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Characteristics of Drift Eliminators of an Evaporative Condenser

A.K. SINGH, A.K. M. MOHIUDDIN AND KESHAV KANT

Mechanical Engineering Department, Indian Institute of Technology, Kanpur - 208 016, India

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Abstract

This paper deals with an experimental study of Drift Elininators used in Evaporative Condensers to determine the pressure drop across them, drift loss and their effects on a refrigeration system performance using an evaporative condenser as one of its components. The experiments were conducted with single and double stages of both wooden drift eliminators (WDE) and concrete drift eliminators (CDE) using both forced and induced draft (ID). The orientation of the drift eliminator plates was varied from 15° to 90° with horizontal for a single fan speed. Based on the cost analysis for the variation of the pressure drop and the drift loss, an optimum angle of inclination of the drift eliminator plates is suggested. The COP of the refrigeration system increased with increase in fan speed and angle of inclination and decreased with increase in the number of stages. The use of forced draft resulted in better performance of the refrigeration system than with induced draft.

Introduction

In evaporative cooling, water is sprayed into flowing air and the temperature of the water is reduced because of the evaporation of water and subsequent absorption of latent heat by the air stream. In case of evaporative condensers, the water which is sprayed over the condenser tubes takes away some heat and the air whose temperature is slightly increased due to the evaporation of water droplets, also removes some heat from the condenser tubes. The hot water past the condenser tubes is cooled by the process of adiabatic saturation.

Drift eliminators (DE) form an integral part of an evaporative condenser or a cooling tower. Typical DE designs embody an array of narrow channels in the vertical flow direction, which force the air to flow in a direction inclined to the vertical. The efficiency of DE is governed by the channel orientation and geometry, air flow rate and the droplet size distribution (Grimble & Roffman, 1973).

Drift has been traditionally defined as mechanically entrained water droplets which are generated inside the tower and carried along with the air flowing through the tower exhausted to the environment (Wistrom & Ovard, 1973). As the air moves counter to or cross-wise to the flow of water, it will pick up much of the mist and droplets and carry them with the air stream out of the cooling tower or the evaporative condenser. These droplets contain dissolved salts in roughly the same concentration as in the cooling water. Elimination if possible or reduction of drift is essential as it can corrode and damage structures in the immediate vicinity of the towers, cause a public nuisance if located near parking lots or high density traffic areas and endanger local vegetation.

In recent years, the demand for industrial water has increased and at the same time, the sources of raw water are limited. To tackle this problem, we need an efficient mechanical system to retain the water since a water distribution system turns the process water into tiny droplets before they are cooled by the air stream. One way of tackling this problem of drift loss is by providing drift eliminators above the water distribution system in a cooling tower or the evaporative condenser. The DE are typically rectangular plates which are set at the desired angle to the air flow direction over the whole exit area of the air flow passage. By adjusting the angles of these plates with the flow direction suitably, most of the water droplets can be made to fall back into the sump, thereby reducing the drift loss.

Cooling tower performance (i.e., heat removal) is a balance between water flow and air volume. Therefore, DE are normally designed to be efficient through a calculated range of air flow. Too great an air speed can result in excessive drift loss of water from the tower, while poorly designed DE will adversely affect the performance of the unit. Thus DE effectiveness is an essential aspect of CT design for many reasons, among them a few are (Burger, 1975):

(i) conservation of water

(ii) retention of chemicals used for the treatment of water in the sump

(iii) prevention of staining by chemical additives e.g., chromates etc.

(iv) avoiding fan blade corrosion in case of induced draft tower.

(v) avoidance of violation of local area environmental protection regulations

The present study concerns with the performance characteristics of multistage wooden as well as concrete drift eliminators with forced and induced flow separately; for an evaporative condenser and their effect on the performance of a refrigeration system.

Test Rig

The complete test rig consists of an evaporative condenser using a FD as well as an ID fan. The FD fan and motor assembly is installed on a concrete foundation specially made for it. The ID fan is mounted on a raised platform supported on two 'L' shape grounded C.S. pipes. The DE used are of two types: wooden and concrete. The evaporative condenser forms one of the components of a R-22, 1.5 ton refrigeration system. The main components of the Test Rig are shown in Fig. 1. Description of main chamber and DE is given below:

a. Main Chamber

Main chamber consists of (1) a rectangular box (1.00 m x 0.90 m x 1.40 m), (2) a DE chamber (1.00 m x 0.52 m x 0.52 m) having top portion tapered and (3) a heater box (0.45 m x 0.45 m x 0.45m). The heater box houses six finned electric duct heaters of 1 kW

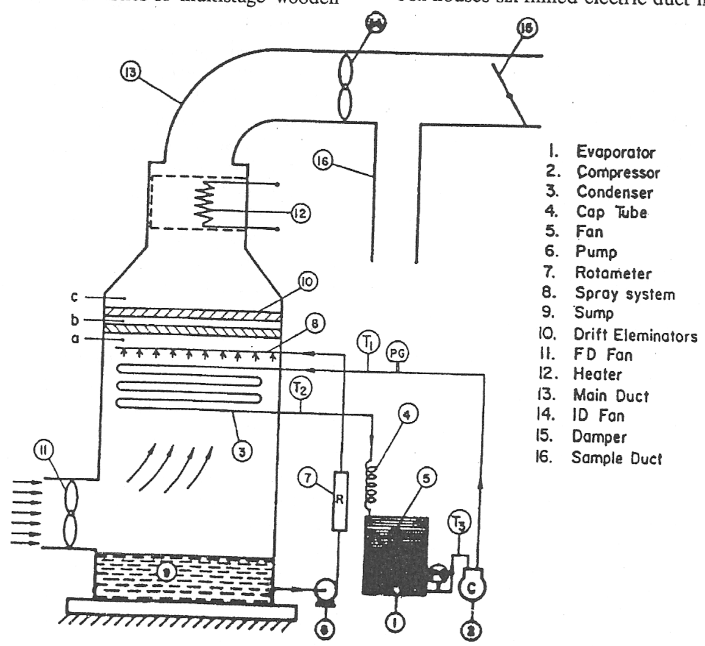

Fig. 1: Schematic diagram of the test rig

capacity each on top of the DE chamber. The bottom portion of the main chamber was used as a water sump.

b. Drift Eliminators

Two types of drift eliminators were used namely (i) Wooden drift eliminators (WDE) and (ii) Concrete drift eliminators (CDE). Strips of the DE were housed in a box of area 0.95 m x 0.495 m and they could be rotated to change the angle of orientation.

(i) Wooden Drift Eliminators

Thickness of each strip = 13 mm

Length of each strip = 900 mm

Number of strips in a single stage = 9

Width of each strip = 46 mm

(ii) Concrete Drift Eliminators

Thickness of each strip = 25 mm

Length of each strip = 900 mm

Width of each strip = 50 mm

Number of strips in a single stage = 7

A main duct is connected at the top of the heater box to the inlet of the ID fan. The discharge duct is connected at the outlet of ID fan to carry the discharge air out of the room. Circulating water pump is installed to provide the spray water over the condenser coil through a spray system consisting of 4 rows of 12.7 mm conduit pipe with holes of 2mm dia. size.

Estimation of evaporation loss

This is the amount of water loss during the cooling process. A simple mass balance of dry air and water over the evaporative condenser is given by

$$\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a \ (say) \tag{1}$$

$$\dot{m}_{a1} w_1 + \dot{m}_e + \dot{m}d = \dot{m}_{a2} w_2 \tag{2}$$

Eqs. (1) and (2) yield

$$\dot{m}_e = \dot{m}_a(w_2 - w_1) \tag{3}$$

where w_1 and w_2 are the humidity ratios which can be easily determined knowing the DBT and WBT of the entering and leaving air. The mass flow rate of the dry air can be calculated knowing w_1, w_2 and the rating of the fan (Das, 1988).

Estimation of the drift loss

In order to determine $\dot{m}_d$, one possible method is to evaporate all the drift going past the DE and then measuring the psychrometric condition of the outgoing air (Golay et al., 1986). If this method is used, a simple mass balance over the evaporative condenser yields:

$$\dot{m}_a w_1 + \dot{m}_e + \dot{m}_d = \dot{m}_a w_a \tag{4}$$

$$\text{or,} \quad \dot{m}_e + \dot{m}_d = \dot{m}_a \ (w_3 - w_1) \tag{5}$$

Eqs. (3) and (5) yield, $\dot{m}_d = \dot{m}_a \ (w_3 - w_2)$ (6)

$$\text{and } \frac{\dot{m}_d}{\dot{m}_e} = \frac{w_3 - w_2}{w_2 - w_1} \tag{7}$$

The quantities $\dot{m}_e$ and $\dot{m}_d$ can be determined by measuring the psychrometric data of the entering and the leaving air streams (Das, 1988).

Procedure

The angle of inclination (θ) was varied from 15° to 90° with horizontal. The number of DE stages used were one or two at a time. The DBT and WBT were measured for the air entering and leaving the evaporative condenser without and with duct heater. At the time of recording the data, the damper of the discharge main duct was kept closed. Psychrometric data with FD fan using one and two stages of CDE and WDE were recorded. Similar data were taken with ID fan (Singh, 1989). The flow rate of air through the unit was changed by changing the speed of the fan. The pressure drop data were also recorded for varying θ and air flow rate, $\dot{m}_a$ with one and two stages of the WDE and CDE using ID fan and then FD fan. The pressure drops across single and double stages were recorded along with the power drawn by the fan in operation. For each set of data (see Table 3), the power input to the refrigerant compressor, pressures and temperatures at suction and discharge were measured to be able to compute the COP of the refrigeration system (Singh, 1989).

Results and Discussions

(a) Drift Loss

The values of M_e and M_d computed from the data are shown in Tables 1 and 2 for the CDE corresponding to various air flow rates. Similar data are available for WDE (Singh, 1989). It can be seen from these tables that the specific drift loss increases if n decreases or θ increases. This is basically due to the fact that in either of the two cases (i.e., decreasing n or increasing θ), the net static pressure available for the flow increases which results in a higher volumetric discharge. The greater discharge of air brings larger amount of air in direct contact with water resulting in a larger value of M_e. The drift loss as expected reaches a maximum for $\theta = 90^o$. As the air flow rate increases, the drift loss also goes up. It is seen from the tables that drift loss is more with FD fan than with ID fan of the same capacity. It is also observed from the data that the drift loss with CDE is 20-25% smaller than that obtained using WDE with either FD or ID fan. This difference may be attributed to the larger thickness of the CDE strips (25 mm) compared to that of WDE (13 mm).

(b) Pressure Drop

The pressure drop (Δp) data recorded for CDE are also shown in Tables 1 and 2. It can be seen that as θ increases, Δp across a particular set of stages decreases with minimum value corresponding to $\theta = 90^o$. It can also be seen that as the number of stages increase, Δp increases which in turn requires a larger amount of power input to the fan. It is also noticed that Δp is more with WDE than with CDE. Also the Δp value across any set of DE was more

Table 1: Drift loss and pressure drop data for CDE with FD fan

Air Flow Rate for $\theta=90^\circ$ m^3/min	Angle of Incln θ deg.	No. of Stages n	Evap. Loss $M_e \times 10^3$ kgw/kgda	Drift Loss $M_d \times 10^3$ kgw/kgda	Static Pressure P_a mm of H_2O	P_b mm of H_2O	P_c mm of H_2O	Pressure Drop Δp mm of H_2O	Fan Power P_{fan} kW
61	15	1	3.3	0.4	15.49	14.22	-	1.27	0.47
		2	4.3	0.3	15.75	-	13.72	2.03	0.48
	45	1	3.8	1.0	16.00	14.86	-	1.14	0.50
		2	4.5	0.7	15.24	-	14.22	1.02	0.48
	60	1	4.6	1.1	14.73	14.29	-	0.44	0.46
		2	4.2	1.0	14.73	-	13.72	1.01	0.46
	90	1	3.4	2.2	14.61	14.35	-	0.26	0.47
		2	3.5	1.9	14.73	-	14.61	0.12	0.46
53	15	1	4.0	0.3	15.24	14.22	-	1.02	0.43
		2	3.8	0.4	15.24	-	13.20	2.04	0.44
	45	1	4.0	0.6	15.24	14.35	-	0.89	0.45
		2	3.1	0.6	15.24	-	14.22	1.02	0.44
	60	1	3.6	1.2	15.24	14.48	-	0.76	0.46
		2	3.6	0.8	14.86	-	13.97	0.89	0.44
	90	1	3.6	2.0	14.98	14.73	-	0.25	0.45
		2	3.7	1.7	14.48	-	14.22	0.26	0.44
46	15	1	4.1	0.2	14.48	13.72	-	0.76	0.42
		2	3.8	0.2	14.48	-	12.95	1.53	0.43
	45	1	3.2	0.4	13.97	13.34	-	0.63	0.42
		2	4.4	0.5	13.97	-	13.21	0.76	0.43
	60	1	3.6	0.5	13.87	13.46	-	0.41	0.42
		2	2.2	0.7	14.22	-	13.59	0.63	0.43
	90	1	3.4	1.0	15.24	14.99	-	0.25	0.43
		2	4.2	1.0	13.46	-	13.20	0.26	0.42

Table . 2: Drift loss and pressure drop data for CDE with ID fan

Air Flow Rate for θ=90° m^3/min	Angle of Incln θ deg.	No. of Stages n	Evap. Loss $M_e \times 10^3$ kgw/kgda	Drift Loss $M_d \times 10^3$ kgw/kgda	Static Pressure P_a mm of H_2O	Static Pressure P_b mm of H_2O	Static Pressure P_c mm of H_2O	Pressure Drop Δp mm of H_2O	Fan Power P_{fan} kW
46	15	1	3.2	0.6	8.13	9.27	-	1.14	0.38
		2	3.8	0.4	8.13	-	9.70	1.57	0.40
	45	1	3.0	1.0	8.13	9.14	-	1.01	0.40
		2	2.4	0.6	8.38	-	9.14	0.76	0.41
	60	1	2.4	1.6	8.51	8.89	-	0.38	0.41
		2	2.0	1.4	8.76	-	9.27	0.51	0.42
	90	1	2.8	2.0	8.76	9.14	-	0.38	0.43
		2	3.6	1.8	8.76	-	8.89	0.13	0.38
41	15	1	8.2	0.5	8.64	9.40	-	0.76	0.37
		2	2.8	0.3	8.38	-	9.65	1.27	0.38
	45	1	2.4	0.8	8.26	8.89	-	0.63	0.36
		2	3.4	0.5	8.13	-	8.89	0.76	0.37
	60	1	1.6	1.3	8.13	8.64	-	0.51	0.38
		2	3.4	1.0	8.58	-	8.89	0.31	0.38
	90	1	3.2	1.8	8.69	8.89	-	0.20	0.40
		2	3.6	1.6	8.51	-	8.64	0.13	0.36
38	15	1	4.2	0.4	8.26	8.89	-	0.63	0.35
		2	3.0	0.3	8.13	-	9.14	1.01	0.35
	45	1	3.7	0.7	8.13	8.64	-	0.51	0.33
		2	4.0	0.4	7.87	-	8.64	0.77	0.34
	60	1	3.9	0.4	8.38	8.63	-	0.25	0.35
		2	4.0	1.0	8.12	-	8.38	0.26	0.34
	90	1	3.9	1.7	8.38	8.38	-	0.00	0.37
		2	2.8	1.2	8.26	-	8.38	0.12	0.34

with FD fan than with ID fan. This may be attributed to a significant amount of leakage of air which was inherent in the test rig because of improper sealing of the joints of the main box ducts resulting in a smaller amount of flow rate through the DE.

(c) COP of the Refrigeration Unit

The system performance data are given in Table 3. It can be seen here that the trend is similar for all the cases and also as the value of θ increases, the COP for the system also increases. This is because as the θ increases, the volumetric discharge of air through the system increases which results in higher evaporation rate of water thus causing more cooling of water. This cold water comes in contact with the condenser coil and results in subcooling of the refrigerant and lowering of the condenser pressure. The COP of the system becomes smaller for lower air flow rate. When the number of stages increases, the net effect is a drop in COP due to increased resistance to flow and resulting lower air discharge rate. COP of the system is found to be higher with FD fan than with ID fan. This can be explained on the basis of greater degree of turbulence created by the FD fan in the evaporative condenser box as compared with ID fan and this leads to higher rate of heat transfer from the condenser coils.

(d) Optimum Angle of Orientation

For determination of an optimum angle of orientation, θ_0 for DE, a running cost analysis was carried out. The main factors considered were power loss due to the pressure drop across the DE and the amount of water lost to the atmosphere in the form of drift. It is found that for a particular value of θ for a given case, the values of the cost of drift loss and that of power loss become equal, i.e., the cost of water lost to the atmosphere per hour in the form of drift is same as the cost of power lost to overcome the pressure drop. This gives the optimum value of

Table. 3: Refrigeration system performance using evaporative condenser with WDE and CDE

Type of DE	Air Flow Rate for $\theta = 90^\circ$ m^3/min	Angle of Incl. θ deg.	No. of Stages n	FD Fan		ID Fan	
				Compressor Power Input $P_{comp.}$ W	COP	Compressor Power Input $P_{comp.}$ W	COP
WDE	61 (FD) 46 (ID)	15	1	1500	3.876	2020	2.010
			2	1600	3.890	1950	2.081
		45	1	1580	2.679	1780	2.076
			2	1540	3.830	1900	2.082
		60	1	1595	2.866	1800	2.286
			2	1780	2.547	1700	2.354
		90	1	1540	4.115	1900	2.701
			2	1500	4.205	1880	2.700
	53 (FD) 41 (ID)	15	1	1590	3.070	1950	1.944
			2	1600	3.160	1915	1.906
		45	1	1580	2.819	1730	2.026
			2	1640	2.950	1820	1.954
		60	1	1595	2.807	1810	2.010
			2	1690	2.650	1700	2.189
		90	1	1580	3.325	1950	2.356
			2	1670	3.350	1940	2.346
	46 (FD) 38 (ID)	15	1	1625	2.949	1985	1.906
			2	1650	2.909	1950	1.981
		45	1	1550	2.688	1725	2.032
			2	1590	3.166	1800	2.000
		60	1	1590	2.506	1810	2.027
			2	1650	2.736	1695	2.119
		90	1	1580	3.202	1990	2.001
			2	1590	3.200	2000	2.040
CDE	61 (FD) 46 (ID)	15	1	1660	2.478	1680	2.471
			2	1610	2.414	1600	2.518
		45	1	1750	2.842	1800	2.052
			2	1640	2.486	1690	2.120
		60	1	1750	2.222	1800	2.151
			2	1610	2.700	1700	2.357
		90	1	1600	2.617	1950	2.087
			2	1550	3.059	1670	2.245
	53 (FD) 41 (ID)	15	1	1650	1.962	1950	2.023
			2	1600	1.964	1660	2.008
		45	1	1770	2.122	1790	2.085
			2	1650	1.965	1660	2.381
		60	1	1720	2.083	1810	2.067
			2	1650	1.968	1680	2.132
		90	1	1620	2.328	1700	2.304
			2	1560	2.677	1600	2.510
	46 (FD) 38 (ID)	15	1	1700	2.231	1950	1.956
			2	1650	2.088	1710	2.066
		45	1	1750	2.134	1800	2.002
			2	1650	2.194	1700	2.034
		60	1	1710	1.963	1950	1.989
			2	1650	2.230	1700	2.085
		90	1	1600	2.226	1715	2.115
			2	1600	2.416	1675	2.109

$\theta = \theta_o$ for DE for which the use of DE for a particular case, will become cost effective.

Similarly, if the cost of pressure drop increases due to the increase in the capacity, cost of power or increased resistance due to algae deposits etc., the optimum value of θ will be higher. It is also found that for a given case, θ_o with single stage of DE will be different than that with multiple stages. As the number of stages goes up, θ_o also increases.

Conclusions

1. Drift loss and pressure drop are greater with FD fan as compared to those with ID fan.
2. Drift loss and pressure drop are smaller for CDE than those for WDE.
3. Orientation of drift eliminators affects the performance of evaporative condenser. A higher value of θ improves the COP of the system.
4. As the number of DE stages increases, the drift loss decreases, but it leads to a higher Δp across the DE stages leading to an increase in the fan power input.
5. Optimum θ_o for DE for single as well as multiple stages can be found out on the basis of cost analysis. The optimum θ_o for the DE plates lies in the range of 45^o-60^o approximately for various stages of WDE and CDE with forced and induced flow. This angle is close to the value used in practice.

Abbreviations

CDE	Concrete drift eliminators
COP	Coefficient of performance
DBT	Dry bulb temperature, ^oC
DE	Drift eliminators
FD	Forced draft
ID	Induced draft
WBT	Wet bulb temperature, ^oC
WDE	Wooden drift eliminators

Nomenclature

$\dot{m}_a$	=	Mass flow rate of air (kg/min)
M_d	=	Specific drift loss (kgw/kgda)
$\dot{m}_d$	=	Rate of drift loss (kg/min)
M_e	=	Specific evaporation loss (kgw/kgda)
$\dot{m}_e$	=	Rate of evaporation loss (kg/min)
n	=	Number of stages
p	=	Static pressure at a point (mm of H_20)
P_{comp}	=	Power input to compressor (W)
P_{fan}	=	Power input to fan (W)
w	=	Specific humidity (kg water/ kg dry air)
w_1	=	Ambient air specific humidity (kg water/ kg dry air)
w_2	=	Specific humidity of the leaving air without duct heater (kg water/ kg dry air)
w_3	=	Specific humidity of the leaving air with duct heater (kg water/ kg dry air)
Δp	=	Total pressure drop (mm of H_20)
θ	=	Angle of inclination with horizontal (degree)
θ_o	=	Optimum inclination angle with horizontal (degree)

Subscripts

a	=	Point just below the DE
b	=	Point just above one stage of DE
c	=	Point just above two stages of DE

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