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AND PARTICULATE REMOVAL EFFICIENCIES

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CONDENSING ECONOMIZERS: THERMAL PERFORMANCE AND PARTICULATE REMOVAL EFFICIENCIES

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ABSTRACT

Condensing economizers can be used to increase the thermal efficiency of boilers and furnaces. This project has involved a study of these specifically for application to coal-water mixture fuels although the results can be extended to other fuels. Experimental studies to evaluate thermal performance and removal of particulates across indirect contact economizers have been performed. The test arrangement incorporates oil firing with the injection of flyash into the flue gas to simulate coal combustion products. Water sprays into the combustion products are used to achieve variable flue gas moisture content and a variable amount of condensation in the economizers. The economizers are tubular with flue gas on the outside of the tubes. Tube surfaces are plastic coated to prevent corrosion.

The gas temperature and condensation profiles through the economizers have been predicted and overall predicted performance has been compared with test results. Mechanisms for particle removal are discussed and predicted removal efficiencies as a function of particle diameter are presented. It is shown that inertial impaction is the dominant mechanism and particle removal efficiencies up to 89% have been realized.

NOMENCLATURE

English Letter

A_1, A_2, A_3	-	constants dependent upon units
A_t	-	$1/(2Ku)$
C_c	-	Slip correction factor
D_p	-	Brownian diffusivity (Stokes-Einstein relation) $kT C_c / 3\pi\mu_g d_p$
d_p	-	Particle diameter
d_e	-	heat exchanger tube diameter
h_{fg}	-	water vapor latent heat of vaporization
h_i	-	heat transfer coefficient between film and cooling fluid. Includes inside convective coefficient and thermal resistance of the tube and Teflon coating
h_o	-	outside convection coefficient

K_m	-	water vapor mass transfer coefficient
Ku	-	$\alpha - 3/4 - \alpha^2/4 - 1/2 \ln \alpha$ = Kuwabara No. for Kuwabara flow field
k	-	Boltzmann constant
P	-	total pressure (roughly atmospheric)
P_c	-	critical pressure
P_g	-	Peclet No = $d_p U_g / D_p$
P_g	-	atmospheric pressure
P_{wb}	-	water vapor partial pressure in the bulk flue gas
Stk	-	Stokes No. = $\rho_p d_p^2 U_g / (6\mu_g d_p)$
$Stke$	-	effective Stokes No. = $Stk \cdot \text{non-Stokes particle drag correction factor}$
P_{vc}	-	water vapor partial pressure (saturation) at the condensate temperature
T	-	absolute temperature
T_b	-	flue gas bulk temperature
T_c	-	condensate film temperature
T_f	-	cooling fluid temperature
U_g	-	free stream gas velocity
Greek Letter		
α	-	volume fraction of heat exchanger tubes
μ_g	-	gas viscosity
ρ_p	-	particle density

INTRODUCTION

Condensing economizers are used to improve the thermal efficiency of small boilers and furnaces. In these systems, the flue gas temperature is reduced below the dew point of the water vapor. Latent heat is recovered and system efficiency increases markedly. Some smaller heating equipment is available with the condensing heat exchanger integral. In other cases a condensing economizer may be added onto an existing boiler or furnace. This is exclusively the case with commercial and industrial scale equipment and both direct and indirect contact condensing economizers are commercially available. With condensing systems flue gas exit temperatures of about 38 C and boiler or furnace efficiencies (based on gas analysis only) of 95% are not uncommon [1,2].

To date, condensing heat exchangers have been applied primarily to gas-fired equipment. However, there have also been some applications to oil- and wood-fired boilers. Relative to other fuels, the condensate from natural gas is less corrosive. In addition, gas combustion products have higher moisture contents which leads to increased water vapor dew points and increased energy efficiency improvements with condensation. Dew points with 30% excess air for gas, oil, and coal firing are 56, 48, and 41 C, respectively (10% moisture in coal as fired).

An important factor in the application of condensing economizers is the utility of heat at the low temperatures needed for condensation. In the residential sector most of the available products are for gas-fired warm air furnaces. In warm air heating systems the return air temperature to the furnace is generally less than 27 C. Hydronic heating systems, in contrast, have return water temperatures well above 49 C and generally above the dew point of the water vapor in the flue gas. Application of condensing economizers to hydronic systems requires oversized radiators in the heated spaces to reduce the return water temperature. Circulating water to preheat building make-up air has also been used. In the commercial and light industrial sectors preheating of make-up or domestic water is a common application [1,2]. In many applications the flue gas temperature after the economizer is slightly higher than the dew point. These "near-condensing" systems must be built to withstand corrosion resulting from transient or local condensation. Corrosion of condensing heat exchanger surfaces has been an area of considerable emphasis and stainless steels, plastics and glass have been used [3,4].

In addition to removing water vapor, condensing economizers remove particulates from the flue gas. In one field test of a boiler plant fired with #6 fuel oil the particulate removal efficiency across the condensing economizer was found to be 70% [5]. Particle removal is dependent primarily upon the following unit mechanisms:

1. Inertial Impaction
2. Interception
3. Diffusion
4. Particle growth due to nucleation and condensation of water vapor
5. Thermophoresis
6. Stefan Flow

In any practical heat exchanger these are affected by factors including: flue gas temperature and moisture content, heat exchanger surface temperature, excess air, total surface area, gas side pressure drop, and heat exchanger configuration.

In some configurations water sprays may be used, either upstream of the condensing heat exchanger in a presaturator section or in the heat exchanger itself. The sprays can enhance the particulate removal by impacting with the particles or simply increasing the amount of water condensed in the heat exchanger. There has been a considerable amount of work done on the use of saturator/condenser combinations for particle growth and collection.

This study is aimed at the application of condensing economizers to coal water slurry-fired systems with an emphasis on enhancing the particulate removal. The work in this program is specifically aimed at residential, commercial, and light industrial applications

where there would be the most likely need for the low temperature heat source.

Figure 1 shows a general sketch of a condensing heat exchanger fit to a small coal slurry-fired boiler. At the exit from the conventional heat exchanger section there is a saturator section, where water sprays are used to increase the flue gas moisture content at the expense of temperature. Figure 2 shows the effect of water spray rate in the saturator on the flue gas temperature and flue gas water dew point, assuming coal slurry firing and an inlet gas temperature of 232 C. Saturation of the flue gas would be expected to increase condensation rates and particulate removal due to some of the mechanisms discussed above. The heat available in the economizer is degraded by presaturation and the possible benefits of sprays in particle removal must be weighed against this.

Figure 1. General Condensing Heat Exchanger Concept

Figure 2. Saturator Spray Water Effects

Following the saturator the flue gas passes through the condensing economizer and is then exhausted through a "wet stack". In many residential applications plastic exhaust vents are used and, in some cases, these may exhaust horizontally through building walls.

This paper presents experimental results on thermal performance and particulate removal efficiency using

a condensing economizer. These results are then compared with predictions.

EXPERIMENTAL

A sketch of the experimental arrangement used is provided in Figure 3. All measurements and testing have been performed firing No. 2 fuel oil and the experiments were set up to simulate coal-fired conditions with respect to the concentration of fly ash found in the flue gas. Fly ash is introduced into the combustion chamber with a spray assembly that injects an ash/water slurry. Flue gas moisture content is adjusted by injecting water into the flue gas upstream of the economizer. EPA Method 5 procedures are used for particulate sampling from the flue gas upstream and downstream of the economizer. With simultaneous sampling at both points the removal efficiencies under different operating conditions can be obtained. Details of the equipment setup and the procedures used are summarized below.

Heating System

The boiler is a sectional cast iron hot water boiler and has a capacity rating of 586,000 W. All of the tests were done at a firing rate of 82,040 W or less firing No. 2 fuel oil with a pressure atomized, retention head burner. The boiler unit was modified such that the combustion area, including most of the top section, is lined with refractory boards. At such low firing rates maintaining desired flue gas temperature requires that only three of the boiler heat exchanger passages remain open; the other passages are blocked with refractory material. The amount of heat exchanger surface which is effective can be adjusted to trim boiler exit temperature.

Combustion gases leaving the boiler are directed through the condensing economizer using a draft induced, side-wall power venter system. The piping consists of a series of 13 cm diameter flue piping upstream of the economizer and 10 cm diameter flue piping downstream.

A bypass system with dampers was set up to control the volume of flue gas entering the economizer. Flue gas velocities were measured at the inlet and outlet of the condenser using a pitot tube.

Condensing Economizers

Both air and water-cooled condensing economizers have been used in tests. The air cooled unit is a cross-current heat exchanger, constructed of aluminum and the passages and tube side that come in contact with the flue gas are Teflon coated to resist corrosion. Figure 4 provides an illustration of this air-cooled heat exchanger as well as the specifications relevant to thermal performance. The heat exchanger consists of two sections, one stacked on top of the other; each section contains 8 rows with each row containing 9 tubes across. The tubes are 2.5 cm in diameter and are arranged in a staggered order. A 1/3 hp blower is connected to the economizer to draw cooling air through the tubes at an average bulk velocity of 2.16 m/s.

The water cooled heat exchanger, illustrated in Figure 5, is much longer than the air cooled exchanger but has a smaller cross section. The total surface area in the water cooled unit is considerably less. Gas velocities in the water cooled exchanger are about 5 times greater than in the air cooled unit and this should improve particle removal due to inertial impaction.

Water and Ash/Slurry Injection System

Fly ash was obtained from the precipitator hopper at a pulverized coal-fired utility boiler. This material is then mixed with water to make a slurry which is injected directly into the combustion chamber using an air siphon nozzle. The system is mounted at the back end of the furnace, which is the end opposite that of the burner.

Figure 3. Experimental Arrangement

Figure 4. Air-Cooled Condensing Heat Exchanger

Figure 5. Water-Cooled Condensing Heat Exchanger

To modify the flue gas water vapor content, water can also be introduced directly into the combustion

chamber or into the flue pipe, upstream of the economizer, using the same type of spray system. The locations of the spray points used have been selected based on numerous tests in which the water injection rate and the flue gas moisture content were measured. In some configurations the flue gas water content was found to be less than predicted, indicating that not all of the water was evaporating but was instead collecting on duct walls or tops of heat exchanger surfaces. In the configurations used for particulate tests, a good mass balance for the spray water was realized.

Sampling Equipment and Locations

The basic sample components and data acquisition equipment consist of the following:

- a. Thermocouples/ Data Logger
- b. Sample Gas Conditioner
- c. Gas Analyzers
- d. Particulate Sampling Trains - modified EPA Method 5

Thermocouples were installed to measure the flue gas temperature changes from the boiler exit to the vent termination and also economizer cooling water or air at the inlet and outlet. Multiple point temperature measurements were made at the cooling fluid outlet in both cases to monitor differences in the parallel flow paths. In the case of the water cooled economizer, the water flow rate was monitored continuously.

Refer to Figure 3 for the locations of three sampling points in the case of the air cooled economizer. The oxygen content of the flue gas is measured at sample point 1, which is located at the boiler outlet, and is analyzed using a paramagnetic oxygen analyzer.

Sampling points for particulate collection are located upstream and downstream of the economizer, sampling points 2 and 3 respectively. These sampling points are also used to collect water vapor to determine the moisture content of the flue gas. EPA Methods 4 and 5, [6] with modifications to the sampling train, were used as reference procedures for collecting the data. In most cases, the measurements upstream and downstream of the condensing heat exchanger were made simultaneously using two sampling trains.

HEAT TRANSFER DESIGN

The method developed by Colburn and Hougen [7,8,9] has been used to calculate the latent and sensible heat transfer rates, the water vapor condensation rates and the temperature profiles through the heat exchangers. This general method for evaluating the condensation of a single vapor from a non-condensable gas has been specifically applied to flue gas condensing economizers by Razgatis, et al. [10].

A heat balance is prepared where heat flow from the flue gas to the condensate film is equal to the heat flow from the film to the cooling fluid. This heat balance, together with an expression for the relationship between temperature and partial pressure for the water vapor provide relations which can be solved for the unknown - the condensate film temperature. Sensible and latent heat flows follow directly. The approach involves several assumptions including: uniform tube temperature along the length of each tube, negligible thermal resistance of the condensate film, partial pressure of water vapor above the condensate film equal to the equilibrium pressure at the condensate temperature.

The heat balance can be written as:

Sensible Heat Transferred From Flue Gas to the Tube Surface	+	Latent Heat Transferred From Flue Gas the Tube Surface	=	Heat to the Cooling Fluid
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or:

$$h_o \cdot (T_b - T_c) + h_{fg} \cdot K_a \cdot P_g \cdot \ln \left(\frac{P - P_{wc}}{P - P_{wb}} \right) = h_f \cdot (T_c - T_f) \quad (1)$$

For given flue gas and cooling fluid temperatures and flue gas moisture content, the above equation contains two unknowns, P_{wc} and T_c . A relation between these is used to complete the system of equations and Antoine's formula [11] has been used.

The outside film coefficient (h_o) is dependent upon gas properties, flow rates, and row number. Correlations by Kays and London [12] have been used (see also [10]). The water vapor mass transfer coefficient has been calculated by analogy with the outside convection coefficient. The inside film coefficient has been calculated using relations from Kays and London [12]. With the air-cooled case this is complicated by a transition Reynolds number (around 3300 in our experimental arrangement) and a small value of tube length/diameter ratio (6.3). With both the air and the water-cooled exchangers, the length of the tube is significantly longer than the width of the flue gas passage. The part of the tube which extends beyond the duct walls (not in contact with the flue gas), contributes significantly to the effective internal heat transfer resistance. This is particularly important in the case of the air-cooled exchanger, where forced convection from the inside of these tube ends and natural convection from the outside of these ends add 50-70% to the heat transfer rate.

Calculations were done on each section of the heat exchanger assuming constant properties across each section. For the results shown in this paper, each section included two rows of tubes. Calculations with more and less rows in each section did not indicate a significant difference in the overall heat exchanger performance.

The thermal performance of the heat exchangers is defined by the rates of sensible and latent heat transfer, which affect the temperature and moisture content of the exiting flue gas. The performance of both of the heat exchangers included in this program has been predicted using the methodology described above. In this section the predicted thermal performance of the exchangers is presented for two specific cases. The first case assumes that a coal water slurry fuel is being fired with 65% concentration of a Pennsylvania bituminous coal. Flue gas enters the condensing heat exchanger at 177 C and there is no water spray in the saturator. The second case assumes the same fuel but uses a water spray to cool the flue gas down to 85 C before entering the heat exchanger.

Figure 6 shows the predicted performance of the air-cooled heat exchanger in the first case. Included are the temperature profile of the gas as it passes through the exchanger, the flue gas mole fraction of water vapor, and the temperature of the condensate film on the outer surface of the tubes ("Cond. Temp."). Also presented are the rates of heat transfer (sensible, latent and total) on a per-row basis. Overall, in this case, the heat transfer rate is 4,340 W in the exchanger. At the assumed firing rate, 82,100 W, the efficiency improvement across the exchanger is 5.8%. Nearly

all of this is sensible heat transfer, however, and the rate of moisture condensation is very small. Examination of the water fraction curve in Figure 6 shows that the surface falls below the dew point, and water begins condensing only at the very end of the heat exchanger.

Figure 7 shows similar curves for the second case for the air-cooled exchanger, where the flue gas has been cooled in the saturator before entering the condensing heat exchanger. Here latent heat transfer is much more important and most of the heat exchanger rows are wet. Overall, the heat transfer rate is considerably less than in the first case, only 3,020 W. This provides an efficiency increase of 3.6%.

The performance of the water-cooled heat exchanger in Case 1 is shown in Figure 8. For all row numbers greater than 13 the surface is below the dew point and latent heat contributes significantly to the total heat transfer rate. Clearly this is doing a much better job of both cooling the gas and removing water vapor than the air cooled case. Overall the heat transfer rate is 7,330 W and the efficiency improvement is 8.9%.

The water-cooled heat exchanger, Case 2, is presented in Figure 9. Again, the addition of water to the flue gas increases the importance of latent heat in the exchanger and all of the surfaces are wet. As in the air-cooled case the addition of water to the flue gas reduces the total heat transfer rate but the effect is not nearly as great. A total of 6,150 W is transferred in this case for an efficiency improvement of 7.4%.

PARTICULATE REMOVAL PREDICTION

In this section the mechanisms which could be expected to influence particulate removal across a condensing economizer are discussed and estimates of the particulate removal efficiency as a function of particle size are presented for the specific economizers used in this work. The following assumptions are made in this analysis:

- The gas flowing around the heat exchanger tube can be treated as a continuum.
- The heat-exchanger tube is treated as an isolated cylinder in cross-flow, with forced-convection dominating natural convection.
- The sticking efficiency is unity because of the condensate on the wall of the heat exchanger tubes.
- The slip correction factor is introduced into the Brownian diffusivity (Stokes-Einstein relation).
- To obtain efficiencies of tube bank heat exchangers we can determine the particle collection efficiency of an isolated single cylinder and then combine these for the entire exchanger.

* Inertial Impaction

Inertial impaction occurs when a particle is unable to follow the rapidly curving streamlines around an obstacle and, because of its inertia, continues to move toward the obstacle along a path of less curvature than the flow streamlines. Thus, collision occurs because of the particle's momentum. Note that the mechanism of inertial impaction is based on the premise that the particle has mass but no size, whereas interception is based on the premise that the particle has size but no mass.

Figure 6. Predicted Performance of Air-Cooled Condensing Heat Exchanger - Case I

Figure 7. Predicted Performance of Air-Cooled Condensing Heat Exchanger - Case II

Figure 8. Predicted Performance of Water-Cooled Condensing Heat Exchanger - Case I

Figure 9. Predicted Performance of Water-Cooled Condensing Heat Exchanger - Case II

Table 1. Summary of Isolated Single Collection Efficiencies

Mechanism	Efficiency	References	Remarks
1. Brownian diffusion	$\eta_B = 3.68A_p^{1/3}P_p^{2/3}$	Friedlander [15]	Diffusing mass less points
2. Thermophoresis	$\eta_{TS} = \eta_B \cdot F(\text{Suction})$	Rosner [16]	Hot to Cold
3. Interception	$\eta_{IX} = 2A_p \left(\frac{d_p^2}{d_v} \right)$	Flagan et al. [17]	Neglect of particle inertia ($d_p/d_v \ll 1$)
4. Inertial Impaction	$\eta_{IM} = 0.01978749 \ln(8 \text{ Stke}) + 0.5136545 (\text{Stke} - 0.125) - 0.0482858 (\text{Stke} - 0.125)^2$	Wessel et al. [14]	0.125 < Stke < 0.5
	$\eta_{IM} = (1 + 1.54424 (\text{Stke} - 0.125)^{-1} - 0.538013 (\text{Stke} - 0.125)^{-2} + 0.2020116 (\text{Stke} - 0.125)^{-3})^{-1}$		0.5 < Stke

Figure 10. Results of Prediction of Particulate Capture with Air-Cooled Condenser

Figure 11. Results of Prediction of Particulate Capture with Water-Cooled Condenser

* Interception

Interception takes place when a particle, following the streamlines of flow around a cylinder, is of size sufficiently large that its surface and that of the cylinder come into contact. Thus, if the streamline on which the particle center lies is within a distance $d_p/2$ of the cylinder, interception occurs.

* Brownian diffusion

Particles in a gas undergo Brownian diffusion that will bring some particles in contact with the cylinder due to their random motion as they are carried past the cylinder by the flow. A concentration gradient is established after the collection of a few particles and acts as a driving force to increase the rate of deposition over that which would occur in the absence of Brownian motion. Because the Brownian diffusivity of particles increases as particle size decreases, the removal mechanism will be most important for very small particles. When analyzing collection by Brownian diffusion, the particles are treated as diffusion massless points.

* Thermophoresis

Heavy molecules and particles in a temperature gradient experience a force (usually in the direction from hot to cold, which produces a net transport called thermophoresis. Thermophoretically driven mass transport is important in some gaseous systems (e.g., flame "structure" and surface combustion of hydrogen (air) and most aerosol systems (soot and ash deposition on cooled heat-exchanger surfaces, sampling targets, and/or turbine blades) as shown in Rosner and Fernandez dela Mora [13].

For each of the mechanisms discussed above a collection efficiency can be defined. For a single cylinder this efficiency is the rate of particle collection divided by the mass flow rate of particulates through an area defined by the tube diameter and length. Table 1 presents the summary of relations used for isolated single cylinder collector efficiencies for four different collection mechanisms. In this analysis the effects of Stefan flow and particle growth due to condensation have not been included. These would be expected to provide some increase in particle collection.

For a single cylinder, the combined collection efficiency due to all of the effects considered is:

$$\eta_c = 1 - (1 - \eta_D) \cdot (1 - \eta_{TM}) \cdot (1 - \eta_{IN}) \cdot (1 - \eta_{IN}) \quad (2)$$

Using this the collection efficiency for the first row of tubes is:

$$\eta_{1st} = \eta_c \cdot \frac{\text{total tube proj. area}}{\text{total cross section area}} \quad (3)$$

For a heat exchanger with n rows the collection efficiency can be calculated as:

$$\eta_{nrows} = 1 - ((1 - \eta_{1st}) \cdot (1 - \eta_{2nd}) \cdot \dots \cdot (1 - \eta_n)) \quad (4)$$

The predicted collection efficiency for both the air and the water cooled economizers used in this work have been estimated using the approach above as a function of particle size. The heat transfer conditions used are those shown in Figures 6 and 8. Based upon these conditions, the particle collection efficiency for four different mechanisms as shown in Table 1 were evaluated. Figure 10 shows the collection efficiency as a function

of the particle size for the air cooled case. The particle collection efficiency from interception is not shown because the value is negligibly small. For larger particles inertial impaction is clearly dominant. For the smaller particles thermophoresis is shown to greatly increase the particle collection efficiency over isothermal diffusion although the removal efficiency for the small particles is very small. The water-cooled condenser as shown in Figure 11 clearly presents the greater potential for particle removal and this is primarily due to the increased gas velocities around the tubes.

EXPERIMENTAL RESULTS

Results are presented for the air and water-cooled units in Tables 2 and 3, respectively. Along with the measured outlet temperatures and flue gas moisture contents values for these parameters predicted using the methods discussed earlier are presented.

With regard to heat transfer performance the calculation method discussed in the previous section did fairly well in terms of predicting the outlet gas temperature and the condensation rate. The particulate removal efficiency was found to range between 42 and 66% for the air cooled case and was found to be 89% in the water cooled case. The particle removal efficiency shows no significant trend with the inlet flue gas moisture content or the total amount of water vapor condensed. Increasing the flue gas water vapor content does not appear to yield any significant benefit in particulate collection.

DISCUSSION

An estimate of the particle removal efficiency which might be expected in these heat exchangers can be made using a simple "cut diameter" approach. In this approach the particle diameter for which a 50% particle removal efficiency is expected is identified and then it is assumed that all particles larger than this size are removed and all small particles pass through. In the case of the air-cooled condenser the cut diameter is about 23 microns (Figure 10). Based on typical size distributions for pulverized coal flyash [18] 50 to 70% of the ash is smaller than this size. Removal efficiency then would only be expected to be about 30 to 50%. The present results show somewhat higher removal efficiencies.

For the water-cooled condenser the cut diameter is about 7 microns. Based on this and typical flyash size distribution the removal efficiency would be expected to range from 60 to 80%. Tests results presented above indicated 89%.

The predictions of particle removal efficiency presented in this paper have neglected the consideration of some mechanisms, including particle growth due to condensation and Stefan flow. These may be leading to somewhat greater particle removal than predicted. Generally, however, the removal of particles appears to be still dominated by simple inertial impaction and consideration of this alone can be used for at least preliminary estimates of performance.

TABLE 2 - AIR COOLED CONDENSING ECONOMIZER

EXPERIMENTAL DATA

Run	1	2	3	4	5	6	7	8	9
Firing rate, W	82,000	82,000	82,000	62,000	62,000	62,000	62,000	62,000	62,000
% Ash in slurry	10	10	10	10	10	10	10	10	10
Slurry Spray Rate cm ³ /s	1.0	1.0	1.0	1.08	1.05	1.05	1.05	1.37	1.26
Particulate Conc. Inlet, gm/dscm	0.706	0.847	0.777	0.742	0.600	0.494	0.512	1.45	1.59
Particulate Conc. Outlet, gm/dscm	0.388	0.494	0.332	0.247	0.212	0.212	0.282	0.586	0.561
Moisture, Inlet % (measured)	24	15	14	ND	22	22	20	14	13
Moisture, Outlet % (measured)	22	10	11	ND	20	17	15	12	12
Moisture, Outlet % (calculated)	20.4	14.3	13	—	18.9	16.5	14.7	12.1	
Particle Removal Efficiency	45	42	57	66	63	55	45	60	65
Flue gas temperature Inlet, C (measured)	177	191	193	127	143	81	81	127	133
Flue gas temperature Outlet, C (measured)	74	71	74	54	66	56	52	53	71
Flue gas temperature Outlet, C (calculated)	77	71	69	—	71	61	58	60	

ND -- No data

TABLE 3 - WATER COOLED CONDENSING ECONOMIZER

EXPERIMENTAL DATA

Run	1	2
Firing rate, W	62,000	62,000
% Ash in slurry	10	10
Slurry Spray Rate, cm ³ /s	0.74	0.84
Particulate Concentration Inlet, gm/dscm	0.625	0.516
Particulate Concentration Outlet, gm/dscm	0.0660	0.0576
Moisture, Inlet % (measured)	11	10
Moisture, Outlet % (measured)	6	7
Moisture, Outlet % (calculated)	5.1	7.6
Particle Removal Efficiency, %	89	89
Flue gas temperature Inlet, C (measured)	146	97
Flue gas temperature Outlet, C (measured)	45	44
Flue gas temperature Outlet, C (calculated)	39	46

CONCLUSIONS

The thermal performance and particulate removal efficiency of condensing economizers applied to flue gas from coal water slurry-fired boilers or furnaces has been evaluated. With regard to heat transfer performance, the calculation method developed by Colburn and Hougan was found to reasonably predict outlet temperature and water vapor content. The mechanisms which could be expected to influence particulate removal across a condensing economizer are discussed and estimates of the particle removal efficiency as a function of particle size are presented for the specific economizers studied. Isolated single cylinder collector efficiencies for diffusion, thermophoresis, interception, and inertial impaction are evaluated. Inertial impaction is found to be the dominant mechanism for flyash with particle diameters in the range of 1 to 50 microns. For heat exchangers with multiple tube rows, the collection efficiency can be approximated by extending the single tube results. Collection efficiencies were found to be somewhat higher than predicted for typical flyash, indicating that some mechanisms not included in the prediction may be contributing. Inertial impaction, however, can still be used to at least roughly predict performance.

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