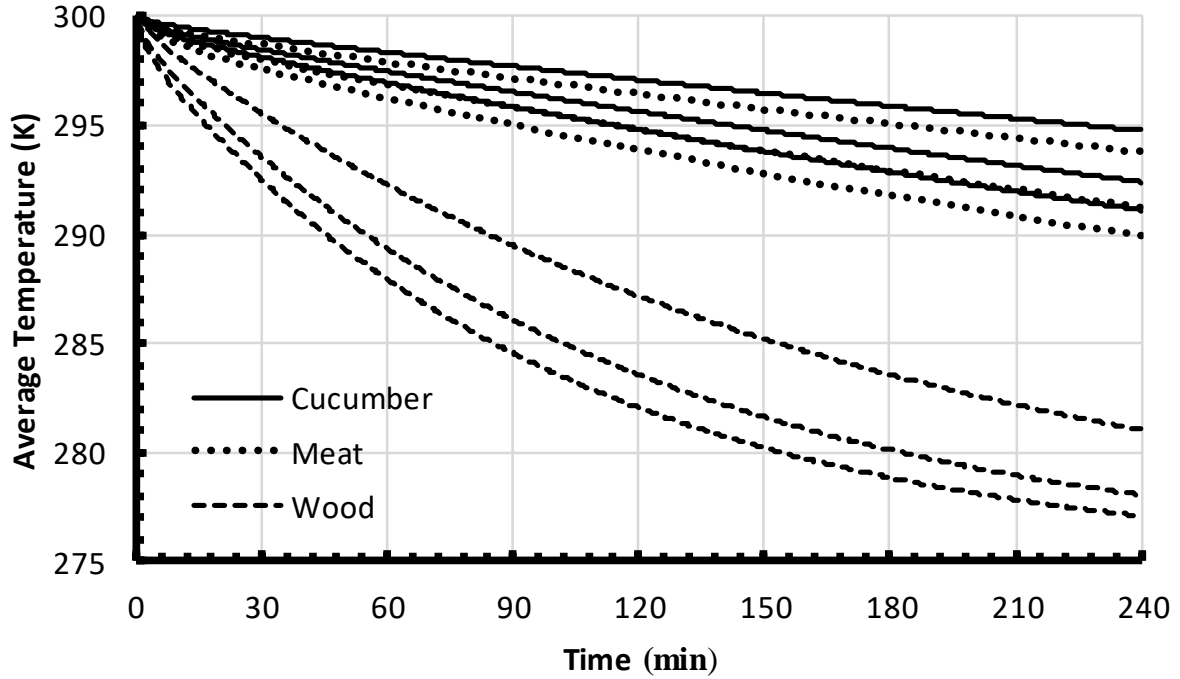


Fig. 7: Variation of the area weighted average temperature at the cylinder wall with cooling time (upper Re = 100; middle Re = 300; lower Re = 500)

hours of cooling by air at Re = 300.

Finally the area weighted average temperature at the cylinder wall is calculated as:

$$T_{av} = \frac{1}{\pi D} \int_0^{\pi D} T_{\theta} ds \quad (9)$$

The results of the area weighted average temperature at the cylinder wall is plotted against the cooling time for different Reynolds number and the three materials as shown in **Figure 7**. The results presented in **Figure 7** show that the area weighted average temperature at the cylinder wall is decreasing sharply for the cases of the cylinder made of wood compare with others for all values of Reynolds number.

This is due to the fact that the wood has low thermal inertia compare to the other two cylinders. The thermal inertia is defined

(Wang et al. 2010) as:

$$I = \sqrt{k\rho c_p} \quad (10)$$

Substituting the values of the materials properties listed in Table (1) in equation (10), the values of the thermal inertia of the material are: $I_{wood} = 413.1922 \text{ Jm}^{-2}\text{K}^{-1}\text{s}^{-1/2}$, $I_{meat} = 1084.251 \text{ Jm}^{-2}\text{K}^{-1}\text{s}^{-1/2}$ and $I_{cucumber} = 1537.322 \text{ Jm}^{-2}\text{K}^{-1}\text{s}^{-1/2}$. The results show also that the area weighted average temperature at the cylinder wall is reduced more with the increase of the Reynolds number. The area weighted average temperature at the cylinder wall after 4 hours of cooling with Re = 500 is reduced from 300K to 277.1 K for the wood compare to 289.9 K for the meat and to 291.1 K for cucumber.

CONCLUSIONS

Transient analysis of the conjugate heat transfer from a cylindrical food, or a cylindrical can filled with food is selected for numerical simulations. Cross flow on a cylinder is considered with the effect of an adiabatic boundary below the cylinder. The present parametric study considered the flowing parameters: Reynolds number, food thermal properties (density, specific heat and thermal conductivity) and the cooling period. The flow is assumed to be laminar and the three different foods were selected according to their thermal properties. The cooling process started with blowing cold air on the cylinder for a period of 4 hours. The results are presented for the steady state isothermal cylinder as well as transient cooling of the non-isothermal cylinder. It is observed from the results at all Reynolds numbers and cooling times that the lowest temperature of the cylinder happened near the stagnation point and its highest temperature occurs near the contact point with the adiabatic wall. The results show that the food with low thermal inertia is losing heat in shorter time than that of high thermal inertia. The present results indicate also that increasing the air velocity and lower its temperature will lead to shorten the cooling period.

NOMENCLATURE

c_p	: specific heat at constant pressure [J kg ⁻¹ K ⁻¹]
D	: diameter [m]
I	: thermal inertia [J m ⁻² K ⁻¹ s ^{-1/2}]
k	: thermal conductivity [W m ⁻¹ K ⁻¹]

n	: normal vector
Nu_θ	: local Nusselt number
\overline{Nu}	: average Nusselt number
p	: pressure [Pa]
Re	: Reynolds number
T	: temperature [K]
t	: time [s]
\vec{V}	: velocity vector [m s ⁻¹]
u, v	: velocity component in x and y directions respectively [m s ⁻¹]
x, y	: Cartesian coordinate [m]
ρ	: density [kg m ⁻³]
μ	: dynamics viscosity [kg m ⁻¹ s ⁻¹]
θ	: angle [degree]

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