**DEMONSTRATION OF ADIABATIC COOLING
TO IMPROVE THE PERFORMANCE OF
AIR-COOLED HEAT EXCHANGERS**

by

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ABSTRACT

A demonstration of improved heat exchanger performance by adiabatically cooling the inlet air to an air-cooled heat exchanger was performed on a condenser that had a design heat rejection rate of 58 million Btu/hr. Pneumatic nozzles were used to inject atomized water into the inlet air stream. Mean droplet diameters ranged from 16 to 28 microns for all of the tests. Results of these experiments comparing conventional and adiabatic cooling showed that adiabatic cooling does improve heat transfer performance in accordance with expectations.

The operation of the system was simple and reliable. Visual observations of the heat exchanger showed that finned surfaces remained dry even when enough water was added to saturate the inlet air. With the use of efficient atomizers, adiabatic cooling improves efficiency at a steam-electric power plant without increasing dry cooling tower size, and thereby avoids the need to reduce plant output during hot weather.

Key Words: adiabatic cooling, atomizers, dry cooling towers,
 water injection

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NOMENCLATURE

A	=	Heat exchanger surface area (ft ²)
C _p	=	Specific heat (Btu/lb _m - °F)
h	=	Enthalpy (Btu/lb _m)
h _{fg}	=	Heat of vaporization (Btu/lb _m)
$\dot{M}$	=	Mass flow rate (lb _m /hr)
Q	=	Heat rejected (Btu/hr)
T	=	Temperature (°F)
U	=	Overall heat transfer coefficient (Btu/hr-ft ² -°F)

SUBSCRIPTS

a	=	Air
f	=	Fluid
LM	=	Log-mean
p	=	Pipe
sat	=	Saturation condition
w	=	Water
1	=	Inlet
2	=	Outlet

1. INTRODUCTION

In a recent theoretical and experimental study, the concept of injecting a fog of fine water droplets into the entering air stream of a dry cooling tower was examined (1)*. It was concluded that the power required to atomize 7 to 10 micron diameter droplets for fog injection would be too high to justify its use for most cases. Here, "fog injection" means injection of more than enough water to saturate the entering air, using droplets that are small enough to evaporate within the finned tube section, without wetting the fins. However, injecting 20 micron droplets could adiabatically cool the inlet air with complete evaporation of the droplets to prevent fouling of fins by dissolved solids. The power requirements for adiabatic cooling would be much less than for fog injection.

The main advantage of adiabatic cooling compared to conventional cooling is a reduction in the condenser temperature without a corresponding increase in dry cooling tower size. This results in avoiding the need to severely reduce plant output when there is a large demand for electricity during hot summer periods.

It is expected that adiabatic cooling would be used periodically during the warm summer months and not throughout the whole year. The purpose of the study reported here was to test a practical demonstration of adiabatic cooling on a large enough scale to permit evaluation of its performance and operation during warm weather.

* See reference at the end of Section 3.

2. EXPERIMENTAL PROCEDURE

A demonstration of adiabatically cooling the inlet air to an air-cooled heat exchanger was performed at the Sun Oil Company refinery in Marcus Hook, Pennsylvania, which is about 20 miles southwest of Philadelphia. The heat exchanger has a design heat rejection rate of 58.4 million Btu per hour and is used to condense an organic fluid to within a specified temperature range. A schematic diagram of the heat exchanger is shown in Figure 2-1. Four 8-ft wide by 24-ft long heat exchanger sections are connected in parallel to give a total frontal area of 768 ft². Fluid enters the center of a header which feeds directly into the heat exchanger. There are six rows of 1" O.D. carbon steel tubes with 0.5 in. aluminum fins spaced at 10 fins per inch. The two-pass heat exchanger has a triangular tube pitch of 2.06 in. and a total finned surface of 107,000 ft². The fluid leaves the heat exchangers through eight insulated pipes before combining in a single outlet pipe.

Four 9.0 ft diameter fans are located in a plan 4.0 ft below the heat exchanger. Two of the fans, A and C, have automatic variable-pitch blades to control the air flow in response to changes in fluid outlet temperature. Each fan is driven by a two-speed, (900 and 1800 rpm) 25-hp motor which yields a maximum total air flow of 420,000 SCFM.

For the experiments, one-half of the heat exchanger was used to test adiabatic cooling, while the other half used conventional air cooling only. Each half of the heat exchanger was cooled by two fans. Since identical heat exchanger sections were connected in parallel, this provided a method to compare both cooling methods simultaneously.

Figure 2-1. Schematic Diagram of Air-Cooled Condenser

As shown in Figure 2-2, shrouds were installed below fans A and B to prevent wind from carrying atomized water away from the heat exchanger. These sheet metal windscreens were 4 ft. long and had a diameter of 9 ft. 2 in. Ten Delavan Swirl-Air Nozzles Model P/N 31693 were installed (5 within each shroud) in a plane 4.0 ft below the fan blades. Each nozzle has a maximum capacity of 4.0 gpm. These nozzles use hydraulic and pneumatic energy to atomize water. Air and water enter a turbulent mixing chamber and then impact on a distributor plate before leaving the nozzle as a finely atomized spray. Data from Delavan on the Sauter mean droplet diameter as a function of water flow rate and air pressure across the nozzle are shown in Figure 2-3. For this demonstration, the mean droplet diameter was less than 20 microns. Data supplied by Delavan on the pneumatic horsepower required to atomize water as a function of air pressure and water flow rate is shown in Figure 2-4. The nozzles operated in a manner such that for a fixed air pressure drop, a decrease in the water flow rate resulted in an increase in air flow rate which also increased the pneumatic horsepower requirement. At the same time, there was a corresponding decrease in mean droplet diameter.

Figure 2-5 shows a close-up picture of the spray from one of the nozzles. In order to get an even distribution of atomized water into the inlet section of the heat exchanger, the 5 nozzles were evenly spaced around a 3-ft diameter circle and tilted at about a 45° angle. The resulting spray pattern from one of the nozzles is clearly seen in Figure 2-6.

A schematic diagram of the water and air flow piping is shown in Figure 2-7. City water was piped to the nozzles which were at an elevation of 50 feet above ground. A booster pump was used to raise the pressure from 80 psig to about 120 psig at ground level. A 60-ft, 1 in. diameter pipe connected the pump to a flow control valve and a Fisher & Porter flow meter. From there, the pipe branched off at a tee connection. Each branch was connected to a manifold under each fan. The manifold supplied water from a common plenum, shown in Fig. 2-8, to each set of five nozzles.

- 1. WINDSCREEN FOR FAN A
- 2. WINDSCREEN FOR FAN B
- 3. FAN C

- 4. OUTLET PIPE FROM HEAT EXCHANGER (1 OF 8)
- 5. MULTIPOINT RECORDER
- 6. FLOW METER

Figure 2-2. Experimental Set-up

Figure 2-3. Droplet Size as a Function of Water Flow Rate and Air Pressure

Figure 2-4. Pneumatic Horsepower Requirements for Delavan Nozzles as a Function of Air Pressure and Water Flow Rate

Figure 2-5. Spray from Delavan Nozzle

Figure 2-5. Spray Pattern into Heat Exchanger

Figure 2-7. Schematic Diagram of Water and Air Flow Piping Arrangement

1. INLET AIR HOSE
2. INLET WATER LINE
3. SUPPORT STRUCTURE

4. PLENUM (SEPARATE FOR AIR AND WATER)
5. AIR LINE TO NOZZLE
6. WATER LINE TO NOZZLE

Figure 2-8. Air and Water Distribution Manifold

A similar piping arrangement was used to supply air to the nozzles from a 600 SCFM Gardner-Denver diesel-powered compressor. A 60-ft, 1.5 in. I.D. rubber hose was connected to a filter/regulator and then to a tee connection. From the tee each air line branched off to a manifold with a common plenum which supplied air to each set of nozzles.

Air pressure and water flow rate controls were located near the fans. Air pressure could be regulated over a range of 0 to about 80 psig, and water flow could be varied from 0 to about 16 gpm. This is significantly more than 9.1 gpm which is the maximum amount of water that would be required to saturate the inlet air on a 96°F day with 40% relative humidity and the fans on high speed. In order to check whether all the nozzles had the same flow, pressure gages were installed on the water and air lines adjacent to each nozzle, as shown in Fig. 2-5.

During all the tests, the fan pitch was set at a maximum setting and the inlet air velocity was measured below the fans using a rotating vane anemometer. Measurements were made at from four to twelve different locations in order to determine the total air flow through the heat exchanger.

Temperature measurements were made using copper-constantan thermocouples, and the data were printed out on a Honeywell multipoint recorder. Inlet air wet and dry bulb temperatures were recorded as well as the temperature of the water that was supplied to the nozzles. The wall temperature of each of the eight pipes leaving the heat exchanger was monitored by thermocouples that were placed under the insulation against the pipe walls. A few of these pipes can be seen in Fig. 2-2. Four of the pipes were at the outlet of the conventional air-cooled section of the heat exchanger, and four others were on the adiabatically-cooled side. A comparison of these temperatures was used to evaluate the enhancement in heat transfer obtained from injecting atomized water.

Data on the combined fluid outlet temperature from both halves of the heat exchanger as well as the total fluid flow rate through the condenser were obtained from instrumentation that was already being used by Sun Oil Company to monitor plant parameters. The output was recorded on a strip chart and could also be read on a digital readout. The fluid temperature entering the heat exchanger was obtained from a thermocouple reading provided by Sun Oil Company.

3. EXPERIMENTAL RESULTS

Testing was done on ten days between August 19 and September 9, 1975. Ambient temperature during the tests ranged from 71°F to 90°F, and the relative humidity was between 25% and 61%. The total heat rejected in the condenser varied from 23 million Btu/hr to 34 million Btu/hr. The fluid inlet temperature ranged from 338°F to 351°F. The minimum and maximum fluid outlet temperatures were 210°F and 280°F respectively. Sun Oil personnel took a daily sample of the liquid out of the condenser in order to determine its density as well as other properties. For the ten days of testing the density varied from 54.4° API to 56.9° API (specific gravity of .761 to .751).

A velocity profile measured within the 9 ft. 2-in diameter shroud is shown in Figure 3-1 for both high and low speed operation. These profiles were used to determine the air volume flow rate through the heat exchanger which was 49,500 ACFM per fan on low speed, and 118,000 ACFM $\pm 8\%$ per fan on high speed. This is in agreement with the design specifications of 110,000 ACFM at a 95°F air temperature.

Barometric pressure was obtained from the National Weather Bureau Service for Philadelphia International Airport. During testing, the minimum and maximum pressure was 29.92 in. Hg and 30.26 in. Hg respectively, so the actual air volume flow rate resulting from a change in pressure would vary by only 1.1%. The temperature range of 71°F to 90°F would correspond to a difference in ACFM of 3.6%. These differences were small compared to the $\pm 8\%$ accuracy in actual measured readings for air velocity, so the effect of temperature and pressure changes on air volume flow rate were neglected. The average inlet velocity at the fan for all of the tests was 12.4 ft/sec and 29.7 ft/sec for low and high fan speed respectively. The ratio of heat exchanger frontal area to fan inlet area was 2.91/1.0, so the air velocity entering the heat exchanger was 4.3 ft/sec for low fan speed and 10.2 ft/sec for high speed.

Figure 3-1. Velocity Profile at Fan

A droplet leaving a nozzle travelled 4.0 ft in the shroud at 12.4 ft/sec or 29.7 ft/sec depending on the fan speed. There was a sudden expansion after the fan, before the heat exchanger, where the average droplet velocity was 4.3 ft/sec or 10.2 ft/sec for a distance of approximately 5.0 ft. Therefore, the average time for evaporation of a droplet before reaching the finned tubes was 1.5 sec with the fan on low speed or 0.6 sec for high speed. In a recent theoretical study (1)* it was found that a 20 micron droplet would evaporate completely in 5 sec when travelling in an air stream. For this experiment, it was physically impossible to locate the nozzles further upstream of the heat exchanger in order to allow more time for the droplet to evaporate.

Observations during testing with the fan at low speed showed that when enough water was added to saturate the inlet air, the finned surfaces remained almost completely dry, but unheated structural supports were wetted. This indicates that the fins, which were at temperatures greater than 200°F, were hot enough to just about completely evaporate the droplets before they reached the metal surface. When the inlet water flow rate was half of the saturation rate, both cool and hot surfaces remained dry. When about 10% excess water was added, the fins become damp in spots; when 50% excess water was added, the heat exchanger was obviously wet, and some water collected and dripped down off cool surfaces.

In order to relate the measured wall temperature of the pipes leaving the heat exchanger to the fluid temperature inside the pipes, a correlation was made between the average wall temperature and the mixed outlet fluid temperature. More than fifty different data points were analyzed without water injection to obtain the following proportionality constant for the ratio of the difference between the average pipe wall temperature, T_p , and the ambient dry bulb temperature, T_a , to the difference between the fluid temperature, T_f , and T_a :

$$\frac{T_p - T_a}{T_f - T_a} = 0.86 \pm 5\%$$

* Numbers in parentheses indicate references at the end of Section 4.

Assuming this linear relationship holds over the range of temperatures when adiabatic cooling was employed, a drop in fluid temperature, ΔT_f , when atomized water is added to the inlet air stream is proportional to the measured drop in the wall temperature, ΔT_p , and can be expressed as,

$$\Delta T_f = \Delta T_p / 0.86.$$

A summary of the data for the adiabatically cooled half of the heat exchanger is shown in Table 3-1. The table consists of ambient wet and dry bulb temperatures for the date the tests were run, the fluid temperature into and out of the condenser, the fluid flow rate, the total water flow rate to the 10 nozzles, the ratio of the injected water flow rate to the rate required to saturate the inlet air, the Sauter mean droplet diameter, and the decrease in the fluid temperature out of the condenser due to adiabatic cooling. The fans were on low speed for all of these runs, and the fan pitch was at its maximum setting. Once a change in test conditions was made, it took approximately 15 to 20 minutes for the temperatures to return to a steady-state condition. However, during this time period, the fluid flow through half the condenser could vary by as much as 310 lb per minute because a portion of the flow to the condenser varied in response to a control that tried to maintain a specified fluid level in a tank beneath the condenser. As a result of fluctuations in flow, the combined outlet temperature out of the condenser also varied. Therefore, in order to evaluate the effect of adiabatic cooling, the average of four outlet pipe wall temperatures on both the conventional and adiabatically cooled sections of the heat exchanger were compared to determine the incremental temperature drop of the fluid due to the injection of atomized water. For example, on September 5th, the average outlet pipe wall temperatures on both halves of the heat exchanger were the same without water injection. Twenty minutes after injecting 1.4 gpm into half of heat exchanger, the average wall temperature on the adiabatically cooled side dropped 4.5°F, while the conventionally cooled side temperature rose 0.5°F. Therefore, adiabatic cooling resulted in

Table 3-1. Summary of Data for Adiabatic Cooling Demonstration

DATE	AMBIENT TEMPERATURE		FLUID TEMPERATURE		FLUID FLOW RATE (lb per minute) (±10%)	ATOMIZED WATER		MEAN DROPLET DIAMETER (Micron)	OUTLET FLUID TEMPERATURE DROP (±1°F)
	Dry Bulb (±1°F)	Wet Bulb (±0.5°F)	Into Con- denser (±5°F)	Out of condenser (±2°F)		Flow Rate (GPM) (±3%)	$R/A_{W sat}$		
8/19/75	79	61	341	260	1535	0	0	-	-
	↓	↓				3.1	.87	19	11.1
						6.2	1.7	18	11.6
8/22/75	84	68	350	280	1600	0	0	-	-
	↓	↓				2.7	.86	17	8.4
						5.4	1.7	18	8.4
8/26/75	86	70	346	265	1115	0	0	-	-
	↓	↓				4.4	1.6	18	14.1
8/27/75	80	62	351	210	980	0	0	-	-
	↓	↓				3.6	1.0	17	14.7
						6.4	1.8	19	16.3
8/28/75	76	60	338	267	1520	0	0	-	-
	↓	↓				2.3	.75	18	6.2
	79	61				3.6	1.2	17	11.1
	↓	↓				4.6	1.5	18	11.1
9/2/75	76	60	352	239	1380	0	0	-	-
	↓	↓				2.7	.96	17	9.3
9/4/75	75	60	351	267	1580	0	0	-	-
	↓	↓				1.5	.5	16	5.5
						3.0	1.0	17	8.7
9/5/75	73	58	350	261	1500	0	0	-	-
	↓	↓				1.4	.5	16	5.8
	75					2.8	1.0	17	10.7
9/8/75	73	64	345	265	1615	0	0	-	-
	74	65				.9	.9	16	2.7
	76	↓				1.8	1.0	17	7.0
	75	64				2.7	1.5	17	6.7
9/9/75	71	54	348	269	1660	0	0	-	-
	↓	↓				.9	.3	16	2.0
						1.8	.6	17	3.8
						2.7	.8		7.0
						3.4	1.1		7.0

a 5.0°F effective temperature drop in the wall temperature which corresponds to a 5.8°F drop in fluid temperature. The inlet water flow was then doubled to 2.8 gpm, and twenty-five minutes later the adiabatically cooled side wall temperature dropped 12.2°F, but the dry side temperature dropped 3.0°F. This resulted in a 9.2°F wall temperature drop, or a 10.7°F decrease in fluid temperature due to adiabatic cooling.

A plot of temperature drop versus water injection rate is shown in Figure 3-2. $\dot{M}_w$ is the water flow rate, ΔT_f is the drop in fluid outlet temperature, and the subscript "sat" designates the condition at which the inlet air is saturated. The solid line shows the expected behavior, and "x's" indicate data points. It can be shown that the change in fluid outlet temperature, ΔT_{f2} , is a linear function of the inlet water flow rate, $\dot{M}_w$. An overall energy balance is shown in Eq. (1).

$$Q = UA \Delta T_{LM} = \dot{M}_a C_{pa} (T_{a2} - T_{a1}) = \dot{M}_f (h_{f1} - h_{f2}), \quad (1)$$

where Q is the total heat rejected, U is the overall heat transfer coefficient, A , is the heat transfer surface area, ΔT_{LM} is the log-mean temperature difference, $\dot{M}$ is the mass flow rate, C_p is the specific heat, h is enthalpy, subscript "a" denotes air, "f" is fluid, "1" is inlet conditions, and "2" is the outlet. When water is injected into the inlet air stream, the increase in heat rejection, ΔQ , is equal to $\dot{M}_f C_{pf} (T_{f2} - T'_{f2})$ where the superscript " ' " denotes adiabatic cooling conditions. Assuming that U , $\dot{M}_a$, C_{pa} , C_{pf} , and T_{f1} are constant, and taking the log-mean temperature difference to be equal to the arithmetic mean temperature difference,

$$\Delta T_{LM} = \frac{T_{f1} + T_{f2}}{2} - \frac{T_{a1} + T_{a2}}{2}, \quad (2)$$

(which is accurate to $\pm 3\%$ for these experiments) the energy balance equations for the conventional and adiabatic cooling conditions can be manipulated to yield the decrease in outlet fluid temperature, ΔT_{f2} , in terms of known and experimental

Figure 3-2. Temperature Drop versus Injected Water Flow Rate

measured quantities,

$$T_{f2} - T_{f2}' \equiv \Delta T_{f2} = \frac{2 h_{fgw} \dot{M}_w}{\dot{M}_a C_{pa} \left[1 + \frac{C_{pf} (T_{f1} + T_{f2} - 2T_{a1})}{(h_{f1} - h_{f2})} \right]}, \quad (3)$$

where h_{fgw} is the latent heat of vaporization of water, and the decrease in inlet air temperature, ΔT_{a1} , is given by,

$$\Delta T_{a1} = - \frac{\dot{M}_w h_{fgw}}{\dot{M}_a C_{pa}}. \quad (4)$$

From Eq. (3) it is easy to show that $\Delta T_{f2} / \Delta T_{f2 \text{ sat}} = \dot{M}_w / \dot{M}_{w \text{ sat}}$, which is represented by the 45° line in Figure 3-2. Once the air has been adiabatically cooled down to its wet bulb temperature, additional water injection will not reduce the air temperature any further. Any additional drop in outlet fluid temperature would then arise as a result of (a) further reduction of outlet air temperature due to water evaporation within the finned tube section, or (b) an increase in air-side film coefficient. Unless the additional water injection is sufficient to cause substantial fin wetting, neither of these effects should be appreciable because the droplets pass through the finned section too quickly for substantial evaporation to occur. Consequently, the expected behavior is that there is practically no additional drop in outlet fluid temperature with further addition of water after the inlet air has been saturated. As shown in Figure 3-2, the ratio of the outlet fluid temperature drop to the temperature drop with the inlet air saturated behaves in accordance with this expectation.

Figure 3-3 compares the measured outlet fluid temperature drop with the temperature decrease calculated by Eq. (3). The dashed line is a line of linear regression that best fits the data points. It shows that the measured temperature drop is about 85% of the predicted drop which is represented by the 45° solid line. Although these results are in good agreement with calculations and are within the accuracy of the experiments, it is possible that the small difference is due to one of the following reasons: (a) there was insufficient

Figure 3-3. Comparison of Measured and Predicted Outlet Fluid Temperature Drop

time to completely evaporate all of the water droplets, (b) some of the water collected on structural components and dripped down on the ground, or (c) the atomized water was not distributed uniformly over the entire heat exchanger frontal area.

For the nozzles used in this demonstration, the pneumatic power required to inject 16 to 19 micron diameter droplets at various water flow rates ranged from 33 hp to 45 hp. Assuming a compressor efficiency of 65%, the compressor horsepower was between 51 and 69 hp. The power that would be required to increase the overall heat rejection an equivalent amount by increasing the inlet air flow rate can be calculated from the appropriate energy balance equations. Equation (1) is an energy balance for the system under normal operations. An increased air flow rate, $\dot{M}_a^*$, is defined as that rate at which the outlet fluid temperature would drop from T_{f2} to T_{f2}' with a constant inlet air temperature, T_{a1} . In terms of known and measured parameters, the ratio of $\dot{M}_a/\dot{M}_a^*$ can be written as,

$$\frac{\dot{M}_a}{\dot{M}_a^*} = 1 - \frac{\dot{M}_a C_{pa} \left[1 + C_{pf} (T_{f2} + T_{f1} - 2T_{a1}) / (h_{f1} - h_{f2}) \right]}{\dot{M}_f \left[C_{pf} + (h_{f1} - h_{f2}) / (T_{f2} - T_{f2}') \right]} \quad (5)$$

Assuming that the fan power is proportional to $\dot{M}_a^3$, the increased power required, ΔP , becomes

$$\Delta P = P \left[\left(\dot{M}_a^* / \dot{M}_a \right)^3 - 1 \right] \quad (6)$$

At low speed, each fan requires approximately 3 hp, so the power to run two fans on one-half of the heat exchanger equals 6 hp. Calculating the air flow rate ratio from Eq. (5) and substituting into Eq. (6) yields the additional fan power that would be required to obtain the same increase in heat rejection that resulted from saturating the inlet air by injecting atomized water. This increase ranged from 3 to 8 hp for these tests. This is much less than the power required to produce 16 to 19 micron diameter droplets with the nozzles used in this demonstration. Therefore, for adiabatic cooling to be practical, the power necessary to atomize inlet water must be reduced significantly. The two possibilities for this are (a) to use different nozzles that are more efficient, or (b) to use larger size droplets

that require less power to produce. It should be pointed out that only commercially available nozzles were investigated for use in this demonstration. Other types, such as rotary pneumatic atomizers have been made for laboratory testing. However, they are not commercially available. Calculations based on published results of parametric values for rotary pneumatic atomizers have shown that the power consumption could be reduced to about 1/12 of the power required by conventional pneumatic nozzles producing 20 micron droplets (1).

On August 27th, tests were run to determine the effect of injecting larger diameter droplets on heat exchanger performance. A comparison was made for two cases, 17 and 28 micron droplets, keeping a constant water flow rate of 3.6 gpm. There was no difference that could be observed in either the wetting of fins or the measured outlet fluid temperature from the condenser. From Figs. 2-3 and 2-4, the power required to generate 30 micron diameter drops is between 1/3 to 1/4 of that which was necessary to produce 16 to 19 micron droplets for the tests in Table 1.

To summarize the power requirements for this demonstration of adiabatic cooling, the compressor power necessary to atomize the injected water with an 18 micron mean droplet diameter was between 51 and 69 hp. The water pump consumed approximately one hp additional. If the same Delavan nozzles were used to generate 30 micron diameter droplets, the power required would have been roughly 13 to 18 hp. If rotary pneumatic atomizers were available, the power requirement would drop to about 5 hp to generate 18 micron diameter droplets and would be even less to produce 30 micron droplets. This compares to an increase in fan power of 3 to 8 hp that would be required to achieve the same cooling effect by increasing the air flow through the heat exchanger.

Therefore, for this particular set of test conditions, the power required to produce droplets would make adiabatic cooling impractical. It should be pointed out that the scope of this program was limited in nature. The main purpose was to validate and demonstrate the performance of the adiabatic cooling concept. This has been done. Previous

work which analyzed the application of atomized water injection for dry cooling systems for power plant applications concluded that

"adiabatic cooling improves net plant efficiency,.... and avoids the need to reduce (plant) output in hot weather. If installation and maintenance costs of atomizer are low enough, adiabatic cooling with 20 micron droplets leads to a lower yearly cost than a high back pressure turbine with a smaller cooling tower" (1).

4.0 SUMMARY AND CONCLUSIONS

A demonstration of improved heat exchanger performance by adiabatically cooling the inlet air to an air-cooled heat exchanger was performed on a condenser that had a design heat rejection rate of 58 million Btu/hr. Pneumatic nozzles were used to inject atomized water into the inlet air stream. Mean droplet diameters ranged from 16 to 28 microns for all of the tests. Results of these experiments showed that adiabatic cooling does improve heat transfer performance in accordance with expectations. The expected improvement in heat transfer was calculated on the basis that heat transfer rate is proportional to the difference between the mean fluid temperature and the mean air temperature, and that adiabatic cooling occurs uniformly throughout the inlet air stream by complete evaporation of the injected water. The temperature data agreed with predictions to within the accuracy of the experiment. The small difference between the actual and predicted thermal performance can probably be explained by a combination of one or all of the following reasons: (1) there was insufficient time to completely evaporate all of the water droplets, (2) some of the water collected on structural components and dripped down on the ground, and (3) the atomized water was not distributed uniformly over the entire heat exchanger frontal area.

The operation of the system was simple and reliable. Although manual controls were utilized to control the water and air flow, automatic controls could be incorporated without much difficulty to regulate the outlet temperature in response to changes in either ambient or plant operation conditions.

The scope of this program was limited to demonstrate and verify the performance of the adiabatic cooling concept which was done. However, for the specific process and operating conditions that were tested, it

appears that adiabatic cooling would not be practical because of the power required to atomize the water. For these tests, the average inlet fluid temperature was about 350°F, whereas in a dry cooling system rejecting condenser heat from a power plant, the water temperature is approximately 130°F. It is expected that adiabatic cooling would be more advantageous for power plant applications than for this demonstration because the effect of a decrease in the inlet air temperature is more significant for systems operating at lower temperature. For example, if air were at a dry bulb temperature of 95°F and 40% relative humidity, adiabatic cooling would result in an inlet air temperature of about 75°F. The initial temperature difference (ITD) for this demonstration would be either 255°F for conventional cooling or 275°F for adiabatic cooling. This is a relatively small percentage increase in ITD. On the other hand, the ITD for a power plant dry cooling system would increase from 35°F for conventional cooling to 55°F for adiabatic cooling.

Based on the test results and the data developed during previous work (1), it can be concluded that:

1. Development of a rotary pneumatic atomizer having a power requirement approximately one-third of that of the test nozzle is necessary if adiabatic cooling is to be competitive on a power requirement basis with the approach of obtaining the same cooling effect by increasing fan power. Therefore, any future effort should be placed primarily on such nozzle development prior to any large-scale tower tests.
2. Depending on the results of the aforementioned nozzle development effort, an economic evaluation should then be conducted of adiabatic cooling compared to other alternatives. A preliminary economic analysis of adiabatic cooling (1) indicated that "adiabatic cooling with small droplets is economically advantageous." It is known, however, that adiabatic cooling from the theoretical standpoint yields higher water consumption than that of a combination dry and wet peaking cooling system. Therefore, any comparative economic evaluation should consider capital, operating, water, and all associated system costs so as to establish the potential of adiabatic cooling with respect to other systems.

REFERENCES

1. G. P. Wachtell, "Atomized Water Injection to Improve Dry Cooling Tower Performance", Report No. C00-2241-1, November 1974, (\$5.45), available from NTIS, Springfield, VA 22161.

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