Proceedings of the
International Conference on Mechanical Engineering 2007
(ICME2007) 29- 31 December 2007, Dhaka, Bangladesh

ICME07-FL-26

DESIGN OF AIR INLET DUCT FOR LABORATORY SCALE FORCED DRAFT AIR-COOLED HEAT EXCHANGERS TO INVESTIGATE THE AERODYNAMICS OF PLUME

Md. Mizanur Rahman

School of Engineering and Information Technology, University of Malaysia Sabah (UMS)
88999, Kota Kinabalu, Sabah, Malaysia.
Email: m92504@gmail.com

ABSTRACT
This paper is mainly focused on the design of a multiple sizes air inlet duct to determine the fluid flow characteristic i.e. velocity and temperature profile to know the effect of face dimension on natural convection of forced draft air- cooled heat exchanger. In the air cooled heat exchanger the average entrainment velocity is used to determine the effective plume-chimney height and hence the buoyancy above the heat exchanger to improve the design and to make the system cost effective. Before going to establish the laboratory facilities and to conduct experiment, the average velocity at constant temperature is calculated from the heat load and the mass flow rate to minimize the plume source size and to maximize for the differential pressure drop of the air inlet duct for ease of measurement in the laboratory. The differential pressure of the air inlet duct will be measured during experiment to determine the average bulk velocity of fluid and existing mathematical model will be readjusted for estimating plume chimney height above a forced draft air cooled heat exchanger operating under natural convection.

Keywords: Air Cooled Heat Exchanger, Natural Convection, Differential pressure

1. INTRODUCTION
In the chemical industries, refineries, power generation industries, waste incineration plants, petrochemical industries, automobile industries and timber treatment industries waste heat is extracted from hot flue gases, chemicals, petroleum products or water that flow through pipes or tube bundles by using air cooled heat exchanger and transfer heat into the ambient air (Meyer and Kroger, 2001 and Garmize et. al., 1994). The air cooled heat exchangers are widely used in the region where the useable water is found very less but source of energy or resources are available for extraction like South Africa, Middle East and North China (Hagens et.al, 2007 and Zhai and Fu, 2006). The main advantage of air cooled heat exchanger is that air can disposed in the atmosphere directly where as other cooling agent like water can not be, due to have high environmental impacts. If the forced draft air cooled heat exchanger operates in natural convection mode, then it can increase operational safety and may lead to the potential savings of fuel cost (Chu, 2002). The main factors are needed to be consider to design a air cooled heat exchanges are tube diameter, tube length, height of fine, number of tube rows, face area and number of passes (Chu, 2002, Vlasov et.al, 2002, Gonzalez, PetraccI and Urbicain, 2001 and Garmize et.al 1994) but the performance of the air cooled heat exchanger either forced draft or natural draft depends on the environmental condition i.e. the cross winds flow and recirculation of air, the inlet air temperature, the wind loading in system and the atmospheric pressure (Zahi and Fu, 2006, Vlasov et.al, 2002 and Garmize et.al, 1994). The fluid flow characteristic such as velocity and the temperature distributions inside the cooling tower is complicated and very difficult to measure under different environmental condition (Su, Tang and Fu, 1999). In addition, the literature review showed that most of the research works have been focused to determine the plume chimney height and dispersion of pollutant to the atmosphere but no literatures are available to describe details about the fluid flow characteristic at the inlet air duct and immediate out let of the chimney or the effect of plume chimney height above the forced draft air cooled heat exchanger operating under natural condition on it enhancement of draft. So, this paper aim to design a inlet air duct for laboratory scale forced draft air cooled heat exchanger to determine the fluid flow characteristic i.e. velocity and temperature profile at the inlet of the air cooled heat exchanger hence to be able to determine the effects of face dimensions on the plume characteristics, rising above the forced draft air cooled heat exchanger.

2. DESIGN DETAILS
To design an air duct assuming that the fluid flow though the inlet is incompressible and isothermal, therefore the mass flow rate at any point of the stream line is constant. The inlet and outlet pressure are equal to

the atmospheric pressure.

2.1 Duct Design

The cross section area of the duct gradually varies with axial distance then the mass conservation for one dimension incompressible mean flow along the axis is

$$d(uA) = 0 \tag{1}$$

or

$$du/u = -dA/A \tag{2}$$

Where u is the velocity and A is the cross sectional area and the total pressure at any point in the duct, can be defined as

$$p_o = p + \frac{1}{2}\rho u^2 \tag{3}$$

Where P is the static pressure and ρ is the density of fluid. The change of pressure along the flow of the duct can be express

$$dP_o = dp + \rho u\, du \tag{4}$$

Put the value of *du* from equation 2

$$dP = dp_o + \left(\rho \frac{u^2}{A}\right) dA \tag{5}$$

$$du = {-[dp - dp_o]}/{(\rho u)} \tag{6}$$

For small wall curvature the pressure P will be nearly constant across a cross section of the duct. Consider the friction loss in the duct is very small and no energy transfer in the duct then the P_o will be constant throughout the flow. The differential pressure though the duct will be

$$dP = \left(\rho \frac{u^2}{A}\right) dA \tag{7}$$

$$du = {-[dp]}/{(\rho u)} \tag{8}$$

In the convergent ducts, increase in the flow velocity and corresponding decrease in the flow static pressure where as in the diverging ducts decrease in the flow velocity and corresponding increase in the flow static pressure along with the axis of the duct. Therefore the diverging duct is used at the down steam of the air cooled heat exchanger.

Different geometrical shape diffusers are commonly used but the flow phenomena encountered in virtually all types of diffusers are qualitatively similar. Without this for small values of 2θ, the boundary layers remain attached (on the average) along the entire length at the walls, and the flow is generally steady (Schetz and Fuhs, 1996).

2.2 Inlet Design

The velocity at the entrance of a circular duct is flat and changed with axis due to the effect of wall where the velocity of fluid is zero. The fluid flow a certain length and a core region that developed due to the entrance effect is completely eliminated. The length is known as entrance length (L_e) that depends on the flow characteristic.

$$\frac{L_e}{D} = 0.06\,\mathrm{Re} \quad \text{(Laminar flow)}$$

$$\frac{L_e}{D} = 0.06\,\mathrm{Re}^{1/6} \quad \text{(Turbulent flow).}$$

Where, Re is the Reynolds number, a dimension less number is used to determine the flow characteristic. $\mathrm{Re} = \frac{\rho u D}{\mu}$ Where, u is the average velocity of the flow, D is the characteristic dimension of the circular cross section, ρ is the fluid density and μ is the absolute viscosity of the fluid. For non- circular cross section, the Reynolds number equation is defined as the hydraulic diameter D_h

$$D_h = \frac{4A}{p_e} = \frac{4 \times Flow\,Area}{Wetted\,Perimeter}$$

If the value of Reynolds number is less than 2000 the flow is usually laminar and is greater than 4000, the flow is usually turbulent. In between the flow can be exhibited as laminar or turbulent characteristic, but it is very difficult to accurately predict its behavior (Johnson, 1998).

2.3 Velocity and Differential Pressure Calculation

In the most commercial operation fluids are flowed in pipes or in tubes. The fluid velocity depends on the dimension of the duct.

Velocity:

Consider, loss due to the roughness is negligible and mass $\dot{m}$ is flowing through the air duct (figure 1) at constant ambient temperature, then $\dot{m} = \rho A u = \rho Q$. The velocity at point 1 and 2 will be same because same amount of fluid flowing though the cross section $A_1 = A_2$ but the velocity at point 2 and 3 will be changed because of it's cross sectional area is changed.

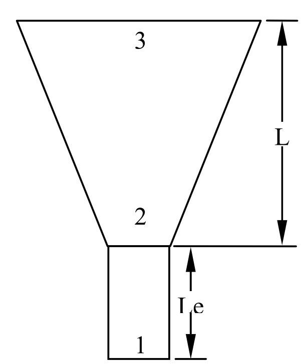

Fig :1 Air flow duct for experiment

Mass flow through point 2 and 3

$$\dot{m}_2 = \dot{m}_3 \text{ or } \rho A_2 u_2 = \rho A_3 u_3$$

$$\therefore\ u_3 = \frac{\rho A_2 u_2}{\rho A_3} = \frac{u_2 d_2^2}{d_3^2}$$

Where d is diameter of that particular point.

Differential Pressure:

The differential pressure of fluid depends on pipes or tube shapes and it roughness. It also depends upon the flow characteristic either laminar or turbulent. For laminar flow and a circular duct, the friction factor is $f = {}^{64}\!/_{\mathrm{Re}}$ and $f = {}^{96}\!/_{\mathrm{Re}}$ (non-circular duct). Moody diagram is used to determine the friction factor in turbulent flow and in the diagram the factor varies roughly from 0.008 to 0.1. In addition, the chen, the Churchill and the Haaland equation can be used to determine the friction factor for turbulent flow.

The modified Bernoulli equation for the fluid flow through air duct consider the friction factors

$$\frac{p_1}{\rho g} + \frac{u_1^2}{2\times g} + z_1 = \frac{p_2}{\rho g} + \frac{u_2^2}{2\times g} + z_2 + \frac{fl}{D}\frac{u^2}{2g}$$

Where z is the elevation to the duct centerline from some reference plan, f is the Darcy Weisbach friction factor and d is a characteristic dimension of the cross section (Kreith, (1999), Johnson, 1998, Banga, Sharma and Manghnani, 1994, Khurmi, 1994, Streeter, 1971 and Lewitt, 1963).

2.4 Minor loss

The pressure loss also found due to the change the direction of the flow, known as minor loss and calculate by using *a loss coefficient or an equivalent length (*L_{eq}). The pressure drop due to minor loss is

$$\Delta P = \frac{fL_{eq}\rho u^2}{d\ 2} \text{ and}$$

the Bernoulli equation is rewritten as

$$\frac{p_1}{\rho g} + \frac{u_1^2}{2\times g} + z_1 = \frac{p_2}{\rho g} + \frac{u_2^2}{2\times g} + z_2 + \frac{fl}{d}\frac{u^2}{2g} + \sum k\frac{\rho u^2}{2}$$

Where k is the minor loss coefficient. Alternatively

$$\frac{p_1}{\rho g} + \frac{u_1^2}{2\times g} + z_1 = \frac{p_2}{\rho g} + \frac{u_2^2}{2\times g} + z_2 + \frac{f\,l_t}{d}\frac{u^2}{2g}$$

where l_t is the pipe length plus the equivalent length of all the fittings (Johnson, 1998, Banga, Sharma and Manghnani, 1994 and Khurmi, 1994).

3. EXPERIMENTAL SETUP

The experimental setup is shown in the Figure -2. The numbers are mentioned in the figure present below to discuss the configuration of the inlet air duct.

An electric heater will be used as source of heat on the other sense the plume will be created by a coil type electric heater. The capacity of the electric heater depends on the heat load calculated from the equation developed by Chu (2002) for estimating the effective plume chimney height above a forced draft air cooled heat exchanger operated under natural convection. From the literature it is found that face area 0.457m x 0.457m does not have any significant effect on plume height but this size is supported by industrial size heat exchanger of 2.0m x 3.1m. In addition, the effective plume chimney height (h_0) for laboratory size air cooled heat exchanger is very small Therefore, for laboratory scale the outlet diameter of the duct or diffuser will be considered 0.75m, 1m, 1.5m and 2m and the dimension of the heat source will be changed respectively. To get the temperature profile inside the duct, K type 36"L, 24"L and 12"L thermocouple will be used and data will be collected through USB Data Acquisition Modules. The differential static pressure will be measured to calculate the average velocity of fluid inside the duct by a differential room pressure transducer. The range of the transducer is fixed -0.5" to 0.5" of water and resolution is 0.001" of water. Air velocity will also be determined by using VelociCalc Air velocity meter. The range of this instrument is 0 to 30m/s and the accuracy is ±0.015m/s.

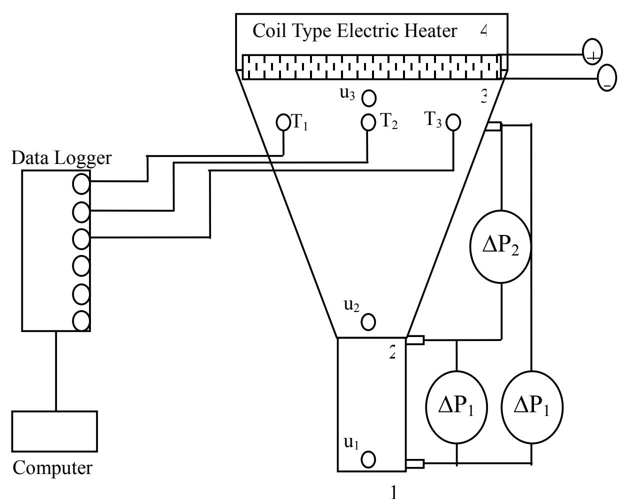

Figure 2: Experimental Configuration for Inlet Duct

4. DISCUSSION AND EXPECTED OUTCOMES

Consider air flowing through the duct at 303^0K and the air temperature difference between the inlet and the outlet of the of the heat exchanger is 50^0K.

The Grashof Number is $Gr = \dfrac{L^3 g_n \beta \Delta T \rho^2}{\eta^2}$

Where, L= Length of the Square Plate =0.75m,

g_n = Gravitational Constant $= 9.81\,{}^{m}\!/_{s^2}$,

β = Volumetric Coefficient of Expansion

$= {}^{1}\!/_{K} = {}^{1}\!/_{353} = 0.0028$,

ρ = Density of Fluid Flow though the Duct

$= 1.000\,{}^{kg}\!/_{m^3}$

η = Dynamic Viscosity $= 0.0000186\,{}^{N.s}\!/_{m^2}$

$\therefore\ Gr = 1.69\times 10^9$

The Prandtl Numbe is

$$\Pr = \frac{c_p \times \eta}{\lambda} = \frac{1005 \times 0.0000186}{0.0264} = 0.71$$

$$Gr \times \Pr = 9.11 \times 10^8 \times 0.7 = 1.20 \times 10^9$$

which is greater than 10^9 Entrainment velocity

$$u_e = \sqrt{\frac{2 \times 10.12 \times \left(L^{0.6} \times g_n \times \beta \times \Delta T \times \rho^{0.8} \times \eta^{0.4}\right)^{0.83}}{\rho}}$$

$= 0.78\, m/s$

The effective plume chimney height

$$h_0 = \frac{\rho \times u_e^2}{(\rho_a - \rho) \times g_n} = 0.19\, m$$

So, Mass flow rate $\dot{m} = \rho \times A \times V \ \ kg/s$ and heat load = $\rho \times A \times V \times c_p \times \Delta T$ =17,464.32 j/s

Consider that air only flow through duct, then the entrainment velocity will be same as maximum velocity at point 3, So, average velocity at u_3 = 0.79 m/s. Now Consider, the inlet diameter is 0.4m the average velocity at point 2 and 1 (u_2 and u_1) will be 2.77m/s and the differential pressure due to diffuser and inlet pipe, between point 1 and 3 will be 2.46 N/m^2.

Similarly velocities and pressure difference are calculated for different estimated temperature difference and presented in the Table 1.

Table 1: Calculated Velocity and Differential Pressure for Different Heat Load (Chu, 2002).

Estimated Diameter	Estimated Inlet Temperature	Estimated Temperature Difference	Calculated Heat Load	Calculated Velocity at 3	Calculated Velocity at 1 & 2	Calculated Differential Pressure 1 to 3
m	^{0}K	^{0}K	J/s	m/s	m/s	N/m^2
0.75	303	50	17464.32	0.79	2.77	2.46
0.75	303	60	21840.78	0.84	2.96	2.82
0.75	303	70	26238.05	0.89	3.14	3.17
0.75	303	80	30685.73	0.94	3.30	3.50
0.75	303	90	35086.63	0.98	3.44	3.81
0.75	303	100	39481.84	1.02	3.57	4.11
0.75	303	110	43788.48	1.05	3.69	4.40
0.75	303	120	47165.98	1.09	3.82	4.71

Table 2: Velocity and Differential Pressure calculated Chu 2002 and Byram, 1974 and.

Estimated ΔT	Chu 2002				Byram, 1974			Velocity Change
	Calculated Velocity at 3	Calculated Velocity at 1&2	Calculated ΔP_{1-3}	Calculated Heat Load	Calculated Velocity at 3	Calculated Velocity at 1&2	Calculated ΔP_{1-3}	
^{0}K	m/s	m/s	N/m^2	J/s	m/s	m/s	N/m^2	%
50	0.70	2.77	2.46	17464.32	0.67	2.37	1.71	4.29
60	0.84	2.96	2.82	21840.78	0.70	2.46	1.85	16.67
70	0.89	3.14	3.17	26238.05	0.72	2.54	1.96	19.10
80	0.94	3.30	3.50	30685.73	0.74	2.60	2.06	21.28
90	0.98	3.44	3.81	35086.63	0.75	2.64	2.12	23.47
100	1.01	3.57	4.11	39481.84	0.76	2.67	2.18	24.75
110	1.05	3.69	4.40	43788.48	0.77	2.70	2.22	26.67
120	1.09	3.82	4.71	47165.98	0.76	2.66	2.16	30.28

Byram (1974) analysis the fire source as a pure heat source and calculate velocity before entering and after the heat source. Velocity of air before entering the heat source will be $u_3 = \left(\frac{T_2}{T_1 - T_2}\right)\frac{I_a}{\rho_2 C_p T_2}$ where I_a is the heat output per unit area, depends on the combustion of fuel of composition $C_xH_yO_z$ but for the above mentioned design the value is calculated from the preliminary method for estimating the effective plume chimney height above a forced draft air cooled heat exchanger operating under natural convection (Chu, 2002). The calculated velocities form Byram (1974) and Chu (2002) are nearly same that presented in Table 2.

5. CONCLUSIONS

To determine the effects of face dimension on plume rising above the heat exchanger, average velocities calculated from Chu (2002) and found that at point 3 varies from 0.70m/s to 1.09m/s and at point 1 & 2 varies 2.77m/s to 3.82m/s when the inlet and outlet dimension is fixed 0.4m and 0.75m respectively and temperature difference will be lie on 50 to 120^0K. The calculated differential pressure between inlet and outlet is found 2.46N/m^2 to 4.71N/m^2. On the other hand by using Byram (1974) developed equation, the calculated velocity at point 3 varies from 0.67m/s to 0.76m/s and at point 1 & 2 varies 2.37m/s to 2.66m/s for the same heat load that calculated from buoyancy. The calculated differential pressure between inlet and outlet is found 1.71N/m^2 to 2.16N/m^2.

Principally, the air flow rate can be measured and mathematical model will be develop or modify previously developed model after establish the lab facilities and measuring the pressure drop in a special inlet air duct. It is expecting that the outcome of this research will eliminate the recirculation and cross wind effects by considering different diameters of inlet ducts for each face dimension (plume source dimensions). An experiment is underway for the research facility to be manufactured.

6. ACKNOWLEDGEMENT

The author would like to give thanks to the School of Engineering and Information Technology, Universiti Malaysia Sabah to provide financial support and to give the opportunity to develop the research facilities.

7. REFERENCES

1. Banga, T.R., Sharma, S.C. and Manghnani, T. (1994) "Mechanical Engineering Hand Book", Classical Printers, Chawri Bazar, Delhi, 3rd Edition, pp- 460 – 466
2. Chu, C. M. (2002),"A Preliminary Method for Estimating the Effective Plume Chimney Height above a Forced Draft Air Cooled Heat Exchanger operating under natural Convection", Heat Transfer Engineering, Vol. 23, No. 3, 2002 pp-3-12
3. Garmize, L. Kh., Dashkov, G.V. Solodukhin A.D. and Fisenko S.P. (1994), "Laboratory Modeling of the Enhancement of Heat and Mass Transfer Processes in Chimney – Type Evaporative Cooling Tower", Journal of Engineering Physics and Thermophysics, Vol. 66, No. 2, 1994, pp -126 -132
4. Gonzalez, M.T, PetraccI, N.C. and Urbicain M.J (2001), "Air Cooled Heat Exchanger Design using Successive Quadratic Programming (SQP)" Heat Transfer Engineering, Vol. 22, No.3, 2001
5. Hagens, H, Ganzevles F. L. A, Van Der Geld, C. W.M and Grooten, M.H.M, "Air heat exchanger with long heat pipes: experiments and predictions", Applied thermal engineering, 27 (2007) 2426-2434
6. Johnson, R.W., (1998), "The Hand Book of Fluid Dynamics" CRC press LLC, USA & Springer Verlag, GmbH & Co. KG, Germany, pp-5-62 -5-89
7. Khurmi, R.S (1994), "A Text Book of Hydraulics Fluid Mechanics and Hydraulic Mechanics", S Chand & Company Ltd., New Delhi, India pp-409-429.
8. Kreith, F (1999), "Mechanical Engineering Hand Book", CRC net BASE, 1999 pp -3.47 – 3.50
9. Lewitt, E.H. (1963), "Hydraulics and Fluid Mechanics" The English Language Book Society and Sir Isaac Pitman & Sons Ltd, London, 10th Edition, pp 43-45
10. Mayer, C. J. and Kroger, D. G., (2001), "Air Cooled Heat Exchanger inlet Flow Loss", Applied Thermal Engineering, 21 (2001) pp-771 - 786
11. Schetz, J. A. and Fuhs, A. E., (1996), Hand Book of Fluid Dynamics and Fluid Mechinary, A Wiley inter-science publication, John Wiley & Sons Inc, New York, Chi Chester, Brisbane, Toronto, Singapore pp515 – 525
12. Streeter, V. L. (1971), "Fluid Mechanics", Fith Edition", International Student Edition, Mcgraw hill Kogakusha, Ltd. Singapore pp- 360-363
13. Su, M. D., Tang, G. F. and Fu, S., (1999) "Numerical Simulation of Fluid Flow and Thermal Performance of a Dry-Cooling Tower Under Cross Wind Condition" Journal of Wind Engineering and Industrial Aerodynamics, vol, 79 (1999) 289 – 306
14. Vlasov, A. V., Dashkov, G.V., Solodukhin A.D. and Fisenko S.P. (2002), "Investigation of the Internal Aerodynamics of the Chimney Type Evaporative Cooling Tower", Journal of Engineering Physics and Thermophysics, Vol. 75, No. 5, 2002, pp -1086 -1091
15. Zhai, Z. and Fu, S. (2006), "Improving cooling efficiency of dry-cooling towers under cross-wind conditions by using wind-break methods" Journal of Applied Thermal Engineering, vol-26 (2006), pp-1008–1017