An Analytical Study for Fluidized Bed Cooler Thermal Design

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Abstract

The paper describes an investigation for the thermal design of a fluidized bed cooler and prediction of heat transfer rate among the media categories. It is devoted to the thermal design of such equipment and their application in the industrial fields. It outlines the strategy for the fluidization heat transfer mode and its implementation in industry. The thermal design for fluidized bed cooler is used to furnish a complete design for a fluidized bed cooler of Sodium Bicarbonate. The total thermal load distribution between the air-solid and water-solid along the cooler is calculated according to the thermal equilibrium. The step by step technique was used to accomplish the thermal design of the fluidized bed cooler. It predicts the load, air, solid and water temperature along the trough. The thermal design for fluidized bed cooler revealed to the installation of a heat exchanger consists of (65) horizontal tubes with (33.4) mm diameter and (4) m length inside the bed trough.

Keywords: Fluidization, powder technology, thermal design, heat exchangers

دراسه تحليليه للتصميم الحراري لبرج متميع مبرد

الخلاصة

يتضمن هذا البحث التنبؤ للتصميم الحراري للمبرد ذو الطبقه المتميعه والتنبؤ بمعدل انتقال الحراره خلال الماده. لقد كثف البحث لدراسه التصميم الحراري لمثل هذه الاجهزه وتطبيقاتها في الحقل الصناعي. يلخص البحث الاستراتيجيه لانتقال الحراره خلال الطبقه المتميعه وتطبيقه في الصناعه. ان التصميم الحراري للمبرد ذو الطبقه المتميعه يستخدم كتصميم متقدم ومتكامل لتبريد كاربونات الصوديوم باستخدام المبرد ذو الطبقه المتميعه. ان توزيع الحمل الحراري الكلي بين الهواء – الصلب وبين الماء على طول المبرد حسب وفقا للتوازن الحراري. لقد تم استخدام تقنيه الخطوه – خطوه وذلك لانجاز التصميم الحراري للمبرد ذو الطبقه المتميعه. تم التنبؤ بالحمل ودرجه حراره كل من الهواء , الصلب والماء على طول الحوض. ان التصميم الحراري للمبرد ذو الطبقه المتميعه يبين ان المبادل الحراري يتكون من (65) انبوبا افقيا بقطر (33.4 mm) وبطول (4 m) داخل الحوض.

INTRODUCTION

The direct contact heat and mass transfer method has been adopted in many engineering fields by using different heat transfer media. In fluidized bed the cooling process is carried out in a trough fluidized by the cooling medium. Fluidized beds are commonly used in chemical, biochemical and petrochemical industries in processes such as hydrocarbon cracking, drying of solids, combustion

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and gasification of coal and biomass. Furthermore, it has been a wide implementation in thermal treatment of metals, recovery of energy from gases and hot solid particles, synthesis reactions and coating of particles. It is extensively implemented in particulate grain drying and cooling and a wide range of industrial applications.

Kim,etal. (2003) determined the effect of gas velocity on the average and local heat transfer coefficients between a submerged horizontal tube(25.4 mm-OD) and a fluidized bed. Fluidized-bed-heat-exchanger is (0:34× 0:50× 0:6 m-high) of silica sand particles. The local heat transfer coefficient exhibits a maximum value at the side of the tube. The bubble frequency increases and the emulsion contacting time decreases with increasing gas velocity [1]. Pécoraand Parise (2006) presented an experimental study of a continuous gas-solid fluidized bed with an immersed tube where cold water is heated by fluidized solid particles having inlet temperature from 450 to 700°C. Experiments were carried out in order to verify the influence of solid particle flow rate and distance between baffles immersed in a shallow fluidized bed. Results showed that the heat transfer coefficient increases with the solid flow rate and with the presence of baffles inside the bed [2]. Al- Dabbagh (2006) presented an experimental study of heat transfer between a shallow fluidized bed and the surface of a single horizontal tube and a tube bundle, which is immersed in it. Carbon, which is prepared from the Date stones, is used as a solid to be fluidized and a compressed air as an external fluid. The experimental research results and the Least Square Method are used to obtain Nusselt number correlation, for single tube [3]. Inaba et al. (2007) studied the effects of heat and mass transfer parameters on the efficiency of fluidized bed drying using two different materials, wheat and corn. Energy and exergy models based on the first and second law of thermodynamic are developed. Furthermore, some unified non-dimensional experimental correlations for predicting the efficiency of fluidized bed drying process have been proposed. With regard to the heat and mass transfer between air bubble and the wet material, it was clarified that reducing the Reynolds number will increase the efficiency of the drying process [4]. Ahn (2010) investigate the effects of circulating solid particles on the characteristics of fluid flow, heat transfer and cleaning effect in the fluidized bed vertical shell and tube type heat exchanger with counter flow. A variety of solid particles diameter and material such as glass (3 mm), aluminum (2～3 mm), steel (2～2.5 mm), copper (2.5 mm) and sand (2～4 mm) were used in the fluidized bed with a smooth tube. The operating conditions of the physical geometry of particles and material, and water flow rate were investigated [5]. Khorshidi et al. (2011) investigated the rate of heat transfer in a fluidized bed dryer. A correlation to predict the solid and outlet gas temperature changes were presented. The results of this study have shown that the maximum variations are occurred at the beginning of fluidization [6]. Ali (2013) studied the effect of baffles on the hydrodynamics of gas-solid circular fluidized bed with particle size (755,424 , 205μm). This study carried out with un-baffled and two types of baffle (rectangular and circular blade type). Results showed that rectangular is more significant than and circular baffle [7]. Ali (2013) studied the effect of material density and bed height on minimum fluidization velocity. The results showed that the minimum fluidization velocity increases with material density increased [8].

In the present study, an analytical study for design model is presented. It is intended to accomplish a general design method for a fluidized bed cooler. It implements a combination of solid-air and heat exchanger fluidization heat transfer modes. Two mediums for heat transfer barriers are used, air and water cooling in an immersed heat exchanger.

Solid – Gas Fluidized Bed without Immersed Tubes

The thermal design of bed cooler without immersed tubes as shown in shown in figure (1) consists of two parts. The first is the enthalpy balance and the second is the calculation of heat transfer rate.

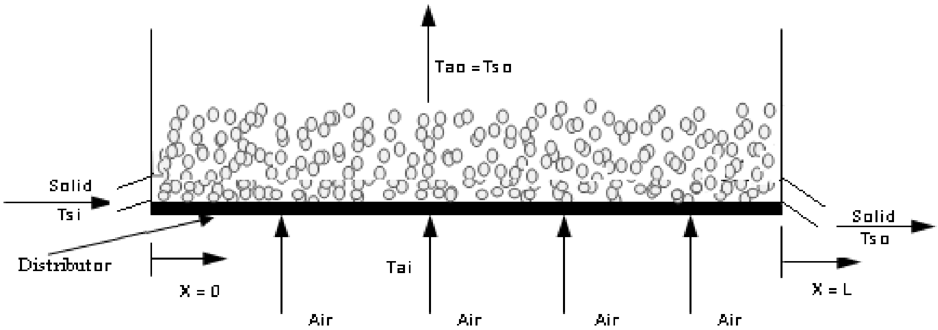

Figure (1): Schematic diagram of the gas-solid fluidized bed without immersed tubes.

Enthalpy Balance Mode

The enthalpy balance achieved between the solid and fluidized gas is shown by the following equation**,** [2]:

$$\dot{m}_g\, Cp_g\, \Delta T_g = \dot{m}_s\, Cp_s\, \Delta T_s \qquad \ldots(1)$$

In general, it is assumed that the final fluidized gas temperature is equal to the outlet solid temperature from the trough as revealed by:

$$\dot{m}_g\, Cp_g \left(T_{sf} - T_{ig}\right) = \dot{m}_s\, Cp_s \left(T_{sf} - T_{si}\right) \qquad \ldots.. \ (2)$$

Heat Transfer Mode

Heat transfer between particles and gas in a fluidized bed may be compared to gas convection from a single fixed particle and to gas convection from a packed bed of fixed particles.

The total load required for the cooling of the process material comprises of the net load that should be rejected to the cooling mediums expressed as:

$$\dot{Q}_{netl} = \dot{Q}_{total} - \dot{Q}_{losses} \qquad \ldots \ (3)$$

In which the following is true:

$$\dot{Q}_{total} = \dot{m}_s\, Cp_s\, \Delta T_s \qquad \ldots. \ (4)$$

And the heat loss to the surrounding of the fluidized bed in a form of heat loss through the trough walls is:

$$\dot{Q}_{loss} = hA\Delta T_s \qquad \text{….. (5)}$$

For immersion heat exchanger fluidized bed cooler, the net load that to be accomplished consists of both parts of the cooling mediums which can be expressed as follows:

$$\dot{Q}_{air} = \dot{m}_a \, Cp_a \, \Delta T_a \qquad \text{….. (6)}$$

And

$$\dot{Q}_w = \dot{Q}_{net} - \dot{Q}_{air} \qquad \text{…. (7)}$$

Noting that the heat transfer coefficient between particle and gas depends on the particle Reynolds number which may be expressed as, [9]:

$$Nu_p = 0.0282 \, \mathrm{Re}_p^{1.4} \, \mathrm{Pr}_g^{0.33} \qquad \text{…. (8)}$$

This correlation is applied in the range of particle Reynolds number of $0.1 \le \mathrm{Re}_p \le 50$

Important Parameters Criterion

Distributor Design

A good distributor design is based on achieving a pressure drop that is a sufficient fraction of the bed pressure drop to ensure an even gas distribution across the cross section of the bed. A common approach is to follow the guidelines proposed by Kunii and Levenspiel 2005 [9]. The most popular simple design of distributers is the multi – orifice plate suggested to be designed. This is done according to the criteria which states that the pressure drop across the distributor plate is preferred to be roughly (10 %) of the pressure drop across the bed. Therefore, ε_{mf} can be calculated by the following equation:

$$\varepsilon_{mf} = 1 - \frac{m_s}{h_{mf} A_c (\rho_s - \rho_g)} \qquad \text{… (9)}$$

Pressure Drop

The pressure drop across the bed is considered to be the main initiative to accomplish bed and air distributor design. The most popular correlation for such object is formulated according to the Ergun equation [10]:

$$\Delta P_b = h_{mf} \left(1 - \varepsilon_{mf}\right)\left(\rho_s - \rho_g\right) g \qquad \text{… (10)}$$

Then, the pressure drop across the distributor is estimated according to the experimental simulation argument is defined by Kunii and Levenspiel 2005 [9]:

$$\Delta P_d = 0.1 \times \Delta P_b$$

Orifice Velocity

The velocity through one orifice opening of distributor was found by Kunii and Levenspiel 2005 [9]:

$$U_{or} = C_D \sqrt{\frac{2g.\Delta P_d}{\rho_g}} \qquad \text{.... (11)}$$

The Fraction of Open Area(ϕ)

The ratio of minimum fluidizing velocity to the velocity of one-hole is expressed as :

$$\phi = \frac{U_{mf}}{U_{or}}$$

Determination of N_{or}

Calculate the number of orifice holes per unit area Nor

$$N_{or} = \frac{4U_{mf}}{\pi.d_{or}^2.U_{or}} \qquad \text{... (12)}$$

The total number of holes then:

$$(N_{or})_t = N_{or}(A_c)_{Trough} \qquad \text{... (13)}$$

d_{or} (cm)	0.05	0.1	0.15	0.2
N_{or} (Number/cm²)	4.7	1.18	0.53	0.3

Minimum Fluidization Condition

To establish the appropriate fluidization regime for any given application, one needs to calculate the minimum fluidization velocity and the terminal velocity of the bed particles.

The superficial velocity of the gas for minimum fluidization (u_{mf}) can be calculated by solving the following equation for R_{emf}, [9]:

$$Ar = 1.75\frac{(\mathrm{Re}_{mf})^2}{\phi_s(\varepsilon_{mf})^3} + 150\frac{(1-\varepsilon_{mf})(\mathrm{Re}_{mf})}{(\phi_s)^2(\varepsilon_{mf})^3} \qquad \text{.... (14)}$$

Ar is the Archimedes number, defined as:

$$Ar = \left(\frac{d_p^3\rho_g(\rho_s-\rho_g)g}{\mu_g^2}\right)$$

$$\mathrm{Re}_{mf} = \frac{\rho_g.u_{mf}.d_p}{\mu_g}$$

Minimum fluidization (u_{mf}) can be calculated by above equation

Drag Coefficient and Terminal Velocity

The drag coefficient calculation is of a vital importance in the design of a fluidized bed, a cooler or a dryer. Yang 2003 [10] had produced two correlations for the drag coefficient equations on the basis of the Archimedes number it may be found from the following according to ***Ar*** number:

$$C_D = \frac{432}{Ar}\left(1+0.0470Ar^{2/3}\right)+\frac{0.517}{1+154Ar^{1/3}} \qquad \ldots\ldots \ (15)$$

for $\text{Ar} < 1.18\times10^{6}\times d_p^2$

$$C_D = 0.95 \qquad \ldots\ldots. \ (16)$$

for $\text{Ar} > 1.18\times10^{6}\times d_p^2$

And the terminal velocity is expressed as:

$$U_t = \sqrt{\frac{4.d_p\left(\rho_s-\rho_g\right)g}{3.\rho_g.C_D}} \qquad \ldots.. \ (17)$$

Heat Transfer Performance

Fluidized Bed and Immersed Tubes

Heat transfer occurs between the fluidized particle/gas medium referred to as the ‘bed’’ and the submerged tube surfaces referred to as ‘‘walls’’. For this situation, one requires a heat transfer coefficient based on the surface area of the submerged wall [10]:

$$\frac{h_w\,(Do)_{tube}}{K_a} = 0.66(\Pr)^{0.3}\left((\mathrm{Re})\left(\frac{\rho_s}{\rho_a}\right)\left(\frac{1-\varepsilon_{mf}}{\varepsilon_{mf}}\right)\right)^{0.44} \qquad \ldots \ (18)$$

Where:

$$\mathrm{Re} = \frac{\rho_g.u_a.(Do)_{tube}}{\mu_g}$$

Then the final expression to the heat transfer coefficient can be expressed as:

$$\therefore \frac{h_w(Do)_{tube}}{K_a} = 0.66\left(\frac{\mu Cp_a}{K_a}\right)^{0.3}\left(\left(\frac{\rho_g.u_a.(Do)_{tube}}{\mu_g}\right)\left(\frac{\rho_s}{\rho_a}\right)\left(\frac{1-\varepsilon_{mf}}{\varepsilon_{mf}}\right)\right)^{0.44} \qquad \ldots(19)$$

Tube side heat transfer coefficient

Most of the available empirical correlations for the prediction of the tube side heat transfer coefficient in horizontal tube bank are based on fitting the data to a postulated expression depends on Reynolds Number. For turbulent flow in circular tubes, characterized by moderate property variations, the equation is recommended [8] in the form:

$$(Nu)_{tube} = 0.023(\mathrm{Re})^{0.8}(\mathrm{Pr})^{0.4} \quad \text{…. (20)}$$

Overall heat transfer coefficient

The overall coefficient of heat transfer between gas-solid suspension and water can be found by equation:

$$\dot{q}_w = U.A.LMTD \quad \text{…. (21)}$$

Where

the heat transfer area (A) represents the surface immersed tubs surface and the logarithmic mean temperature difference is given by:

$$LMTD = \frac{(T_{si} - T_{wo}) - (T_{so} - T_{wi})}{\ln\left(\frac{T_{si} - T_{wo}}{T_{so} - T_{wi}}\right)} \quad \text{….. (22)}$$

This procedure is based on the hypothesis of almost instantaneous thermal equilibrium between gas and solid particles. The average bed-to-tube heat transfer coefficient can be calculated by equation:

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{D_o}{2K_w}\ln\frac{D_o}{D_i} + \frac{D_o}{D_i} * \frac{1}{h_i} \quad \text{……. (23)}$$

Temperature Distribution along the Bed

If there is indirect cooling by water to the trough, the energy balance over the length of bed experience cooling by air and water, figure (2), may be expressed by:

$$\dot{m}_a\, Cp_a (T_s - T_{ia})\frac{dL}{L} - \dot{m}_w\, Cp_w\, dT_w = -\dot{m}_s\, Cp_s\, dT_s \quad \text{….. (24)}$$

$$-\dot{m}_w\, Cp_w\, dT_w = UA(T_s - T_w)\frac{dL}{L} \quad \text{….. (25)}$$

Solve above equation:

$$T_s = T_{ia} + K_1 e^{l\alpha 1} + K_2 e^{l\alpha 2} \quad \text{… (26)}$$

$$T_w = T_{ia} + K_1\left(\frac{a+\alpha 1}{b}\right)e^{l\alpha 1} + K_2\left(\frac{a+\alpha 2}{b}\right)e^{l\alpha 2} \quad \text{……. (27)}$$

The above mathematical energy representations should be solved with the following boundary conditions:

B.C.1 *l=0* $T_s = T_{si}$

B.C.2 $l=L$ $T_w=T_{wi}$

Where
the constants in the energy balance equations are corresponding to

$$a=\frac{\dot{m}_a\,Cp_a+U_oA_o}{\dot{m}_s\,Cp_sL} \qquad \ldots(28)$$

$$b=\frac{U_oA_o}{\dot{m}_s\,Cp_sL} \qquad \ldots\ (29)$$

$$c=\frac{U_oA_o}{\dot{m}_w\,Cp_wL} \qquad \ldots.\ (30)$$

$$\alpha_{1,2}=\frac{c-a\pm\sqrt{(a+c)^2-4ac}}{2} \qquad \ldots.\ (31)$$

With B.C.:

$$K_2=(T_{si}-T_{ai})-K_1$$

$$T_{iw}-T_{ia}=K_1\left(\frac{a+\alpha 1}{b}\right)e^{l\alpha 1}+((T_{si}-T_{ai})-K_1)\left(\frac{a+\alpha 2}{b}\right)e^{l\alpha 2}$$

$$K_1=\frac{(T_{iw}-T_{ia})-(T_{is}-T_{ia})\left(\frac{a+\alpha 2}{b}\right)e^{l\alpha 2}}{\left(\left(\frac{a+\alpha 1}{b}\right)e^{l\alpha 1}-\left(\frac{a+\alpha 2}{b}\right)e^{l\alpha 2}\right)}$$

Then the final solution for water and solid of temperature distributions are:

$$T_s=T_{ia}-0.84e^{0.012l}+105.84e^{-0.246l} \qquad \ldots\ldots\ (32)$$

$$T_w=T_{ia}-2.31e^{0.012l}+6.985e^{-0.246l} \qquad \ldots\ldots..\ (33)$$

Verification of the Model
Case Study

The above model has been implemented and verified by conducting a full thermal design of a Sodium Bicarbonate powder. A continuous flow of the powder with a mass flow rate of (25) Ton/hr enters at temperature of (130) C to be cooled down to a

temperature of (60) C in a fluidized bed. To accomplish the design of such product in the trough a combined cooling process was used. An immersed tube bundle heat exchanger was installed with water cooling medium supplied from a cooling tower at the rate of (41.8) m^3/hr at entering temperature of (25) C.
The air volumetric flow rate was estimated from the above stated method to accomplish acceptable heat transfer rate. This was chosen to hit both goals of the object, fluidization process and cooling medium at the same time. **Table (1)** shows the characteristics of air, solid and water in this investigation.

Table (1): The variables range of the work

Operational Conditions	Entering Temp. (C°)	Outlet Temp. (C°)	Flow rate (kg/s)	Specific heat (KJ/Kg.C°)
solid	130	60	6.94	1.26
Air	25	60	5.83	1.009
Water	25	36	11.6	4.2

Model Methodology and Results

Results of Solid – Gas Fluidized Bed without Immersed Tubes

In this work the analysis was done for two different diameter of a Sodium Bicarbonate powder, for (0.2mm) and (2mm). In order to get an understanding of the effects of heat parameters in cooling process, the Nusselt number and heat transfer coefficient are analyzed. **Table (2)** shows the results of air and solid fluidized bed in this investigation.

Table (2): Results of air and solid fluidized bed in this investigation without immersed tubes.

$\dot{Q}_{total}$ KW	$\dot{Q}_{losses}$ KW	$\dot{Q}_{net}$ KW	$\dot{Q}_{air}$ KW	h_{mf} m	Ua m/s	ΔPb N	ΔPd N	Number of holes	Distance between holes mm
612	31	581	206	0.68	0.42	2398.4	239.84	36000	18
dp = 0.2 mm									
Re_{mf}	Umf m/s	CD	Ut m/s	Re_p	Nup	h W/m².C°			
0.145	0.0115	0.95	1.45	5.066	0.285	38.475			
dp = 2 mm									
Re_{mf}	Umf m/s	CD	Ut m/s	Re_p	Nup	h W/m².C°			
71.058	0.56	0.95	4.6	50.66	5.9194	79.911			

Results of Solid – Gas Fluidized Bed with Immersed Tubes

The design was furnished by applying the suggested numerical model which implements the step by step technique throughout the trough. Here, the tube bundle was considered in the water flow direction and divided into increments each of (0.5) m. In each portion of the tube the solid and water temperatures were calculated by the use of equations (20.a) and (20.b) respectively, figure (2).

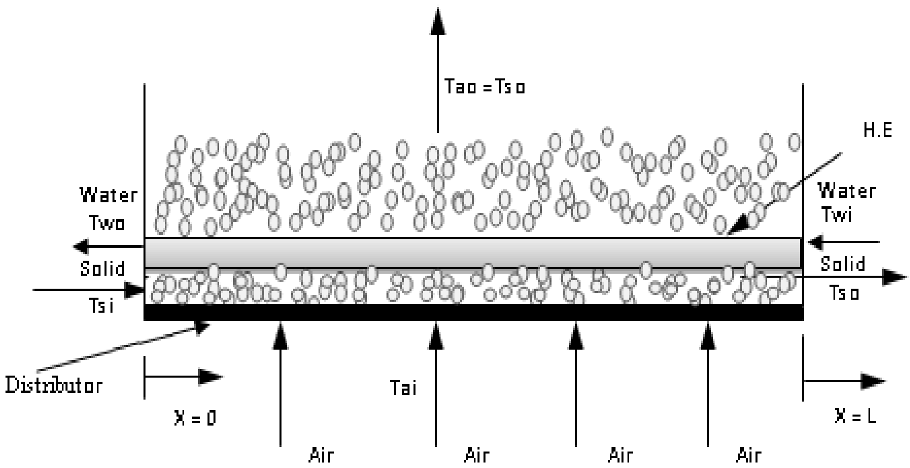

Figure (2): Schematic diagram of the trough to describe the temperature distribution.

It should be noted that all of the parameters incorporated in the calculations such as, h_w, h_s, U, T_s and T_w were updated according to the increment position with respect to water and solid flow. In other words, this technique eliminates the effect of property variation due to the temperature of air or water throughout the fluidized bed. The final results of the model are shown in tables (3 and 4). Here, the air, solid and water temperature distribution is well established by the model. The final verification technique showed the need to install a heat exchanger consisting of (65) tubes with (33.4) mm diameter and (4) m length. It is arranged in a (1) passes along the fluidized bed. The amount of heat removed is about (588) KW and heat excess is about (16) KW. The value of the excess load may be divided into two parts for the air and solid cooling.

Table (3): Results of fluidized bed cooler in this investigation with immersed tubes.

L m	Ts C°	Tw C°	Ta C°	$\dot{Q}_s$ KW	$\dot{Q}_w$ KW	$\dot{Q}_{air}$ KW
0	130	29.7	25			
0.5	116.9	28.8	36.52	114.6	43.85	67.6
1.0	105.4	28	46.60	100.6	39.0	59.1
1.5	95.38	27.35	55.41	87.57	31.67	51.8
2.0	86.6	26.75	63.11	77.12	29.2	45.3
2.5	78.84	26.23	69.83	67.5	25.3	39.6
3.0	72.1	25.77	75.71	58.88	24.4	34.6
3.5	66.13	25.36	80.85	52.21	20.9	30.24
4.0	60.93	25	85.35	44.86	19	26.5
				603	233	354.7
					588 Kw	

Table (4): Analytical model results of fluidized bed cooler with immersed tubes which implements the step by step technique throughout the trough.

Uw m/s	Water flow rate Kg/s	$\dot{V}_w$ m³/h	Rew	Nu)tube	hw W/m².C°	Uo W/m².C°	LMTD	$\dot{Q}_w$ W
0.35	11.6	42	7093	67.6	1536	132	59.72	2613.89

Figure (3) shows the solid and air temperature distribution along the trough. It is obvious that as the powder temperature decreases resulting in an increase for the air temperature towards the exit. The same argument can be applied for the cooling water temperature, figure (4).

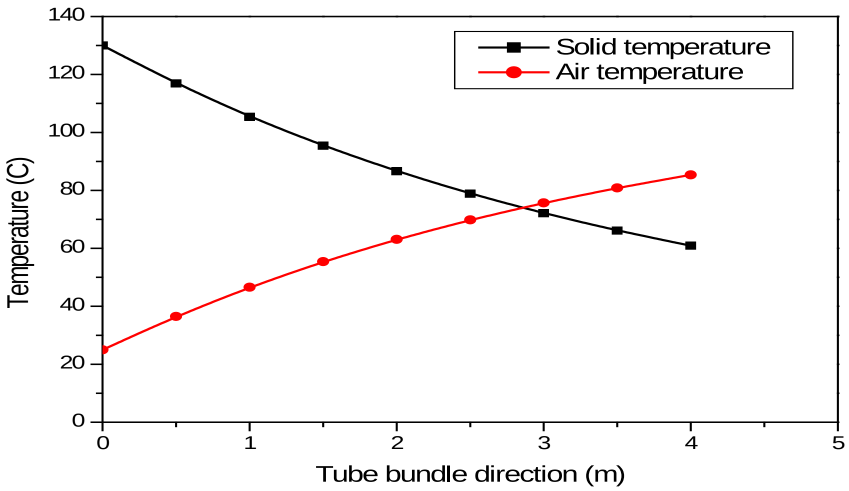

Figure (3): Temperature distribution along tube bundle with the step by step technique throughout the trough.

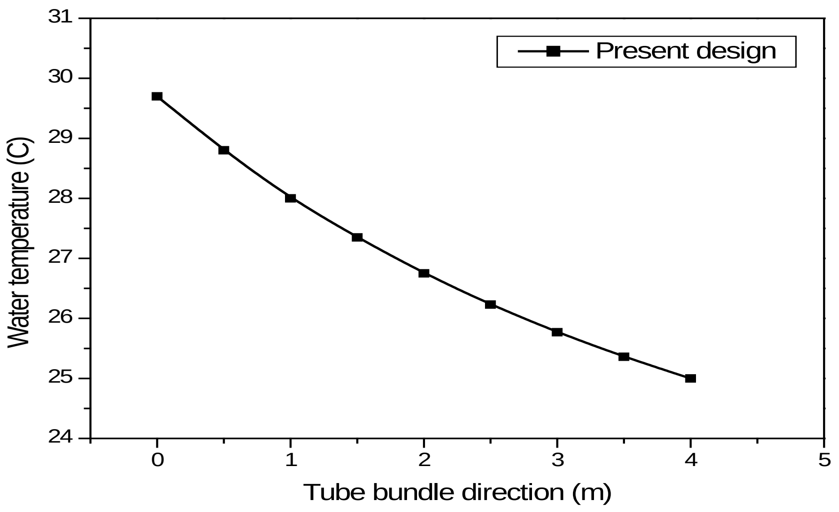

Figure (4): Water temperature distribution along tube bundle with the step by step technique throughout the trough.

Conclusions

The thermal design of the present work can be summarized as:

1- A numerical model has been established for the purpose of thermal design of the fluidized bed cooler. A step by step technique has been constructed to accommodate the idea of using a tube bundle heat exchanger to assist the cooling purpose of the product.

2- A complete thermal design has been conducted with the installation of a designed heat exchanger to be immersed inside the powder trough.

3- The fluidized bed mode for cooling of the powder is a useful tool for heat exchange in the industrial application. It only requires a rough knowledge for the product powder physical dimensions and properties.

4- The velocity of the fluidizing medium in the design of the fluidized bed is importance in the design. Therefore, it should be chosen to satisfy the fluidization condition as well as minimization of the carryover of material. However, in the industrial application of fluidized bed cooler, carry over up to 20% in acceptable.

Nomenclature

A	Area, (m^2)
Ac	Cross section area, (m^2)
Ar	Archimedes number

CD	Drag coefficient
Cp	Specific heat, (kJ/ Kg.C)
dp	Particle diameter , (m)
Do	Outside tube diameter, (m)
ε_{mf}	Fluidized bed porosity
g	Gravitational acceleration , (m/s^2)
h	Heat transfer coefficient, (W/m^2 K)
h_{mf}	Minimum fluidization bed height, (m)
k	Thermal conductivity, (W/m K)
L	Trough length, (m)
$LMTD$	Logarithmic mean temperature difference
m_s	Solid mass, (kg)
$\dot{m}$	Mass flow rate, (kg/s)
Nt	Total Number of tube
Nu	Nusselt number, Dimensionless
Pr	Prandtl number (Dimensionless)
$\dot{Q}_{air}$	Air cooling load, (kW)
$\dot{Q}_{total}$	Total cooling load, (kW)
Re	Reynolds number (Dimensionless)
T	Temperature, (℃)
Δpb	Bed pressure difference, (N/m^2)
Δpd	Distributor pressure difference, (N/m^2)
ΔT	Temperature difference, (deg C)
U_{mf}	Minimum fluidization velocity, (m/s)
uo	Gas velocity, (m/s)
Uo	Overall heat transfer coefficient, (W/m^2 K)
Uor	Velocity through orifice distributor, (m/s)
Ut	Terminal particle velocity, (m/s)

Subscript

a	Air
g	Fluidizing gas
i	Initial value
mf	Minimum fluidization condition
o	Outlet value
p	particle
s	Solid
w	Water

Greek Symbols

Δ	Difference
μ	Fluid viscosity, (Pa.s)

ρ	Fluid density, (kg/m^3)
ϕ_s	Sphericity

5-
6-
7-
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