# Adiabatic pre-cooling of the inlet air to an air-cooled heat exchanger

H. Botes[1] and D.G. Kröger[2]

## Nomenclature

| | | |
|---|---|---|
| $A$ | area | [m$^2$] |
| $AMD$ | arithmetic mean diameter | [$\mu$m] |
| $c_p$ | specific heat | [J/kg°C] |
| $d$ | diameter | [m] |
| $H$ | height | [m] |
| $K$ | flow loss coefficient | [ – ] |
| $m$ | mass flow rate | [kg/s] |
| $N$ | revolutions | [rpm] |
| $NTU$ | number of heat transfer units | [–] |
| $Ny$ | characteristic heat transfer parameter | [l/m] |
| $n$ | number | [ – ] |
| $p$ | pressure | [N/m$^2$] |
| $Q$ | heat transfer rate | [W] |
| $RY$ | characteristic flow parameter | [l/m] |
| $SMD$ | Sauter mean diameter | [$\mu$m] |
| $T$ | temperature | [K] |
| $VMD$ | volumetric mean diameter | [$\mu$m] |
| $v$ | velocity | [m/s] |
| $w$ | humidity ratio | [kg/kg] |
| $\Delta p$ | pressure differential | [N/m$^2$] |
| $\Delta T$ | temperature differential | [K] |
| $\alpha$ | kinetic energy coefficient | [–] |
| $\varepsilon$ | effectiveness | [–] |
| $\mu$ | dynamic viscosity | [N/ms] |
| $\rho$ | density | [kg/m$^3$] |

## Subscripts

| | |
|---|---|
| a | air |
| db | dry bulb |
| de | drift denominator |
| do | downstream of the fan |
| F | fan |
| Fc | fan casing |
| Fe | fan effective (fan-casing fan hub) |
| Fsys | fan system (fan and plenum) |
| Fs | fan static |
| fr | frontal |
| he | heat exchanger |
| heiso | heat exchanger under isothermal conditions |
| p | process fluid |
| pl | plenum |
| sp | sprayer/atomiser |
| ts | tower supports |
| up | upstream of the fan |
| w | water |
| wb | wet bulb |
| 1 | before cooling |
| 2 | after pre-cooling |

## Abstract

*The use of adiabatic pre-cooling to regulate the temperature of the inlet air to an air-cooled heat exchanger during periods of high ambient temperatures is investigated theoretically and experimentally. Pre-cooling is achieved by spraying a rotary atomiser generated water mist into the air stream. The design of such an atomiser is briefly discussed. The use of drift eliminators to ensure a dry finned tube surface is also investigated. Pressure drop characteristics and droplet collection efficiencies of common drift eliminator shapes are determined experimentally. A conceptual air-cooled heat exchanger with pre-cooling is also modelled numerically.*

## Introduction

Air-cooled heat exchangers (ACHEs) are commonly found in petrochemical and process plants. In recent years, environmental considerations have further encouraged their increased use. However, the main drawback of an air-cooled heat exchanger is its dependence on the air dry bulb temperature. Because of the daily and yearly variation in ambient dry bulb temperature, the efficiency of an ACHE changes correspondingly. By pre-cooling the inlet air, especially during periods of high ambient temperatures, effective plant operation can be maintained.

Adiabatic pre-cooling of air is the process whereby the non-saturated inlet air to the heat exchanger is cooled by the injection of a fine water spray into the air. The air can then theoretically be cooled down to the air wet bulb temperature following the wet bulb temperature line on a psychrometric chart as shown in Figure 1. Various researchers have conducted experiments on adiabatic pre-cooling:

Wachtell [1] stated that the inlet air to an air-cooled heat exchanger can be cooled to nearly the wet bulb temperature by the injection of 20 micron or smaller droplets. He found that no visible fin wetting took place with 20 micron droplets, even if too much water was injected. In

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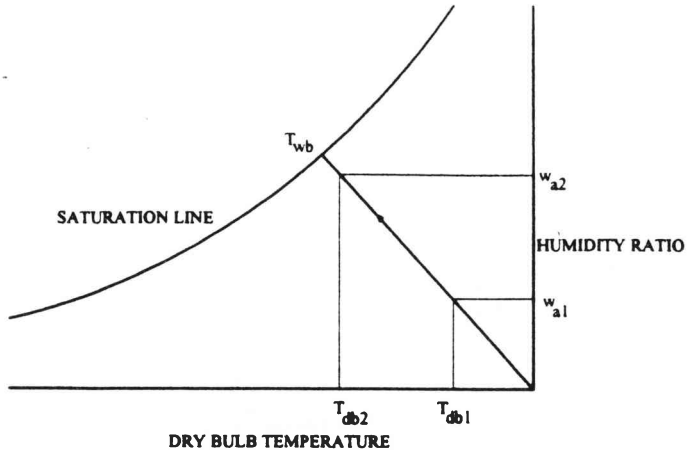


Figure 1 Psychrometric chart depicting the adiabatic pre-cooling process

his tests Wachtell used pneumatic nozzles and concluded that for adiabatic cooling to be practical, the power necessary to atomise the inlet water must be reduced significantly, possibly by employing more effective nozzles or rotary atomisers. Rubin [2] conducted tests to determine the improvement in capacity of an air-cooled heat exchanger with adiabatic pre-cooling and he too found that with droplets smaller than 20 micron there was no visible fin wetting.

If the mass flow of injected water, $m_w$, is specified, the final humidity ratio can be calculated from:

$$w_{a2} = w_{al} + \frac{m_w}{m_a} \tag{1}$$

It is then possible to calculate the final dry bulb temperature (in K) from [3]:

$$T_{db2} = \frac{1}{1.8577 w_{a2} + 1.00416} \times \left[ \begin{array}{l} w_{a2} \left\{ \begin{array}{l} 4.184\,(T_{wbl} - 273.15) \\ -1994.169 \end{array} \right\} \\ \\ + \{2501.6 - 2.3263\,(T_{wbl} - 273.15)\} \\ \\ * \left\{ \frac{0.62509 p_{vwb}}{p_{a2} - 1.005 p_{vwb}} \right\} + 1.00416 T_{wbl} \end{array} \right] \tag{2}$$

However, it must be noted that the mass flow of water, $m_w$, is the mass of water that must evaporate. The injected mass flow of water may be higher than this value, since not all the water injected necessarily evaporates and the water flow rate depends on the droplet size of the spray and the humidity of the air.

A conceptual ACHE incorporating adiabatic pre-cooling by means of rotary atomisers and a drift eliminator is shown schematically in Figure 2. To predict the performance characteristics of such a system, the properties of the various components that make up the system must be known.

## Atomiser experiments

The first part of this investigation was devoted to the testing of commercially available spray nozzles. The experiments conducted by Wachtell [1] and Rubin [2] were conducted on relatively low structures (4 m). In most practical applications the inlet height of the ACHE is approximately 10 −12 m and there is subsequently more time to evaporate the spray. Wachtell proposed a model to determine the time of evaporation of a droplet in an air stream and by using this model it was found that a 50 micron droplet had sufficient time to evaporate in 12 m. This was calculated for air having a wet bulb temperature of 18°C and that was cooled down from a maximum of 36°C to a final temperature of 28°C. It was also assumed that the heat exchanger normal approach velocity was approximately 3 m/s.

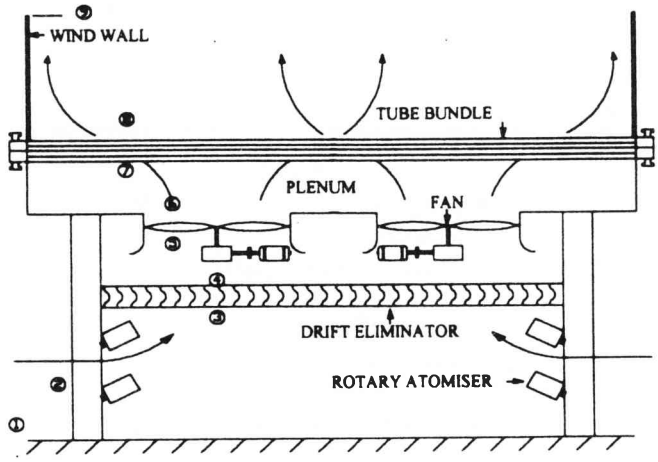


Figure 2 Schematic of ACHE used for simulation of adiabatic pre-cooling

The criteria which were taken into consideration during the present tests were:

(a) droplet size of 50 micron or smaller,
(b) relatively high water flow rate,
(c) low maintenance (wear of nozzle parts and clogging of nozzle),
(d) uniform droplet size, and
(e) low capital and operating cost.

Three types of atomisers were tested, namely

(a) pneumatic nozzles,
(b) high pressure nozzles, and
(c) industrial rotary atomisers.

A test rig was constructed onto which the atomiser could be mounted with water and high pressure air supplied as needed. The compressed air mass flow rate was measured with a BS 1042 standard orifice,[4] while the water flow rate was measured mechanically. Droplet sizes

were measured with a Malvern 2600 laser diffraction instrument. In the case of the rotary atomisers, water sensitive paper was used to sample the spray and a microscope and camera arrangement was used to measure the droplet sizes.

From these tests it was evident that none of the pneumatic or high pressure nozzles were capable of complying with the above-mentioned criteria. They could not achieve the high water flow rates required and their capital and operating costs for large scale applications made them economically unsuitable. The rotary atomisers tested displayed a much more uniform droplet distribution than the spray nozzles and their operating costs were approximately $\frac{1}{16}$ of the conventional nozzles. Although their water flow rates were also substantially higher, they were still too low to make them practical in an ACHE set-up. It was subsequently decided to design a rotary atomiser from basic atomiser principles to comply with all the criteria listed above.

According to Masters,[5] the mechanism of atomisation has been studied by many workers, but remains a controversial subject despite much published data. It is however generally accepted that the degree of atomisation depends on

(a) the relative velocity between the liquid and air at the atomiser periphery,

(b) liquid feed rate,

(c) liquid properties (surface tension and viscosity), and

(d) the atomiser design.

Maximum centrifugal energy is imparted to the liquid when the liquid acquires the wheel or disc peripheral speed before discharge. If a flat smooth disc is rotating at high speed and liquid is fed onto its top surface, severe slippage occurs between the feed liquid and disc. The velocity of liquid from the disc edge is much lower than the disc peripheral speed. To prevent slippage in commercial atomisers, radial vanes are used. The liquid is confined to the vane surface and at the periphery the maximum liquid release velocity possible is obtained. This maximum velocity is the resultant of the radial and tangential components acquired by the liquid. Slippage can also be reduced by increasing the friction between the liquid and rotating surface. This is often done by feeding the liquid onto the lower surface of a disc shaped as an inverted cup. In the design of the atomiser under discussion this was achieved by forming the liquid film on a very fine wire mesh. The extent of slippage also depends upon feed rate and physical properties of the liquid, namely the viscous drag and the ability of the liquid to wet the surface. A further prerequisite of the atomiser is a uniform droplet size distribution. According to Levebvre,[6] the homogeneity of the spray can be ensured by complying with the following guidelines:

(a) a large centrifugal force compared to the gravitational force,

(b) atomiser rotation should be vibrationless,

(c) the liquid feed rate should be constant, and

(d) the disc surfaces should be smooth.

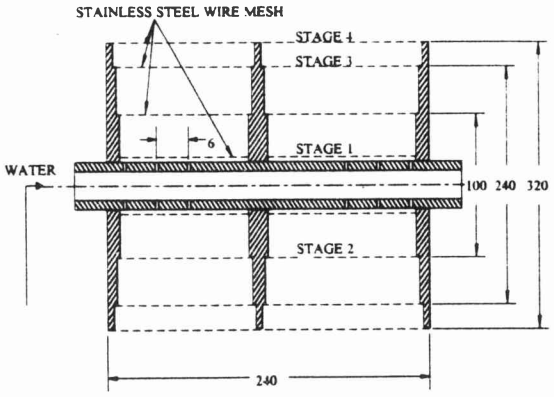


Figure 3 Schematic of designed rotary atomiser used to produce the required water spray

A schematic of the final atomiser design is shown in Figure 3. It essentially consists of three discs of 320 mm diameter mounted on a rotating shaft that is powered by an electrical motor through a belt drive. The shaft is supported in a frame between two bearings. Concentric wire mesh stages are mounted at different diameters on these discs. Water is fed into one end of the rotating shaft and subsequently flung outwards through openings in the shaft due to the action of centrifugal force. The testing of the atomiser consisted of changing the combinations of wire mesh stages and increasing the water flow rate up to approximately 800 l/hr while measuring the droplet sizes. The three-phase electric motor had a speed rating of 2 800 rpm and the motor power consumption was measured by the three watt meter method. Between the motor and the atomiser a 1.5 pulley ratio ensured that the atomiser rotated at approximately 4 200 rpm. This was measured for every flow rate with a hand-held tachometer. The outlet of a centrifugal fan was mounted in line with the atomiser shaft to deliver an air cross flow velocity of approximately $4 - 5$ m/s at the atomiser. Droplet collection was done at a distance of 3 m from the atomiser face by means of the water sensitive paper method.[7]

To determine the flow resistance of the atomiser, it was mounted in the 4 m × 4 m inlet section of a subsonic wind tunnel. By measuring the static pressure difference between the test section and inlet of the wind tunnel it was possible to evaluate the pressure drop characteristics of the atomiser at various air velocities.

## Results of atomiser experiments

The atomiser is capable of delivering 800 $\ell$ per hour at approximately 4 200 rpm (peripheral speed of 70 m/s). The peripheral speed was kept as low as possible to minimise

noise. The power requirement at the above operating conditions is approximately 4 kW and the arithmetic mean droplet size ($AMD$) is in the range of 50 – 75 micron. In Figure 4 the droplet size is given as a function of the water flow rate. Evidently the atomisation quality is relatively independent of the water mass flow rate. This means that an arithmetic mean diameter of between 50 and 75 micron will be obtained for all water flow rates up to 800 l/hr. The Sauter ($SMD$) and mass median ($MMD$) droplet sizes were also calculated. According to Lefebvre,[6] the closer the ratio of $MMD/SMD$ is equal to unity, the more uniform the spray. For the rotary atomiser the average value is 1.05 which means that the generated spray is relatively uniform. At no water load the atomiser consumes approximately 3.1 kW. This is due to the discs of the atomiser working as a centrifugal air pump and accelerating the air radially outward. The increase in power consumption with an increase in water flow rate results from an increase in kinetic energy imparted to the water. A linear relationship between the atomiser peripheral speed and water flow rate also exists.

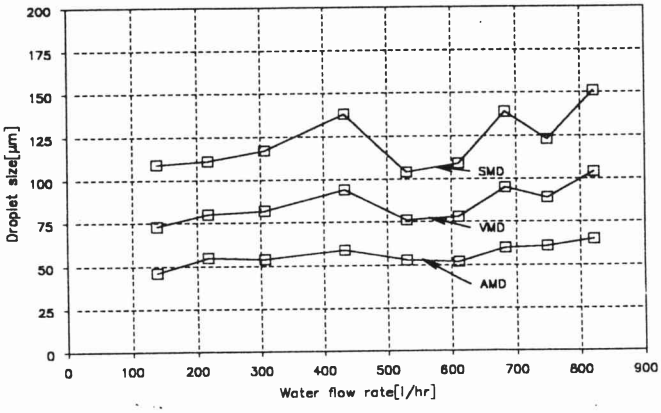


Figure 4 Droplet sizes versus water flow rate characteristics for the atomiser

The results of the pressure drop experiments conducted on the atomiser in the 4 m × 4 m wind tunnel are displayed in Figure 5. The number, $Ry$, is termed the characteristic flow parameter and relates the air mass flow to the flow area under consideration. In this case the relevant area is the inlet area of the wind tunnel, $A_{in}$. It can be seen that the pressure drop across the non-operating atomiser is smaller than when it is rotating. This is due to the added flow resistance of the radially sprayed water.

### Drift eliminator experiments

From the investigation into atomisation it was obvious that to generate a very fine and uniform spray would be difficult and expensive. Wetting of the finned surfaces in an ACHE is however not desirable. This is so because the surrounding industrial atmosphere may contain sulphur or chlorine or other corrosive substances that when combined with moisture, tend to encourage corrosion of the finned surfaces. Other airborne particles such as dust, fly ash and coal dust cause fouling of the tube bundles when wet. Fouling tends to increase the thermal and flow resistance on the air side, resulting in a net reduction in the heat transfer rate. Wetting of the finned surfaces can be avoided by installing droplet or drift eliminators upstream thereof.

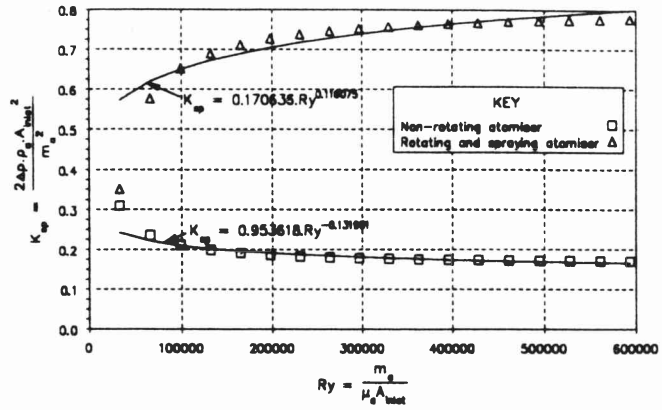


Figure 5 K-Ry curves for the atomiser in the operating and non-operating mode

The performance characteristics of three types of industrial drift eliminators (Figure 6) were determined experimentally. To conduct tests under wet and vertical air flow conditions, it was necessary to construct a vertical wind tunnel or tower to which air was supplied by a centrifugal fan and with high pressure spray nozzles installed to deliver the water to the system. A schematic of the experimental tower is shown in Figure 7. It has a total height of approximately 7 m and a cross section of 600 × 600 mm. The air flow was measured by means of a calibrated orifice plate in the supply line to the test tower and was regulated by a butterfly valve, while the water flow rate was measured by the displacement method. To ensure a uniform approach velocity to the eliminator packs, a flow straightener is installed at the diffuser inlet and guide vanes change the flow direction from the horizontal to the vertical.

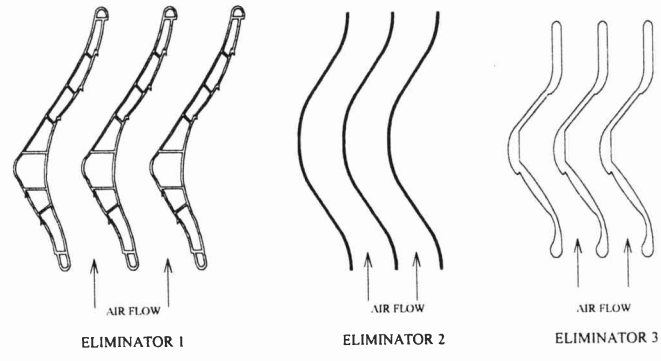


Figure 6 Schematic of the drift eliminators tested in the vertical experimental tower

Two different models of high pressure nozzles were installed. Nozzle A has a low water mass flow rate and an average droplet size of 60 $\mu$m, while Nozzle B has a much higher flow rate and an average droplet size of 95 $\mu$m. During every test there were four nozzles spraying concurrently. The drift eliminator packs were mounted on trays that were accessed through a removable perspex side of the tower.

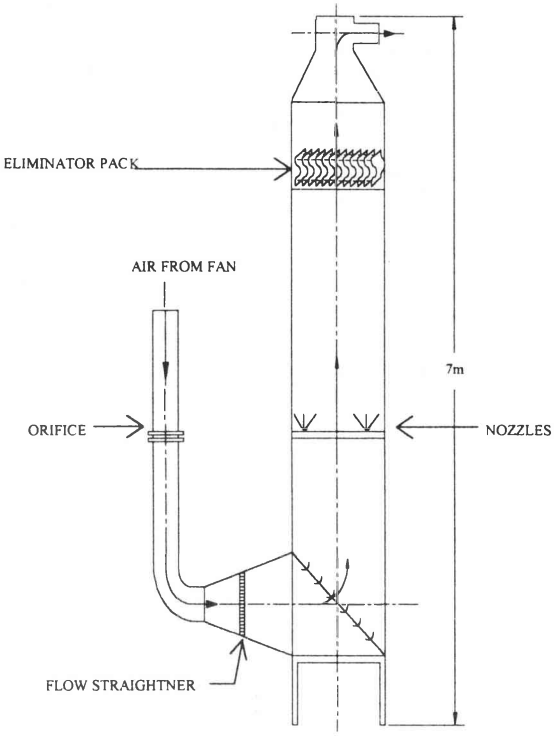


Figure 7 Schematic of vertical experimental tower used for drift eliminator tests

Droplet sizes were measured by collecting the spray on water sensitive paper. To expose the paper to the air stream containing the droplets, the paper was placed in a cylindrical tube with a sliding window that was opened gradually. A uniform spray collection distribution was thus obtained. At least three measurements were taken per mass flow rate per eliminator pack and an average value was calculated.

To quantify the accuracy of pressure drop measurements in the experimental tower, drift eliminator No. 1 was tested under dry conditions and the result was compared to a subsequent set of pressure drop measurements conducted in a code wind tunnel. The air velocities were varied from zero to approximately 3.5 m/s. A mass balance of the water flow in the experimental tower was also performed to establish the percentage of water trapped by the drift eliminators.

### Results of drift eliminator experiments

In Figure 8 the results of the eliminator pressure drop experiments are illustrated. It is evident that drift eliminator No. 2 has the lowest pressure drop characteristics of the three eliminators tested. The difference in the pressure drop curves for the dry and wet experiments is also small for the liquid loadings tested(0–4 kg/m$^2$s). The effectiveness of an eliminator pack was determined in the following way:

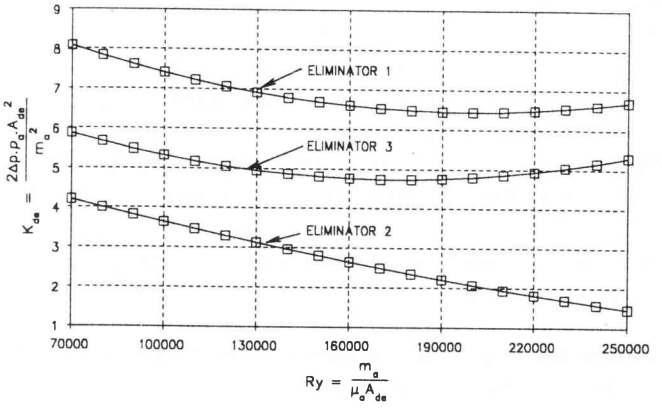


Figure 8 K-Ry curves for the three eliminator types tested under wet operating conditions

The water flow rates of the nozzles were firstly determined by collecting the water at the bottom of the tower without the fan in operation ($m_{oa}$). The drift eliminator was then installed and tests conducted at varying air speeds, with the water flow rate again measured at the bottom of the tower ($m_{ob}$). The amount of water that evaporated in the tower was calculated ($m_{evap}$). The effectiveness [%] of the drift eliminator can then be expressed as:

$$\eta_{de} = [1 - \{m_{oa} - (m_{ob} + m_{evap})\} / m_{oa}] \times 100 \qquad (3)$$

At low air velocities the droplets tend to flow through the eliminator passages, because they possess a sufficiently low momentum to follow the eliminator curvature. At higher air velocities the droplets are unable to undergo the sudden direction change in the flow channels and impinge on the side of the channel. Drift eliminator 1, and to a lesser extent drift eliminator 3, are equipped with serrated edges that entrain the droplets at the lower air velocities and they are therefore more effective in this region when compared to eliminator 2. At the high air velocities, the change in momentum of the droplets is the predominating entrainment mechanism and at approximately 3 m/s frontal velocity, the difference in the effectiveness of the three eliminators is negligible. It can thus be concluded that drift eliminator 2 is the most suitable eliminator to be installed underneath a heat exchanger bundle for the range of $Ry$ values tested. This conclusion is based on the low pressure drop values measured, a droplet collection efficiency of between 90% and 93% and from the fact that 80% of the droplets exiting the eliminator are smaller than 25 micron. A further positive aspect of this eliminator is its simplicity and therefore ease of manufacturing.

### Computer simulation of ACHEs

A computer program was written to evaluate the performance characteristics of a forced draught ACHE. The program is called ACHES, which is an acronym for Air-Cooled Heat Exchanger Simulation. ACHES is written in the FORTRAN computer language and makes provision for the adiabatic pre-cooling of the inlet air by means of atomisers. The option of including drift eliminators is also built into the program. In the program the energy and draught equations of a horizontal ACHE are solved simultaneously to determine its operating point. This is explained in more detail in Appendix B. An existing petrochemical plant fan bay was simulated to determine the effects of adiabatic pre-cooling thereon. The fan bay consists of two fans of 4.265 m diameter and two bundles, each with a frontal area of 36 $m^2$, supported at an elevation of 12.7 m as shown in Figure 2. The tube bundles have 6 rows of finned tubes, with 2 passes of three rows each and with mixing of the process fluid between the passes. The ACHE is designed for an ambient temperature of 28°C dry bulb and 18°C wet bulb to ensure a process fluid outlet temperature of 55°C. A complete list of the specifications of the ACHE is given in Appendix A.

As discussed previously, the installation of rotary atomisers and a drift eliminator in an existing ACHE set-up is proposed as a means to achieve adiabatic pre-cooling. Both the atomisers and the drift eliminator constitute an added flow resistance to the inlet air flow. It is thus evident that for a given fan blade angle setting and fan revolution, the inclusion of these resistances will result in a lower air volume flow rate through the bundles and consequently a higher process fluid outlet temperature will result. It must also be noted that at the design inlet air temperature of 28°C, it will be necessary to perform adiabatic pre-cooling to ensure the process fluid outlet temperature of 55°C. This is clearly illustrated in Figure 9. From Figure 9 it is also evident that a near linear relationship exists between the ambient temperature and the process fluid outlet temperature, with a differential of approximately 0.8°C. Since the eliminators and atomisers are installed permanently, they also constitute an added flow resistance at ambient temperatures lower than 28°C. With the use of ACHES this added power consumption was found to be approximately 10 300 kWh/a for the particular plant. This equates to 92% of the running cost of the proposed pre-cooling system. If no eliminators are required, this cost essentially disappears, since the atomisers' flow resistances are smaller than that of the droplet eliminator. The water requirements for pre-cooling to ensure a process fluid outlet temperature of 55°C was also found to be a near linear function of the air dry bulb temperature. This is illustrated in Figure 10.

An alternative method of increasing the heat transfer in ACHEs, is to increase the fan revolutions. In existing plants with single-speed fan drives, this would require the installation of variable speed drives. Increasing the fan speed does however also result in an increase in fan power consumption as indicated in Figure 11 and in the noise generated by the fan blades. This approach does however have the advantage of having a lower power consumption at low ambient temperatures.

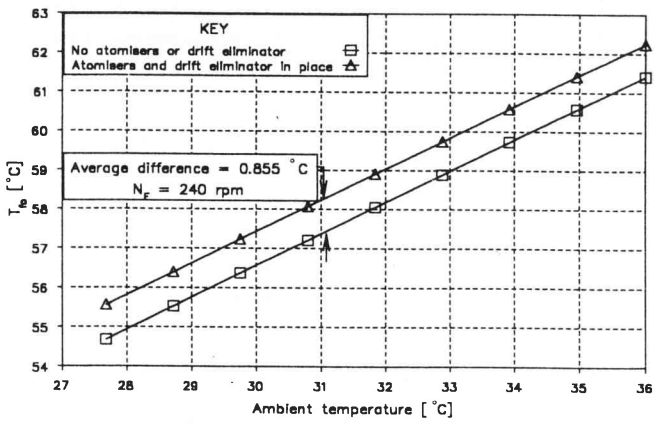


Figure 9 Increase in process fluid outlet temperature due to a lower air mass flow rate caused by the atomisers and drift eliminator flow resistances

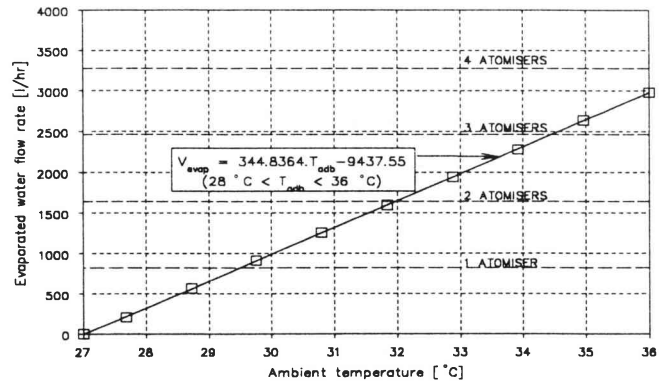


Figure 10 Water requirements for adiabatic pre-cooling to maintain a process fluid outlet temperature of 55°C, as a function of the ambient dry bulb temperature. The wet bulb temperature is 18°C.

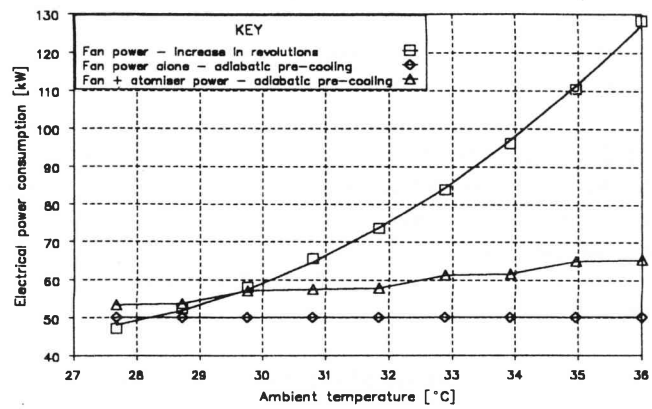


Figure 11 Comparison of power requirements for pre-cooling and increasing the fan revolutions

## Results and conclusions

The practical and economical application of adiabatic pre-cooling of the inlet air to an air-cooled heat exchanger was investigated. A rotary atomiser, capable of producing droplets with an arithmetic mean diameter ($AMD$) of between 50 and 75 micron and at a water flow rate of 800 l/hr, was designed and built. Three drift eliminator packs were tested to determine their droplet collection efficiency and pressure drop characteristics. It was found that the plain sinusoidal drift eliminator possesses the lowest pressure drop characteristic, while it has the same droplet collection efficiency (90–93%) as the other eliminator packs at approach velocities above 3 m/s.

A computer simulation program (ACHES) was written to evaluate the effect of adiabatic pre-cooling on the operation of an air-cooled heat exchanger. It was found that pre-cooling can be used successfully to control the outlet temperature of an ACHE during periods of high ambient temperatures. Adiabatic pre-cooling was also compared to the alternative of increasing the air flow through the bundles by increasing the fan revolutions. The latter method has the advantage of a lower annual running cost than pre-cooling, but its capital costs are higher. If a drift eliminator is required in an adiabatically cooled system, the cost of pre-cooling is approximately 72 % of that of the alternative and if no eliminator is necessary, this reduces to 51 %. Increasing the fan speed also has the disadvantage that the fan peripheral speed increases with a corresponding increase in fan noise.

It is thus concluded that adiabatic pre-cooling is a potential cost effective method of regulating the process fluid outlet temperature in times of high ambient temperatures in installations where the fan has a single-speed drive. Pre-cooling can also be considered in plants where the process fluid mass flow rate is increased above the design value and the outlet temperature is required to remain the same. In this way productivity can be increased.

## Acknowledgement

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## Appendix A

### Geometric and other design parameters for the forced draught ACHE used as example

The relevant design specifications of the ACHE used as an example to illustrate adiabatic pre- cooling are as follows:

The tube bundles are arranged horizontally with two tube bundles per fan bay of two fans. Adiabatic pre-cooling is achieved by means of rotary atomisers located in the inlet section of the heat exchanger as is shown schematically in Figure 2. A drift eliminator is installed directly beneath the tube bundles to ensure no fin wetting. The user also enters a fan curve in table format for each fan blade setting desired. A regression analysis is then performed on this data and subsequently used in conjunction with the fan laws to determine the air volume flow rate through the fan.

**Fan installation specifications**

Total number of fans per fan bay = 2.0

Height of fans above ground level, $H_5$ = 12.70 m

Fan diameter, $d_F$ = 4.265 m

Fan blade angle = 10.0°

Efficiency of fan drive system = 80.0 %

Loss coeff. for flow obstacles (fan suction), $K_{up}$ = 0.10

Loss coeff. for flow obstacles (fan discharge), $K_{do}$ = 0.15

ACHE support loss coefficient, $K_{ts}$ = 1.50

Height of wind wall, $H_w$ = 1.70 m

**Heat exchanger bundle specifications**

Number of HE bundles in a fan bay, $n_{heb}$ = 2.0

Frontal area of one bundle, $A_{fr}$ = 35.96 $m^2$

Effective finned tube length, $L_t$ = 10.203 m

Minimum HE free flow area/HE frontal area, $\sigma$ = 0.511

Contraction loss coefficient, $K_C$ = 0.05

Longitudinal tube pitch, $P_l$ = 63.5 mm

Tangential tube pitch, $P_t$ = 60.0 mm

Number of tube rows, $n_{tr}$ = 6.0

Maximum number of tubes per row, $n_{trmax}$ = 55.0

Mean number of tubes per row, $n_{trm}$ = 54.5

Number of tube passes in the HE bundle, $n_p$ = 2.0

Tube inside hydraulic diameter, $d_i$ = 0.02110

**Atmospheric and process fluid design specifications:**

Barometric pressure at ground level, Pa = 85000.00 $N/m^2$

Ambient temperature gradient, $dT_a/dz = -0.00975$ K/m

Process fluid-inlet temperature, $T_{pi}$ = 152.00°C

Process fluid specific heat, $c_{pp}$ = 3960.0 J/kg°C

Process fluid dynamic viscosity, $\mu_p$ = 0.00172 kg/ms

Process fluid heat transfer coefficient, $h_p$ = 1208 W/m2°C

Total process fluid mass flow rate per bay, $m_p$ = 30.550 kg/s

The annual ambient temperature distribution during which adiabatic pre-cooling would be required for the particular location is given in Table A.1.

Table A.1 Annual ambient temperature distribution during which adiabatic pre-cooling is required

| $T_{adb}$ [°C] | $T_{awb}$[°C] | Time [hr/a] |
|---|---|---|
| 27 | 15.9 | 76 |
| 28 | 16.1 | 51 |
| 29 | 16.4 | 30 |
| 30 | 16.8 | 15 |
| 31 | 17.4 | 8 |
| 32 | 18.0 | 4 |

## Appendix B

**Energy and draught equation for the ACHE**

The energy equation applicable to an ACHE describes the heat transfer between the process fluid and the air flowing across the tubes, while the draught equation takes into account all the flow resistances that the fans have to overcome. These two equations have to be satisfied simultaneously for a given set of design parameters.

The NTU effectiveness method is used to evaluate the heat transfer. The effectiveness is defined as:

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{(m.c_p)_{\min}\,\Delta T_{\text{minimum fluid}}}{(m.c_p)_{\min}\,\Delta T_{\max}} = \frac{\Delta T_{\text{minimum fluid}}}{\Delta T_{\max}} \tag{4}$$

where $\Delta T_{\text{minimum fluid}}$ = the temperature difference of the fluid having the smaller value of mass times specific heat; and $\Delta T_{\max}$ = the maximum temperature difference in the heat exchanger.

The heat transfer rate can now be calculated from:

$$Q = (m.c_p)_{\min}\,\varepsilon.\Delta T_{\max} = C_{\min}\varepsilon\Delta T_{\max} \tag{5}$$

The effectiveness of numerous practical arrangements can be found in various sources in the literature.[8; 9; 10] An iterative procedure is necessary to calculate the heat transfer for a specific ambient temperature. The iteration process is started by assuming a value for the effectiveness, $\varepsilon$. For this particular ACHE the process fluid is the minimum fluid. This is often the case where a process fluid is cooled in the petrochemical industry. There are however instances where it is not the case and therefore the program is able to apply the correct equation. Equation (4) can then be rewritten as:

$$\varepsilon = \frac{T_{\text{pi}} - T_{\text{po}}}{T_{\text{pi}} - T_{\text{ai}}} \text{ or } T_{\text{po}} = T_{\text{pi}} - \varepsilon\,(T_{\text{pi}} - T_{\text{ai}}) \tag{6}$$

The heat transfer from the process fluid can also be calculated from equation (5) and since the heat rejected by the process fluid is equal to the heat gained by the air, the air outlet temperature can be calculated from:

$$T_{\text{ao}} = T_{\text{ai}} + Q/\,(m_a c_{pa}) \tag{7}$$

In equations (6) and (7) the air inlet temperature, $T_{\text{ai}}$, is the temperature after adiabatic pre-cooling has taken

place and its value lies anywhere between the wet and dry bulb temperatures of the ambient air.

The mean bundle temperature is now calculated to determine the air properties at:

$$T_{am} = (T_{ao} + T_{ai})/2 \tag{8}$$

For the heat exchanger, an equation for the characteristic heat transfer parameter can be given in the following format:[11]

$$Ny = a_1 Ry^{b_1} \tag{9}$$

where

$$Ny = h_{ae} A_a / k_a A_{fr} \, \mathrm{Pr}_a^{0.333} \tag{10}$$

and

$$Ry = m_a / (\mu A_{fr}) \tag{11}$$

The coefficients $a_1$ and $b_1$ are determined experimentally. The air mass flow rate is initially chosen as the value corresponding to the maximum fan efficiency on the fan curve. From equation (10) the overall heat transfer coefficient can be calculated:

$$UA_a = [1/(h_{ae} A_a) + 1/(h_p A_p)]^{-1} \tag{12}$$

The number of transfer units is then defined as:

$$NTU = UA_a / C_{min} \tag{13}$$

With the number of transfer units known, a new effectiveness can be calculated and used in equation (6) to calculate a new process fluid outlet temperature.

The normal-flow, isothermal pressure loss coefficient through the heat exchanger bundles is given by:

$$K_{heiso} = a_2 RY^{b_2} \tag{14}$$

where the $K$-value is defined as

$$K = 2\Delta p / (\rho_a v_a^2) \tag{15}$$

As the air flows through the tower, different flow resistances are encountered. The draught equation for a horizontal ACHE with the assumption that $H_3 = H_4 = H_5 = H_6 = H_7 = H_8$ in Figure 2, can be written as:

$$\begin{aligned} & p_{a1} \left[ \begin{array}{c} \left(1 - 0.00975 \frac{H_2 - H_8}{T_{a8}}\right)^{3.5w^*} \\ - \left(1 - 0.00975 \frac{H_2 - H_8}{T_{a1}}\right)^{3.5w*} \end{array} \right] \\ & = K_{ts} \frac{1}{2\rho_{a2}} \left(\frac{m_a}{A_2}\right)^2 + K_{sp} \frac{1}{2\rho_{a2}} \left(\frac{m_a}{A_2}\right)^2 \\ & + K_{de} \frac{1}{2\rho_{a3}} \left(\frac{m_a}{A_{de}}\right)^2 + K_{up} \frac{1}{2\rho_{a5}} \left(\frac{m_a}{A_{Fc}}\right)^2 \\ & - K_{Fsys} \frac{1}{2\rho_{a5}} \left(\frac{m_a}{A_{Fc}}\right)^2 + + K_{do} \frac{1}{2\rho_{a6}} \left(\frac{m_a}{A_{Fc}}\right)^2 \\ & + K_{he} \frac{1}{2\rho_{a78}} \left(\frac{m_a}{A_{fr}}\right)^2 \end{aligned} \tag{16}$$

where

$$w^* = (1 + w)\left[1 - \frac{w}{w + 0.62198}\right] \tag{17}$$

In equation (16), $K_{he}$ includes the isothermal loss coefficient defined in equation (14), a correction due to acceleration effects during non-isothermal conditions and the loss of kinetic energy after the bundle:

$$K_{he} = K_{heiso} + \frac{2}{\sigma^2}\left[\frac{\rho_{a7} - \rho_{a8}}{\rho_{a7} + \rho_{a8}}\right] + 1.13\left[\frac{2\rho_{a7}}{\rho_{a7} + \rho_{a8}}\right] \tag{18}$$

In forced draught heat exchanger arrangements, some of the kinetic energy after the fan is dissipated in the fan plenum. The plenum loss coefficient, $K_{pl}$, is normally expressed as a percentage of the kinetic energy coefficient after the fan, $\alpha_F$, as obtained in a standard installation test.[12; 13]

The fan system pressure rise coefficient can be defined in terms of these values.

$$K_{Fsys} = \frac{2\Delta p_{Fs}}{\rho_{a5} v_F{}^2} + \alpha_F - K_{pl} = K_{Fs} + (\alpha_F - K_{pl}) \tag{19}$$

where $v_F$ is the mean velocity through the fan based on the fan casing cross-sectional area and $\Delta p_{Fs}$ is the fan static pressure rise. In practice there is some uncertainty as to the value of the plenum loss coefficient and all the variables influencing it. A comprehensive CFD analysis was done for this particular plenum and fan arrangement and it was found that 63.9% of the kinetic energy was lost after the fan for $\alpha_F = 2.469$. Equation (19) can therefore be re-written as:

$$\begin{aligned} K_{Fsys} &= \frac{2\Delta p_{Fs}}{\rho_{a5} v_F{}^2} + 2.469\,(1 - 0.639) \\ &= \frac{2\Delta p_{Fs} \cdot \rho_{a5} \cdot A_{Fc}{}^2}{m_a{}^2} + 0.891 \end{aligned} \tag{20}$$

The ACHE operating point is reached when both thermal and pressure balance is obtained. This will be the case when the energy equation (5) as well as the draught equation (16) are satisfied.