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CONDENSING ECONOMIZERS FOR SMALL
COAL-FIRED BOILERS AND FURNACES
Project Report - January 1994

Thomas A. Butcher and Wailin Litzke

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Energy Efficiency and Conservation Division

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CONDENSING ECONIMIZERS FOR SMALL COAL-FIRED BOILERS AND FURNACES

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January 1994

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ABSTRACT

Condensing economizers increase the thermal efficiency of boilers by recovering sensible and latent heat from exhaust gas. These economizers are currently being used commercially for this purpose in a wide range of applications. Performance is dependent upon application-specific factors affecting the utility of recovered heat. With the addition of a condensing economizer boiler efficiency improvements up to 10% are possible.

Condensing economizers can also capture flue gas particulates. In this work, the potential use of condensing economizers for both efficiency improvement and control of particulate emissions from small, coal water slurry-fired boilers was evaluated. Analysis was done to predict heat transfer and particulate capture by mechanisms including: inertial impaction, interception, diffusion, thermophoretic forces, and condensation growth. Shell-and-tube geometries were considered with flue gas on the outside of Teflon-covered tubes. Experimental studies were done with both air- and water-cooled economizers refit to a small boiler. Two experimental arrangments were used including oil-firing with injection of flyash upstream of the economizer and direct coal water slurry firing. Firing rates ranged from 27 to 82 kW (92,000 to 280,000 Btu/hr). Inertial impaction was found to be the most important particulate capture mechanism and removal efficiencies to 95% were achieved. With the addition of water sprays directly on the first row of tubes, removal efficiencies increased to 98%. Use of these sprays adversely affects heat recovery. Primary benefits of the sprays are seen to be the addition of small impaction sites and future design improvements are suggested in which such small impactors are permanently added to the highest velocity regions of the economizer. Predicted effects of these added impactors on particulate removal and pressure drop are presented.

ACKNOWLEDGMENTS

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SECTION 1. INTRODUCTION

BACKGROUND

Condensing economizers are heat exchangers which recover energy from flue gas leaving boilers. In these systems, the flue gas temperature is reduced below the dew point of the water vapor. Both sensible and latent heat are recovered from the flue gas and boiler system thermal efficiency increases markedly. Some residential scale heating appliances are now commercially available with integral condensing economizers. In other cases a condensing economizer may be added onto an existing boiler or furnace. With commercial and industrial scale equipment these economizers have been used to a limited degree and are always refit to conventional boilers. Both direct and indirect contact economizers are available although the indirect are more common. With condensing systems flue gas exit temperatures of about 38 C (100 F) and boiler or furnace system thermal efficiencies (based on gas analysis only and fuel higher heating value) of 95% are not uncommon [1,2].

To date, condensing economizers have most commonly been used on 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 content which leads to increased water vapor dew points and increased energy efficiency improvements with condensation.

An important factor in the application of condensing economizers is the utility of heat at the low temperatures required 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 (80 F). Hydronic residential heating systems, in contrast, often have return water temperatures well above 49 C (120 F) and generally above the dew point of 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.

In many applications the flue 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, Teflon coatings or coverings, Teflon heat exchangers, glass heat exchangers, and plastic exhaust system components have been used [3,4]. In gas-fired units, where acid attack is less severe, economizers have been built with mild or stainless steel. In the economizers used in the experimental part of this project, all surfaces exposed to the flue gas were manufactured with a thin covering of Teflon to prevent acid attack.

Utility applications of condensing economizers have been very limited to date. Recently Consolidated Edison Company of New York has installed a condensing economizer at their 74th Street Station in a test/demonstration effort. This site has both steam send-out and electric capacity and the economizer simply preheats cold make-up water. The economizer is on a slipstream of the flue gas flow from three oil-fired boilers with a steaming capacity of 272,000 kg/hr (600,000 lbs/hr) each. Two economizers were designed to handle a total of 145,000 kg/hr (320,000 lbs/hr) of flue gas, which represents one third of the flue gas flow at average conditions. The typical gas temperature entering the economizer is 163 C (325 F). The gas leaves the economizer at 29 to 38 C (85 to 100 F) and the mixed gas enters the stack at about 121 C (250 F). This arrangement avoids the need for a new, corrosion resistant stack for the economizer exhaust. The improvement in the plant heat rate with the economizer averages 800 Btu/kWh. It is planned to keep the economizer in operation for 20 years [5].

An analysis of the potential application of condensing economizers to closed cycle power generation plants has been done by Foster Wheeler Development Corporation [6]. In contrast to the Consolidated Edison Co. application described above these plants do not have very substantial make-up water heating loads. When used in conventional reheat cycles producing electricity at 35% cycle efficiency the heat rate reduction which might be realized using the economizer for feedwater preheating was found to be 2 to 4%. Potential heat rate reduction was found to be greater in non-reheat cycles and lower in combined cycle power plants.

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 was found to be 70% [7]. In this case particles were captured on the outside (gas side) surfaces of the heat exchanger and washed off with the condensed water from the flue gas. Ideally particulate capture in such a system could be affected by the following mechanism:

Inertial Impaction
Interception
Diffusion of particles to the economizer surface
Particle growth due to nucleation and condensation of water vapor
Thermophoresis

In any practical heat exchanger these are affected by: flue gas temperature and water vapor content, heat exchanger surface temperature, excess air, total surface area, gas velocity, and heat exchanger configuration. In some configurations water sprays may be used, either upstream of the heat exchanger in a presaturator section or in the economizer 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 particulate growth and downstream collection by inertial mechanisms [8].

A two-stage condensing economizer concept has been developed by Condensing Heat Exchanger Corporation of Warnerville, N.Y. and is intended to act as an Integrated Flue Gas Treatment System (IFGT), combining heat recovery, particulate capture, and flue gas desulfurization. It involves two separate economizer sections. The first recovers sensible heat bringing the flue gas close to saturation. At the exit of this section flue gas passes through a fiberglass reinforced plastic transition/contactor section which has sprays of a reagent solution for SO_X capture. The second heat exchanger section provides most of the latent heat recovery. Initial tests of the IFGT were done at a 6360 kg/hr steam (14,000 lb/hr) boiler firing #6 oil located outside of Albany, N.Y.. Results were very encouraging using sodium bicarbonate as a desulfurization reagent. Removal efficiencies were: 89.3% for particulates and over 99% for SO_2 and SO_3. Mercury was removed with an efficiency of 84%. The Consolidated Edison Company of New York is currently involved in a more comprehensive testing program with two-stage pilot unit at their Ravenswood Station. In addition to the two heat exchanger sections this pilot unit has both a mesh type and centrifugal mist eliminator at the system exit. Water sprays are located at several positions to saturate the flue gas, assist in particulate removal and increase the liquid-to-gas transfer ratio when reagent is used for SO_2 removal. The evaluation effort includes detailed measurements of particulates, SO_x, and trace metal emissions over a range of conditions. Tests to date have shown particulate emissions to be limited to 0.005 lb/million Btu and about half of this is sulfuric acid mist [5].

OBJECTIVES

The work conducted during this project and described in this report has been very specifically aimed at the application of condensing economizers to small boilers and furnaces which are fired with coal-water slurries. The use of these economizers for thermal efficiency improvement and also as the sole device used for controlling particulate emissions may greatly improve the economic attractiveness of coal in these applications. Emphasis was placed during the project on understanding and enhancing particulate capture. The work has included both analysis of expected thermal and particulate capture performance as well as experimental studies. This project was planned to contribute to programs at the Pittsburgh Energy Technology Center (PETC) in the development of coal-fired combustion equipment for commercial and residential applications.

REPORT ORGANIZATION

The work described in this report includes both analysis and experimental components. Section 2 of the report describes the construction of the specific economizers used in both the analysis and experimental work. This section also provides a description of the experimental arrangement used. Details of the methods used to predict heat transfer and particulate capture are not included in the main part of the report but rather in the appendix in consideration of those who wish to avoid involvement with the details of this analysis. Appendix A covers heat transfer and Appendix B particulate capture. In Section 3 the results are presented of analyses done for the specific heat exchangers described earlier in Section 2. A summary of all of the experimental results is given in Section 4. Section 5 provides a discussion of the results including possible reasons for differences between predicted and measured particulate removal, suggestions for future improvements in particulate removal, and the potential that these economizers have for meeting emission limits with coal-water slurry firing. Finally, conclusions of the work are summarized in Section 6.

SECTION 2. EXPERIMENTAL

GENERAL ARRANGEMENT

A schematic of the experimental arrangement used is shown in Figure 2-1. The main components of this setup consist of the heating system, water and slurry spray systems, the condensing economizer, and multiple sampling points used for flue gas analysis. Two types of condensing economizers, an air-cooled and a water-cooled unit, were evaluated for thermal and particulate removal performance.

As part of this study, the measurements and testing were conducted in two ways: (1) firing No. 2 oil, and fly ash spray-dried into the flue gas stream to simulate coal-firing conditions with respect to the concentration of fly ash; and (2) with actual coal water slurry firing. Most of the tests were conducted with the oil-fired case. The purpose for these two different experimental setups was to initially demonstrate economizer performance without the added complications of coal-slurry firing, and then to compare and/or validate those results with a few tests consisting of actual coal-slurry firing.

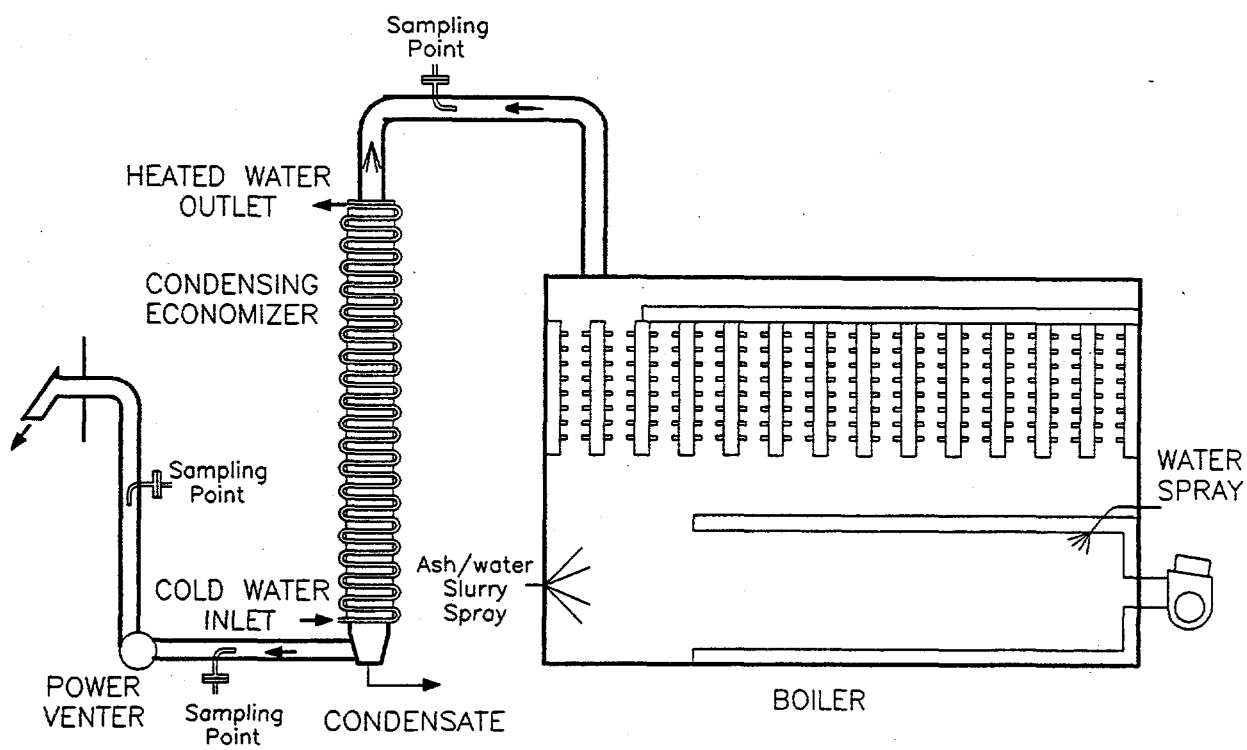

Figure 2-1. Experimental Arrangement (shown with water-cooled economizer installed)

In the oil-fired tests fly ash was introduced into the combustion chamber with a spray assembly that injects an ash/water mixture. The ash was obtained from a precipitator hopper of a coal-fired utility and it was assumed that this provided a representative size distribution for fly ash in the flue gas. The mixture consisted of 10-15% solids (ash) in water. This mixture could then be sprayed with a siphon-type air atomizing nozzle into the combustion chamber through the back end of the boiler, opposite from the burner assembly. Flue gas moisture content was adjusted by spraying water with a siphon nozzle into the flue gas stream at locations upstream of the economizer or directly into the combustion chamber. The air atomized, siphon nozzle provides a low flow rate and a very fine spray which helped to assure that all of the water was vaporized. The moisture content of the flue gas was measured with a sampling train as prescribed in EPA Methods for monitoring of emissions from combustion sources (Environmental Protection Agency, Code of Federal Regulations 40 CFR 60). Samples were taken simultaneously with multiple sampling trains at locations upstream and downstream of the economizer and , in some tests, at the outlet of the induced draft fan.

The locations of the spray points used were selected based on numerous tests in which the water injection rate and the flue gas moisture content were measured. A mass balance on the amount of water input, assuming total evaporation, and the amount measured indicated that in some configurations the spray system was more effective than in other configurations. When the flue gas water content was found to be less than predicted, and therefore not all of the water was evaporated, it was instead collecting on duct walls or on tops of the heat exchanger surfaces. In the configurations used for the particulate tests, a good mass balance for the spray water was realized.

An in-stack, multistage, cascade impactor which adapts to the EPA sampling train was used to measure size distribution of particulate emissions. With simultaneous sampling at the various locations the removal efficiencies with respect to particle size under different operating conditions was determined.

Some of the data was collected with coal-water slurry firing. The slurry was provided by Tecogen Inc. and some of its properties are listed in Table 2-1. At the low firing rates at which these studies were performed atomizer pluggage was a particular concern. Several air-atomized nozzles were tested. The most consistent performance was obtained using an unmodified "Aero" nozzle supplied by Delavan Inc. Coal water slurry was strained through a 16 mesh screen just before firing and then fed to the atomizer using a peristaltic pump.

BOILER

The boiler used is a sectional cast iron hot water boiler, manufactured by Peerless Industries, Inc., and has a capacity rating of 586 kW (2,000,000 Btu/hr). However, all of the tests were done at a firing rate of 82 kW (280,000 Btu/hr) or less. These lower firing rates were used simply to match the economizers used. In the oil-fired case, No. 2 fuel oil was burned with a pressure atomized, flame retention head burner. At such low firing rates maintaining the desired flue gas temperatures required that the boiler unit be modified such that the combustion area including most of the top section was lined with refractory boards. In addition, most of the boiler heat exchanger sections were blocked with refractory material and only several of the passages remain open. The amount of heat exchanger surface which was effective could be adjusted to trim boiler exit temperature.

Combustion gases leaving the boiler were directed through the condensing economizer and then exhausted outside with an induced draft fan, (also called a "side wall power venter"). A bypass system with dampers was set up as part of the flue piping to control the volume of flue gas entering the economizer. Flue gas velocities were measured at the inlet and outlet of the condenser with a standard pitot tube.

Table 2-1. Properties of the Coal-Water Slurry Used in Combustion Tests

Coal Properties		**Slurry Properties**	
Type:	Kentucky Hazard	Coal Concentration:	59.%
Ultimate Analysis (% dry basis):		Specific Gravity:	1.15
C	81.09	Heating Value	19,500 kJ/kg (8500 Btu/lb)
H	5.39		
N	1.67		
S	0.76		
Ash	3.72		
O	7.37		
Proximate Analysis (% dry basis):			
Volatiles	36.92		
Fixed Carbon	59.36		

COAL WATER SLURRY BURNER

The burner arrangement used to fire the coal water slurry is illustrated in Figure 2-2. The system includes a direct-fired air preheater which burns kerosene. In some of the coal water slurry tests this burner was used to preheat the combustion air to 120 C (250 F). Under these conditions the oxygen content of the vitiated combustion air is about 19.5%. Strong burner swirl is achieved using axial swirl vanes at the burner head. Typical pressure drop across this head is 750 Pa (3 inches of water). The burner can also directly co-fire #2 fuel oil using a pressure atomizer adjacent to the air-atomized slurry nozzle. This was used for initial warm-up of the combustion chamber refractory liner and the boiler itself. During some of the combustion tests with coal water slurry the #2 oil was co-fired to assure stable conditions at the economizer. One test was conducted without oil co-firng at the head. The back end of the "nozzle cup" (see Figure 2-2) is perforated to allow a small part of the combustion air to flow around the nozzles, preventing coking.

CONDENSING ECONOMIZERS

In the experimental work both air- and water- cooled condensing economizers, built by Condensing Heat Exchanger Corp., have been evaluated. Figure 2-3 provides an illustration of the air-cooled heat exchanger along with the specifications relevant to thermal performance calculations. A photo of this economizer installed during the test program is included here as Figure 2-4. This unit is a cross-current tube and shell heat exchanger constructed of aluminum. The passages and tube surfaces that come in contact with the flue gas are covered with Teflon to resist corrosion. The heat exchanger consists of two sections, one stacked on top of the other; each section contains eight (8) rows with each row containing nine (9) tubes across. The tubes are 2.5 cm (1 inch) in diameter and are arranged in a staggered order. A 0.25 kW (1/3 hp) blower was connected to the economizer to draw cooling air, at ambient conditions, through the inside of the tubes at an average bulk velocity of 2.2 m/s (3.3 f/s). The thickness of the Teflon covering on both the air and water-cooled economizers was 0.381 mm (0.015 inches).

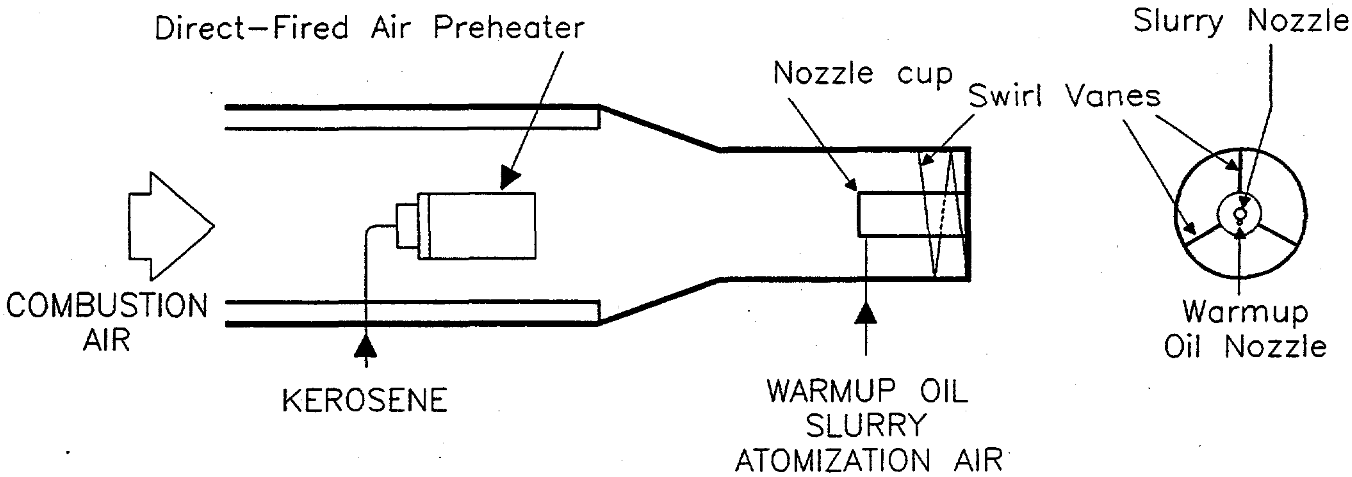

Figure 2-2. Burner arrangement used for coal water slurry firing

The water-cooled heat exchanger illustrated in Figure 2-5 is much longer than the air-cooled unit but has a smaller cross section. The total surface and minimum flow areas in the water-cooled unit are considerably less than in the case of the air-cooled economizer. Gas velocities in this unit are about five times greater than in the air-cooled unit and this improves particle removal due to inertial impaction.

Total particulate removal across these heat exchangers, removal based on particle size distribution, and the thermal performance were measured and evaluated based on varying heating system firing rate, condensation rate, water spray locations, and gas side pressure drop. The methods developed to predict economizer performance were also used for comparison with these experimental results.

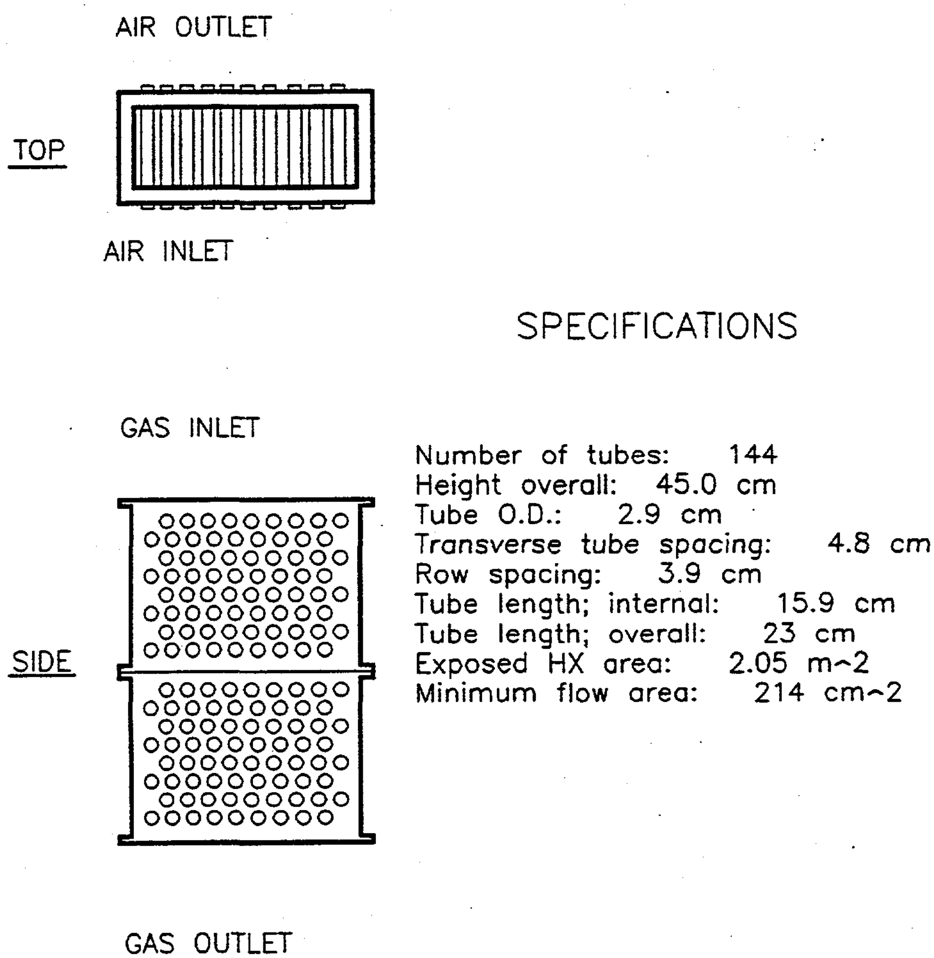

Figure 2-3. Illustration of air-cooled condensing economizer.

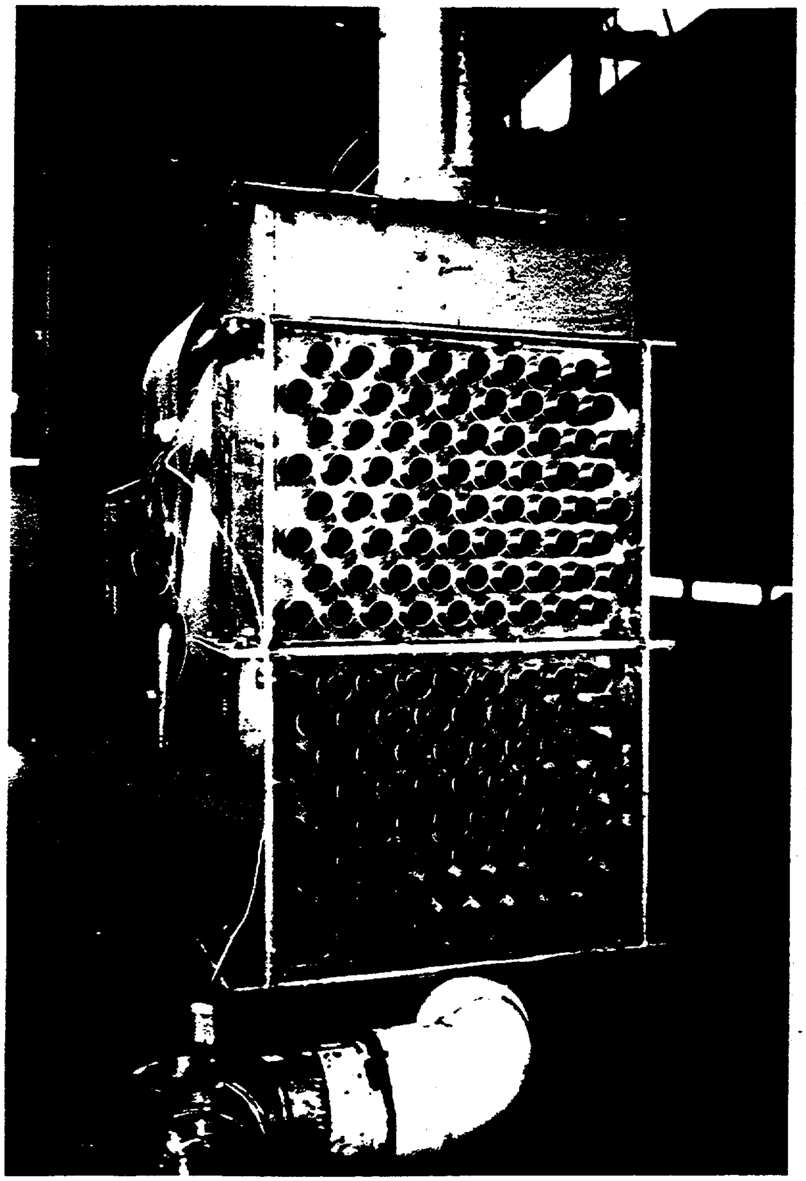

Figure 2-4. Photo of air-cooled condensing economizer

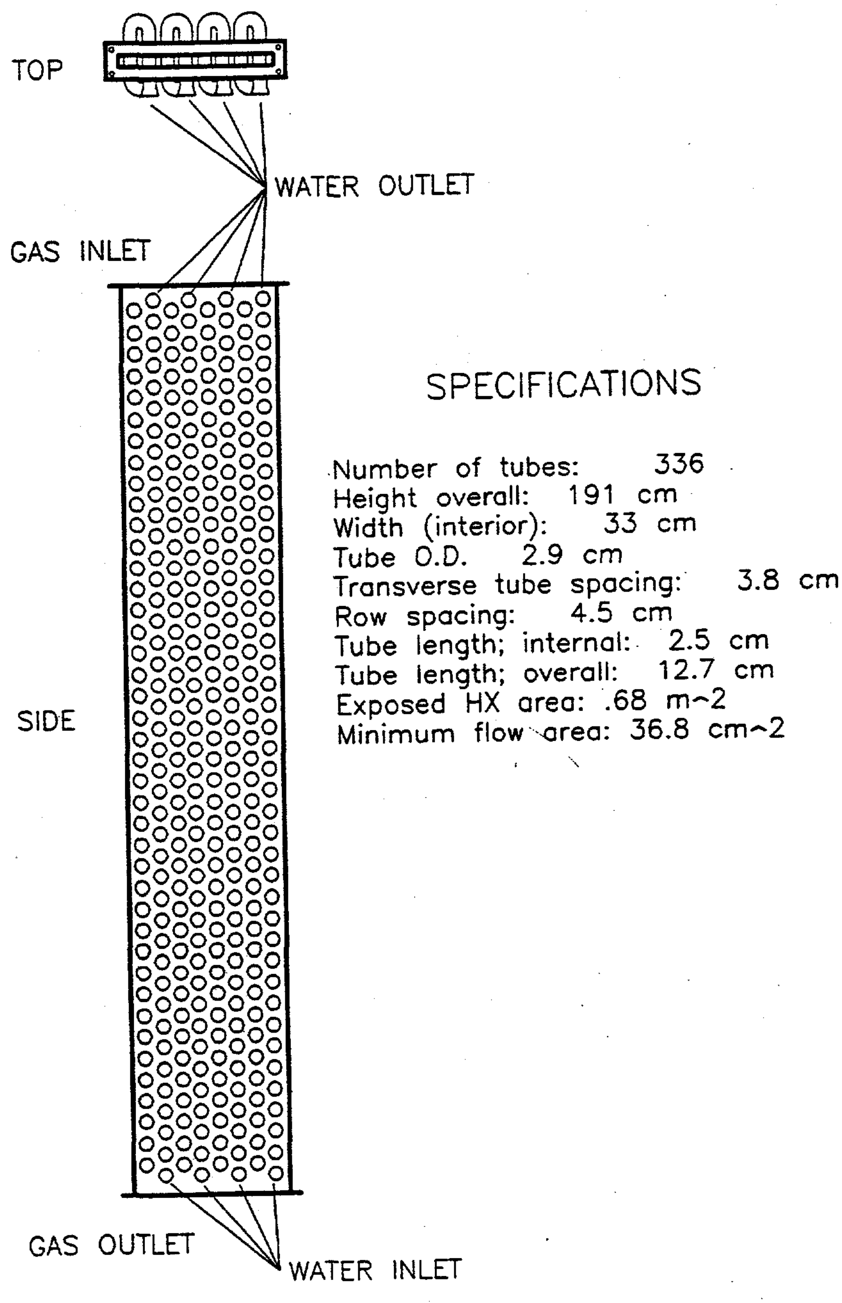

Figure 2-5. Illustration of water-cooled condensing economizer

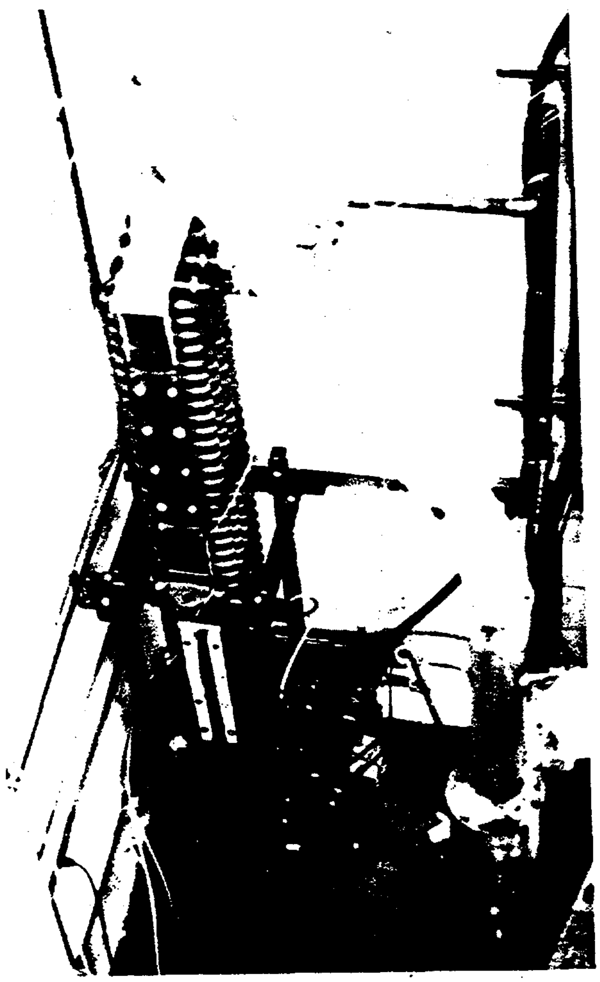

Figure 2-6 Photo of water cooled condensing economizer

SAMPLING METHODOLOGY

The particulate testing procedures and equipment are based on EPA Method 5 for evaluating the standard of performance for stationary sources. The basic components of the sampling train consist of the nozzle, heated filter holder, a set of condenser impingers, sampling pump and dry gas meter. The filter holder contains a glass fiber filter mat for particulate capture and the entire holder was maintained at 120 ± 14 C. (248 ± 25 F) The material collected on the filter in addition to that collected in the nozzle, which was rinsed out with acetone, were weighed to give the total mass of particulate matter. The condenser impingers were kept at 0 C (32 F)with an ice bath and the total volume of water collected was measured. The volume of flue gas sampled was measured with a dry gas meter. The particulate matter is defined as the material collected on the filter or rinsed from the sampling nozzle and filter holder.

A multistage, in-stack cascade impactor (manufactured by Anderson Samplers, Inc., Model Mark III) was used for simultaneous measurement of the particle size distribution and total mass concentration of the particulate matter in the flue gas. This in-stack sampler adapts to the standard EPA-5 sampling trains. Particles with aerodynamic diameters under 20 microns can be separated onto eight stages on glass fiber collection media.

Thermocouples were installed throughout the system to measure the flue gas temperature changes from the boiler to the final exhaust. In addition, temperatures were measured of the economizer cooling medium - 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 through the unit was monitored continuously with a strain gauge target flowmeter. All data recorded were stored on a multichannel datalogger interfaced with a desktop computer.

The sampling points for particulate, moisture content, and gas analyses were located upstream and downstream of the economizer, and at the outlet of the induced draft fan. Simultaneous measurements with multiple sampling trains allowed for comparisons of the analyses for the same operating conditions. A standard type pitot tube was used to measure flue gas velocity at the sampling locations to determine sampling nozzle size and flow rate. The oxygen content of the flue gas was measured at various points to determine excess air or dilution air through the piping system.

SECTION 3. ANALYSIS OF EXPECTED PERFORMANCE

DEW POINT AND POTENTIAL ENERGY RECOVERY

To date, the most popular application for condensing economizers has been gas-fired equipment, although there have also been some oil-and wood-fired systems. Condensate from natural gas is less corrosive. In addition, gas combustion products have higher moisture contents leading to increased water vapor dew points and increased energy efficiency improvements with condensation. Combustion products from coal-fired boilers have low water vapor contents because of low fuel hydrogen content and, so, these are poor candidates for condensing systems. However, burning coal as a water slurry obviously increases flue gas water content and dew point and therefore has greater potential for improved efficiency with condensing economizers. Table 3-1 provides a comparison of water vapor content and dew point for selected fuels.

Table 3-1. Effect of Fuel Type on Flue Gas Water Vapor Content and Water Vapor Dew Point
Assumptions: 30% excess air, typical eastern bituminous coal.

FUEL:	Flue Gas Volume Fraction Water Vapor	Water Vapor Dew Point-C (F)
Natural Gas	.16	56 (132)
Distillate Oil	.11	48 (118)
Coal	.08	41 (106)
Coal-Water Slurry	.12	50 (122)

Energy efficiency gains that can be achieved using a condensing economizer are shown in Figure 3-1. The efficiency shown only includes consideration of energy losses with the sensible and latent heat content of the flue gas. Radiation losses from the boiler exterior, unburned carbon losses, losses credited to auxiliary electric power and other smaller losses are not considered. Adding these factors would reduce all efficiency values by about 3-5%. For a given flue gas discharge temperature the efficiency decreases with increasing fuel water content. Reducing the flue gas temperature to the lowest level shown leads to recovery of nearly all of the latent heat losses and eliminates the efficiency penalties of the added fuel water content. In the case of the coal water slurry with additional water added to the flue gas ("65/45 CWS+spray"), flue gas water content has been increased to about 25 volume %.

Figure 3-2 shows the ideal amount of water which can be condensed from flue gas as a function of flue gas temperature. Again coal, a coal water slurry, and a coal water slurry with flue gas water sprays are shown, With dry coal firing the potential water condensation rate is very small and

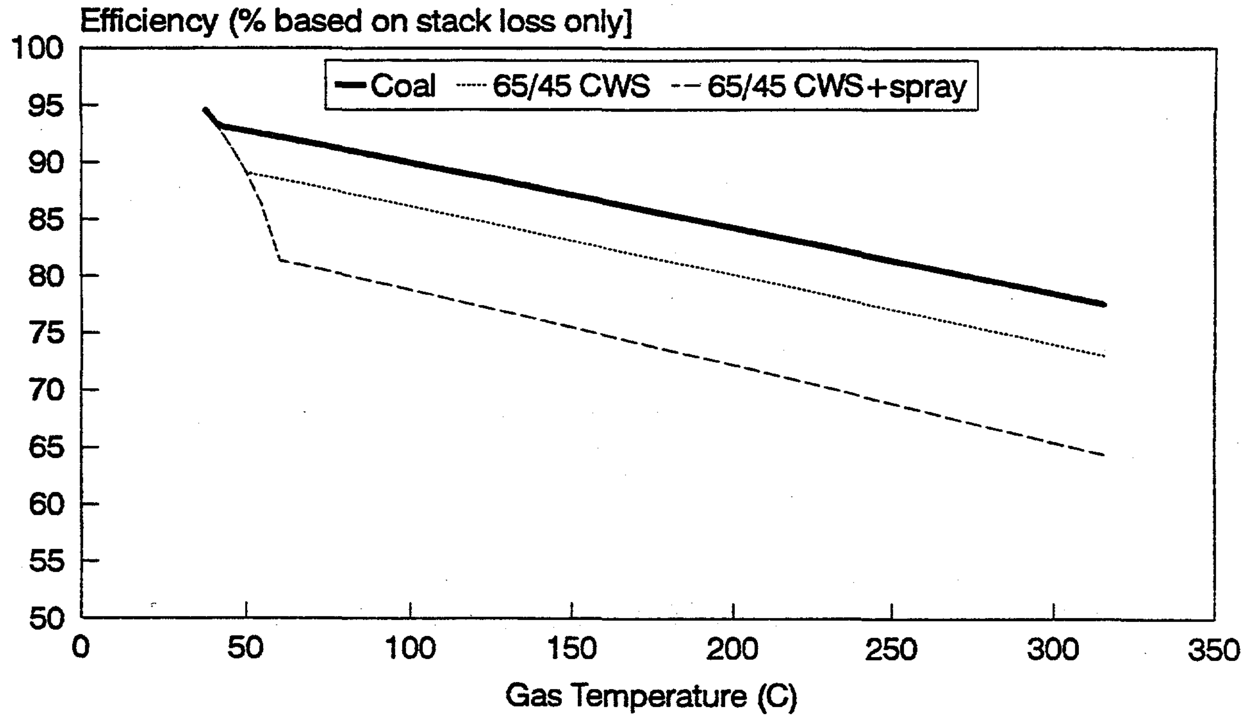

Figure 3-1. Effect of flue gas discharge temperature on boiler thermal efficiency (gas loss only). Coal, coal water slurry and coal water slurry with additional water added to flue gas.

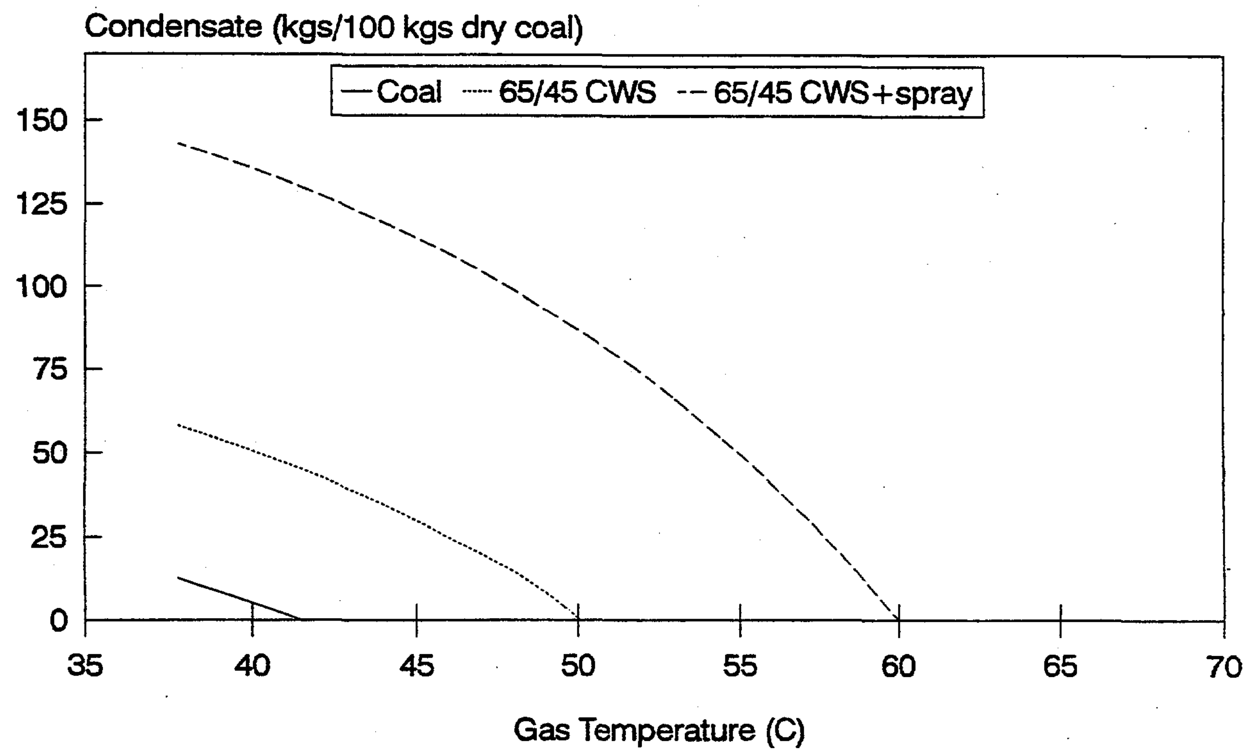

Figure 3-2 . Effect of flue gas discharge temperature on the ideal amount of water condensed from flue gas. Coal, coal water slurry, and coal water slurry with additional water added to flue gas.

FY94#2\gpetc\section.prs for section 4of petc report economizer surfaces would have a high probability of rapidly fouling. Condensation potential is increased when water sprays are added. With a high flue gas water content it is easier to achieve condensation rates high enough to rinse collected particles away.

The efficiencies and condensation rates in Figures 3-1 and 3-2 are based on the assumption that equilibrium conditions exist at every point in the gas path. In an actual condensing economizer it is much more likely that water vapor will begin to condense on the relatively cold heat exchanger surfaces before the average flue gas temperature has reached the water vapor dewpoint.

EXPECTED PERFORMANCE OF AIR- AND WATER-COOLED ECONOMIZERS WITHOUT ADDED WATER SPRAYS

The thermal performance of the two condensing economizers included in this project has been predicted using the methodology discussed in Appendix A. In both cases it was assumed that a slurry with 65% of a Pittsburgh seam coal was being fired with an excess air level of 30%. The assumed firing rate is 82.1 kW (280,000 Btu/hr).

Figure 3-3 shows the calculated performance for the air-cooled economizer. Included are the temperature profiles of the gas as it passes through the tube bank, the flue gas mole fraction of water vapor, and the temperature of the condensate film on the outer surface of the tubes. 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 4.34 kW (14,812 Btu/hr) giving an efficiency improvement of 5.8%. Nearly all of this is sensible heat transfer and the rate of moisture condensation is very small. Examination of the water fraction curve in Figure 3-3 shows that the surface falls below the dewpoint, and water begins condensing only at the very end of the heat exchanger, at around row 10.

The performance of the water-cooled economizer for this case is shown in Figure 3-4. Cooling water enters the economizer at 15.6 C (60 F) and exits at 38 C (100 F). The unit is arranged in counterflow. For all row numbers greater than 26 the surface is below the water vapor dewpoint and latent heat transfer contributes significantly to the total heat transfer rate. Overall, the heat transfer rate is 8.33 kW (25,000 Btu/hr) and the efficiency improvement provided by the economizer is 8.9%. Clearly this is doing a much better job of both cooling the gas and removing water vapor than the air-cooled case.

Appendix B to this report provides a description of the methods used to estimate capture of particulate flyash through the economizers. With the relatively large particle size associated with flyash from coal firing direct inertial impaction on the tube surfaces is expected to be the most important collection mechanism. Figure 3-5 shows a comparison of the predicted removal efficiency by inertial impaction alone as a function of particle diameter for the air- and water-cooled economizers. The conditions assumed for this comparison are again coal water slurry firing at a rate of 82.1 kW (280,000 Btu/hr) without additional water being added to the flue gas.

The capture of particulate is clearly better in the case of the water-cooled economizer. This is simply a result of the smaller "minimum free flow area" and higher gas velocity with this economizer.

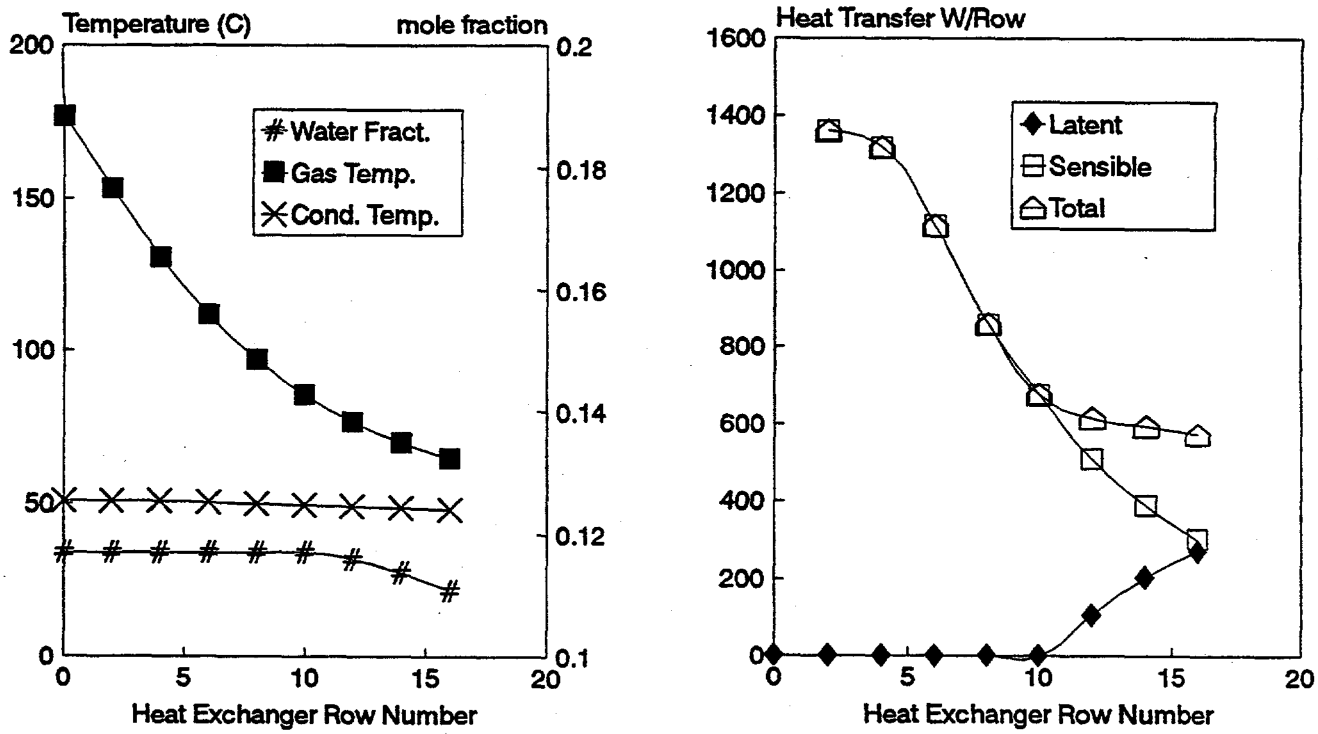

Figure 3-3. Calculated performance of air-cooled condensing economizer. Firing coal water slurry without use of water spray in flue gas.

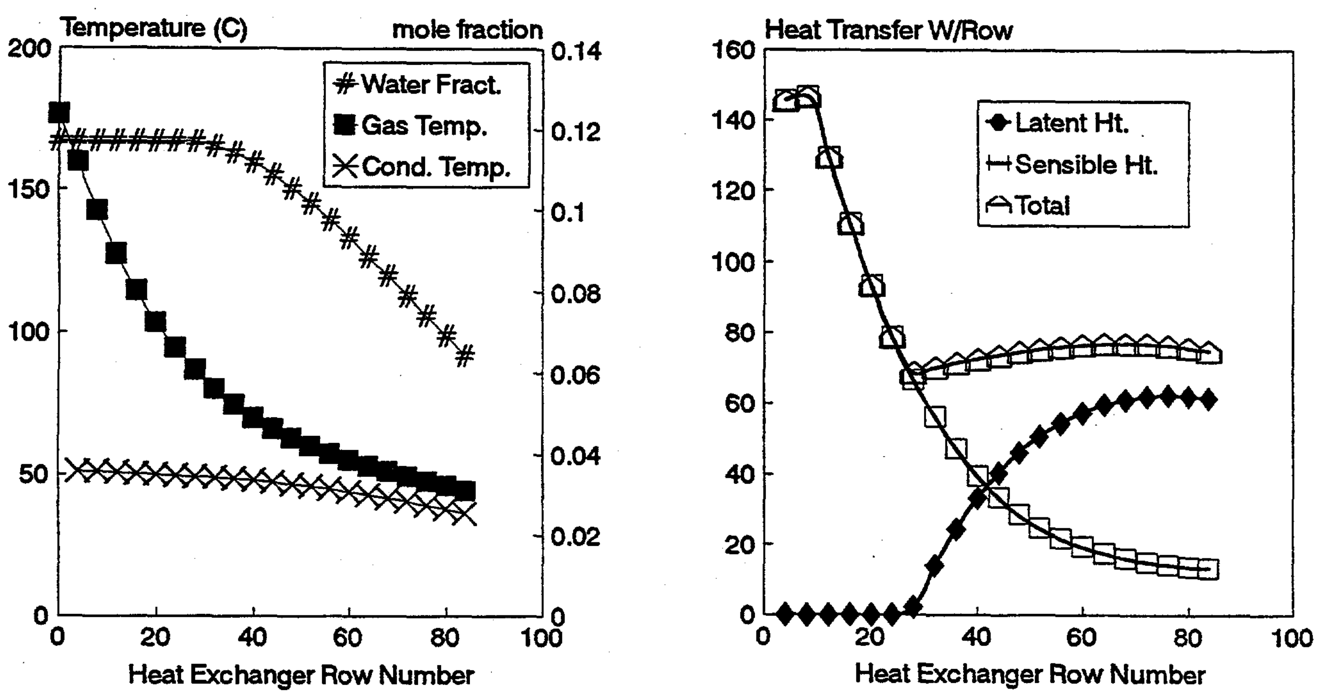

Figure 3-4. Calculated performance of water-cooled condensing economizer. Firing coal water slurry without use of water spray in flue gas.

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 smaller particles pass through. In the case of the air-cooled condenser the cut diameter is about 23 microns (Figure 3-5). Based on typical size distributions for pulverized coal flyash 50 to 70% of the ash is smaller than this size. Removal efficiency would only be about 30 to 50%. For the water-cooled condenser the cut diameter is about 10 microns. Based on this and typical flyash size distribution the removal efficiency would be expected to range from 60 to 80%.

In Appendix B the potential for removal of particulate by direct water condensation on the particles, leading to particle growth and downstream removal by inertial impaction is discussed. Generally, this can occur only if there is a probability that a supersaturated region can occur within the boundary layer. Figure 3-6 illustrates, for the water-cooled economizer case, the change in bulk flue gas temperature and water vapor content through the tube bank. Also shown is the predicted temperatures of the tube surface through the economizer and the equilibrium volume fraction of water vapor at that temperature. As discussed in Appendix B, a very simple approximation for the change in state of the flue gas through the boundary layer can be made by drawing a straight line from the bulk point to the surface point. If this line passes below the saturation curve, supersaturation will exist in the boundary layer. Based on Figure 3-6 the presence of such a supersaturated condition is definitely not expected.

EXPECTED PERFORMANCE OF AIR- AND WATER-COOLED ECONOMIZERS WITH ADDED WATER SPRAYS

Some of the mechanisms which affect particle removal (see Appendix B) can be enhanced by increasing water vapor condensation rate within the economizer. This would be accomplished by using water sprays in a pre-saturator section to increase flue gas water content.

Using water sprays in the presaturator section increases the flue gas water vapor content, decreases flue gas temperature and increases the flue gas water vapor content. Figure 3-7 illustrates these effects for the case of flue gas leaving the boiler at 332 C (600 F). In this case it is assumed that the fired fuel is coal with 10% moisture content and burner excess air is 30%. In this case the flue gas volume percent water vapor can be adjusted over the range from 9% (without any water sprayed) to a maximum of 26%. The inlet gas temperature assumed in this case is about the highest which might be found in a practical boiler system. Often the normal boiler exit temperature would be less and, as a result, the maximum increase in the flue gas water vapor content which could be achieved in a presaturator would be less. In some incinerator systems presaturation of flue gas upstream of a high energy wet scrubber is used as part of an overall emissions removal system. Very often in these cases, however, the gas temperature entering the presaturator is much higher, 980 C (1800 F), enabling much higher flue gas water vapor contents to be achieved in the presaturator.

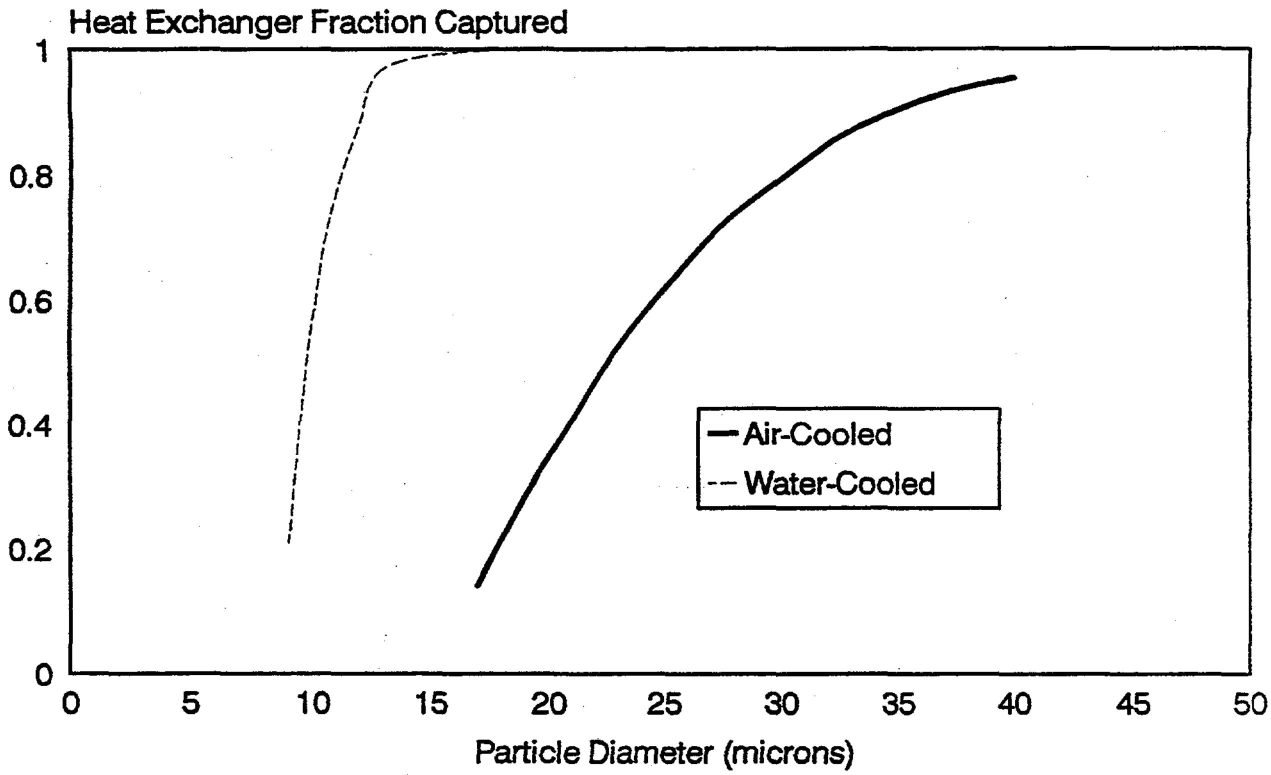

Figure 3-5. Predicted removal of particles by inertial impaction. Water- and air-cooled economizers with burner firing rate of 82.1 kW (280,000 Btu/hr)

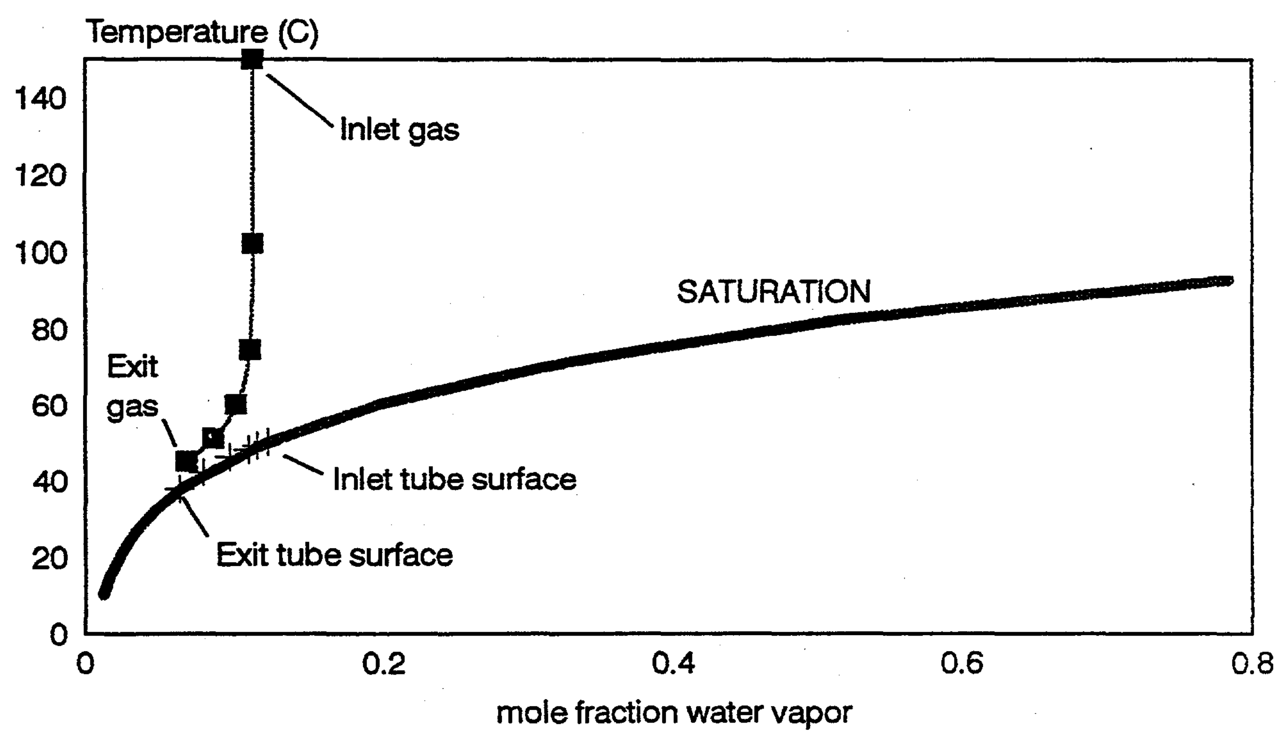

Figure 3-6. Illustration of the change in temperature and water vapor content through the economizer. Example results of calculations. "Surface" points represent surface temperature and equilibrium water vapor content at surface temperature.

In Figure 3-3 the results of an analysis of heat transfer in the air-cooled economizer without using water sprays was presented. Figure 3-8 again shows the results of analysis of heat transfer in the air cooled economizer but here it has been assumed that water was added in the presaturator section to increase the water vapor content to 16%. Flue gas temperature entering the economizer is much lower due to the water addition. Water vapor condensation occurs throughout the entire economizer and latent heat transfer is more important than sensible heat transfer overall. The total heat transfer rate is considerably less than for the case without water addition in the presaturator - 3.02 kW (10,300 Btu/hr) with the water addition giving an efficiency improvement of 3.6%.

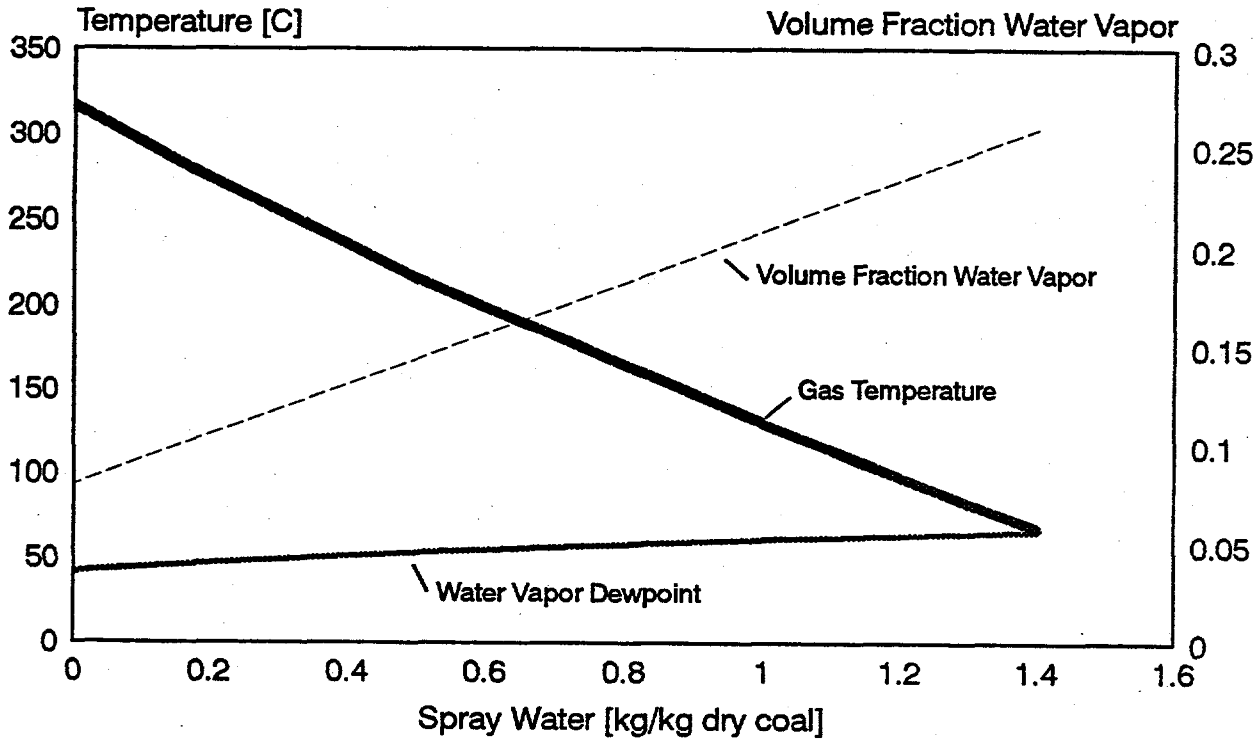

Figure 3-7. Effects of spraying water into flue gas upstream of economizer on flue gas temperature, water vapor content, and water vapor dewpoint temperature.

The case of the water-cooled economizer with water addition in the presaturator is shown in Figure 3-9. Again the addition of water to the flue gas increases the importance of latent heat transfer in the exchanger and water vapor is condensing throughout the economizer. As in the air-cooled case the addition of water reduced the total heat transfer rate but the effect is not nearly as great. Overall the heat transfer rate is 7.330 kW (21,000 Btu/hr) and the efficiency improvement due to the condensing economizer is 7.4%.

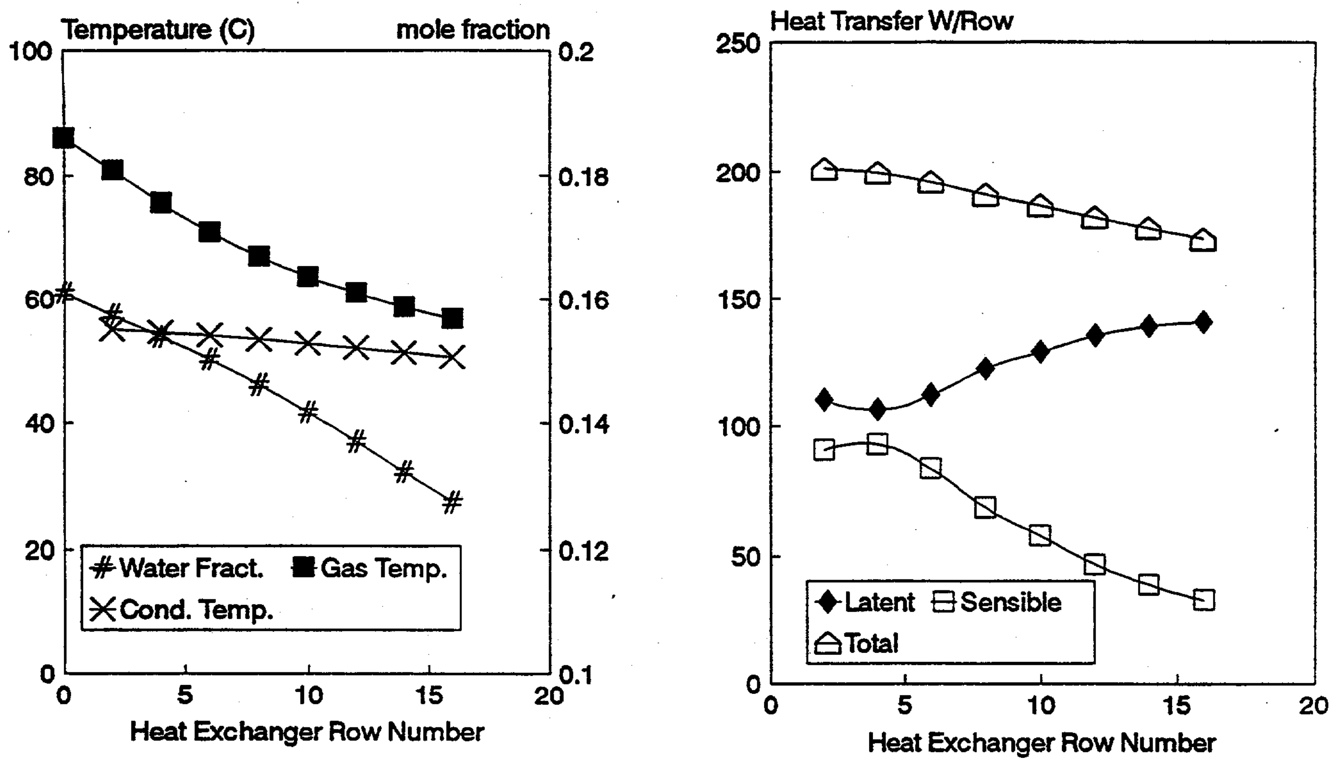

Figure 3-8. Calculated performance of air-cooled condensing economizer. Firing coal-water slurry with use of water spray in flue gas upstream of economizer.

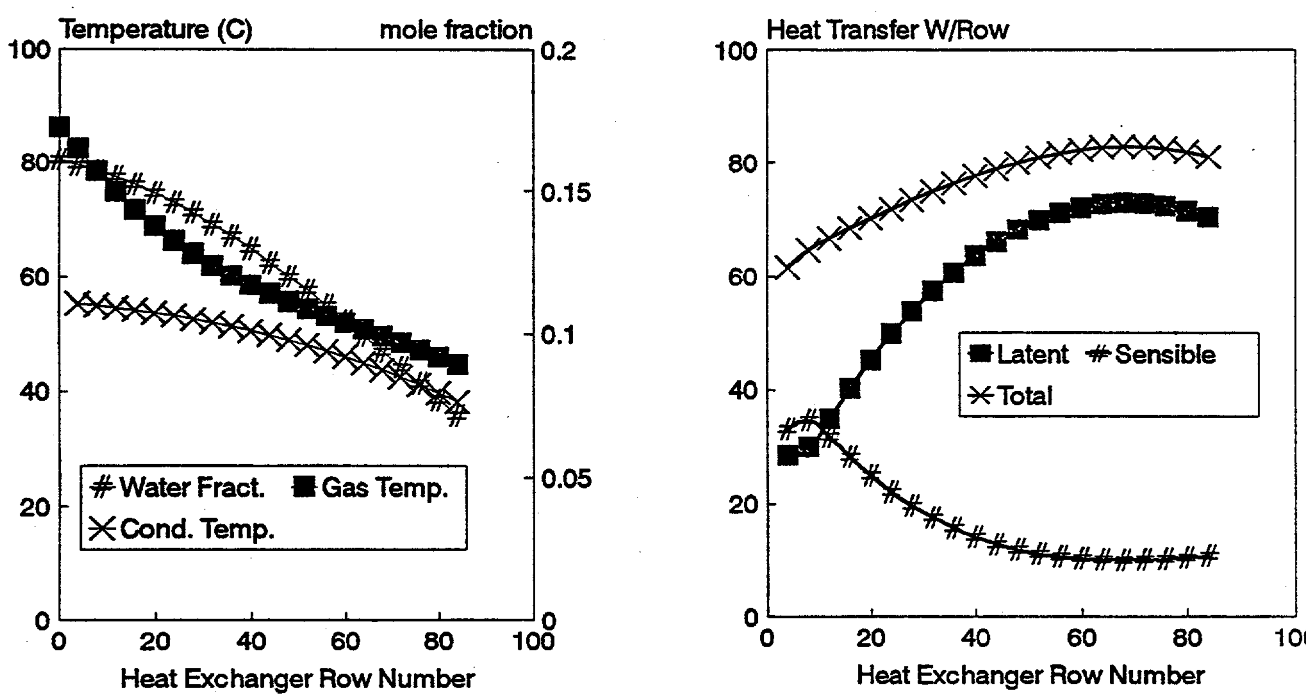

Figure 3-9. Calculated performance of water-cooled condensing economizer. Firing coal-water slurry with use of water spray in flue gas upstream of economizer.

The impact that the use of water sprays has on the total amount of heat recovered in a condensing economizer depends upon the temperature of the cooling water (or air) entering the economizer. To give an extreme example of this, if the flue gas is cooled to its dewpoint in the presaturator section but the cooling water enters at or above this temperature the economizer will be completely ineffective. In Figure 3-10 the effect is illustrated for a range of cooling water inlet temperatures, all of which are below the water dewpoint.

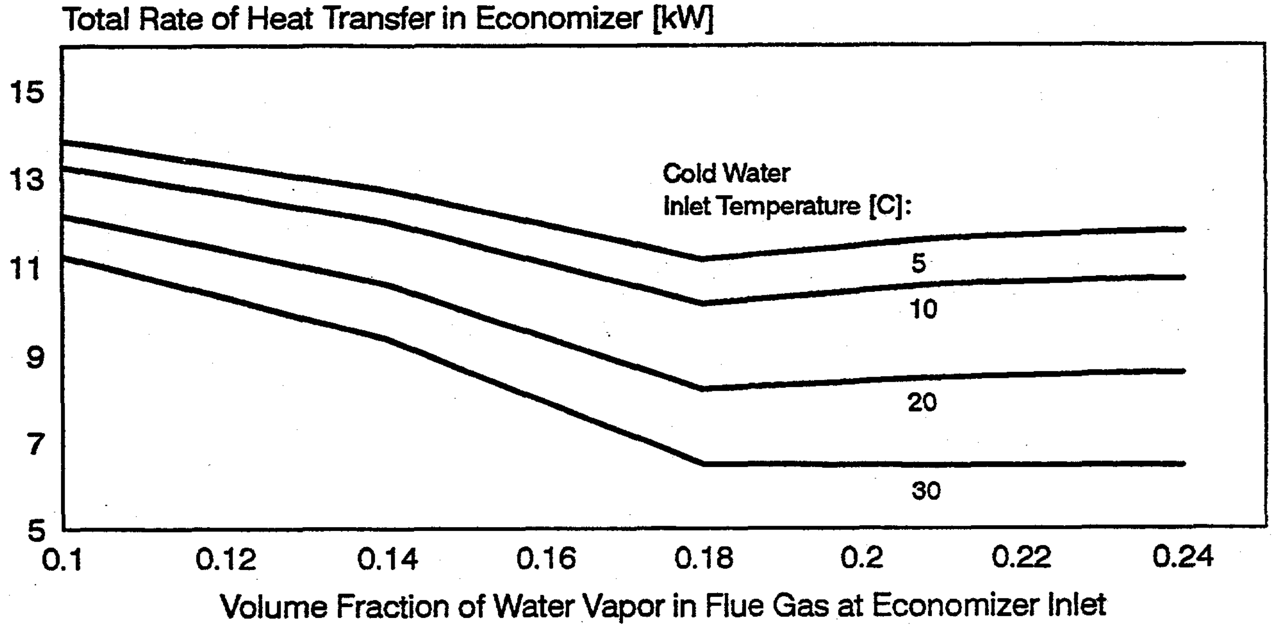

Figure 3-10. Effect of the use of water sprays upstream of the condensing economizer on the total heat transfer rate. Assumptions: cooling water flow rate adjusted to maintain 11 C [20 F] temperature rise across economizer, burner firing coal slurry at 82 kW [280,000 Btu/hr].

For all points in this figure the cooling water flow has been adjusted in the calculation procedure so that the water temperature rise across the economizer is 11 C (20 F). The variable flue gas water content would be achieved by varying the amount of water sprayed in the presaturator section upstream of the economizer. With increasing flue gas water content the temperature of the flue gas entering the economizer decreases. The relationship for the case shown in Figure 3-10 is as in Figure 3-7. As flue gas water content increases the sensible heat transfer decreases but latent heat transfer increases. These effects combine to give a minimum at about 18% flue gas water content. Adding more than this level of water leads to either no change in the predicted total heat transfer rate or a small increase. With a relatively low inlet water temperature (10 C) the reduction in heat transfer as the flue gas water content is increased from 10 to 18% is 9.7%. With a high inlet water temperature, 30 C, the reduction in heat transfer as the flue gas water content is increased from 10 to 18% is much greater - 42%. The profiles of flue gas bulk temperature and mole fraction water vapor and also tube surface temperature and equilibrium mole fraction water vapor at the tube surface temperature are shown in Figures 3-11 and 3-12 for two different levels of flue gas saturation. In Figure 3-11 the flue gas enters the economizer with 21% water vapor and a temperature of 130 C (266 F). For the case illustrated in Figure 3-12 the flue gas enters the economizer fully saturated. The inlet water vapor content is 26% and the flue gas inlet temperature is 68 C (154 F).

The case illustrated in Figure 3-11 is similar to the case examined earlier in which no additional water is added to the flue gas in the pre-saturator section (see Figure 3-6). In both cases it would seem unlikely that the flue gas would become supersaturated in the boundary layer.

For the case illustrated in Figure 3-12, where the gas enters fully saturated there is only a small temperature difference between the bulk flue gas and the tube surface. Essentially all of the heat transfer is by water condensation (latent heat transfer). In this case the gas is essentially saturated or possibly slightly supersaturated everywhere.

EFFECT OF LOAD ON PERFORMANCE

As boiler load changes, the flue gas mass flow through the condensing economizer changes affecting the temperature profile, water condensation rates, load heat and mass transfer coefficients, pressure drop and the gas velocity. Here predicted performance of the water-cooled condensing economizer is presented over a range of boiler heat input rates, using the simplifying assumption that the excess air and gas temperature entering the economizer are constant.

Figure 3-13 shows the economizer pressure drop over the range of burner heat inputs examined. It is really the pressure drop that establishes the highest practical heat input rate which could be considered for the economizer application. Here the assumption has been made that at the maximum rated heat input, the economizer pressure drop would be about 2000 Pa [8.0 in of water]. Higher pressure drops and higher firing rates could certainly be used; it is simply a question of fan power. The lowest firing rate considered here represents a 4:1 turn-down relative to the maximum rated firing rate.

Figure 3-14 shows effects of load on amount of heat recovered in the economizer and the improvement in system efficiency due to the economizer heat recovery. As load increases, the heat and mass transfer coefficients increase, average economizer gas temperature increases, average flue gas water vapor content in the economizer increases, and the heat transfer rate increases. The efficiency improvement, which is simply heat recovered divided by burner firing rate, decreases with increasing load. Figure 3-15 shows the economizer outlet flue gas temperature and water vapor content as load varies. At the very lowest load, the economizer is clearly removing practically all of the recoverable heat and water vapor from the flue gas.

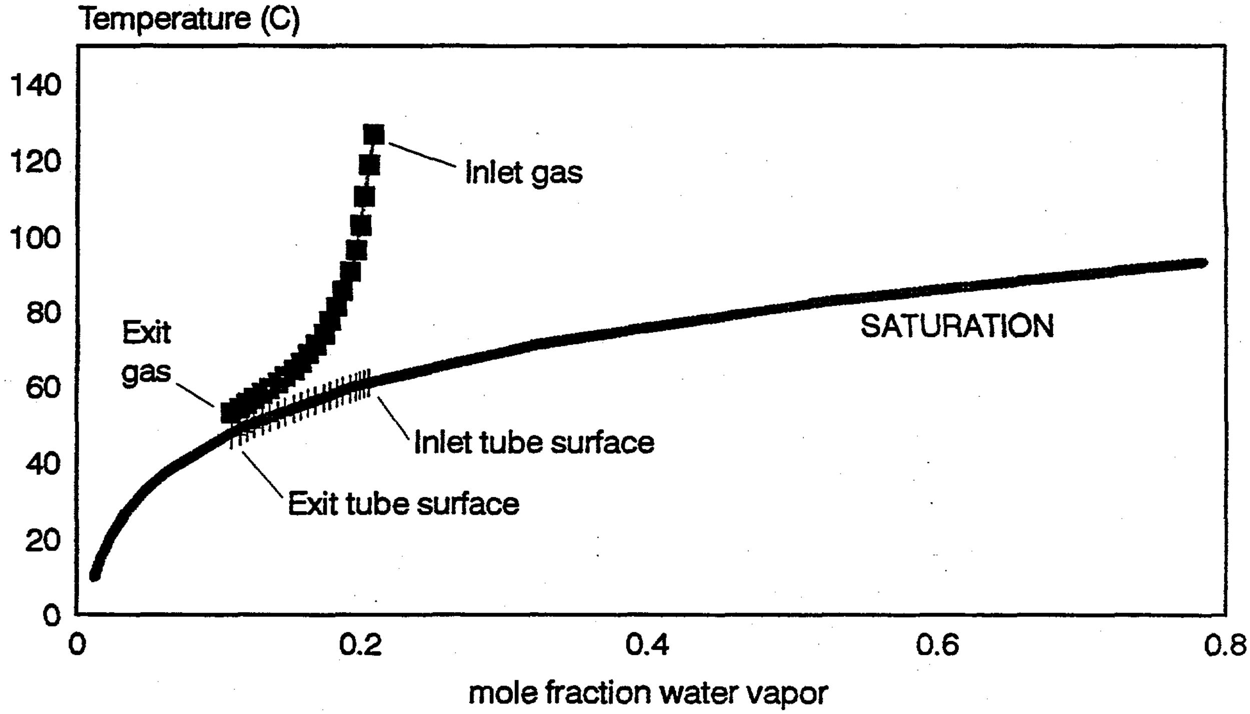

Figure 3-11. Illustration of change in temperature and water vapor content through the economizer with spray in use. Inlet gas temperature - 130 C (266 F). Inlet gas water vapor fraction - .21.

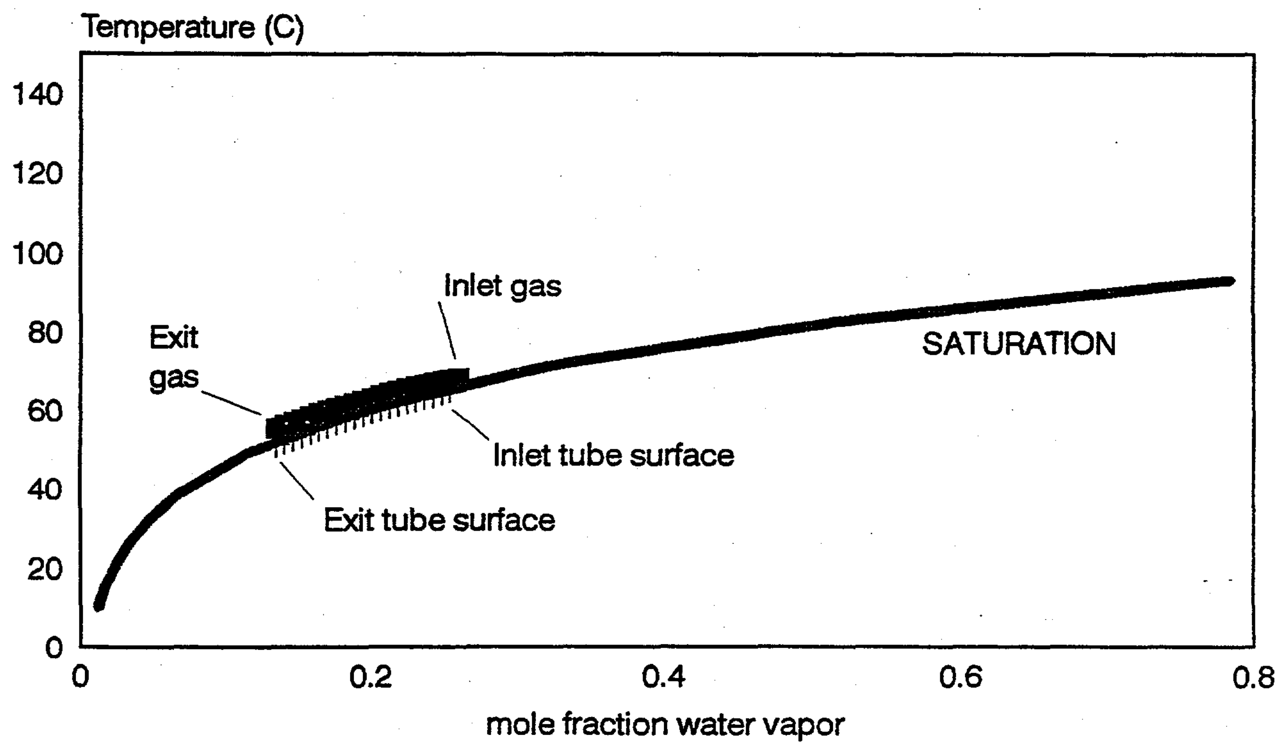

Figure 3-12. Illustration of change in temperature and water vapor content through the economizer with spray in use. Inlet gas temperature - 68 C (154 F). Inlet gas water vapor fraction - .26.

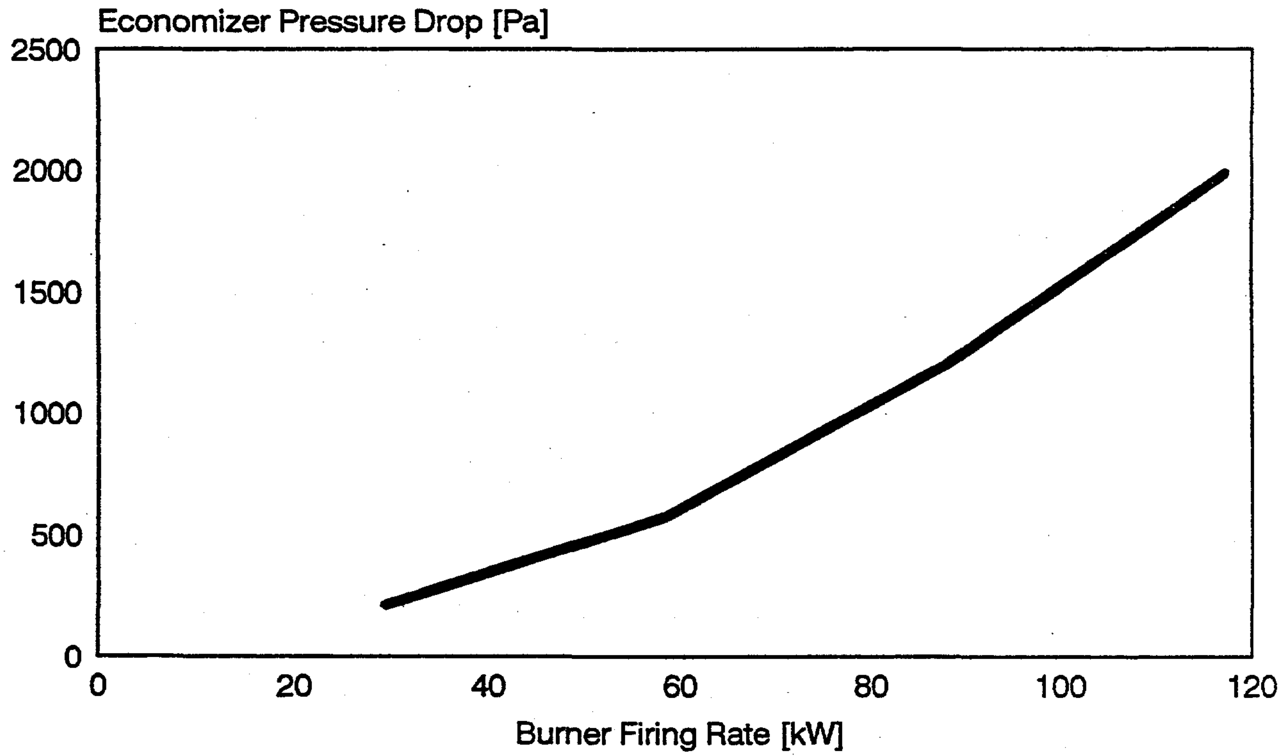

Figure 3-13. Predicted pressure drop as a function of boiler firing rate. Water-cooled condensing economizer.

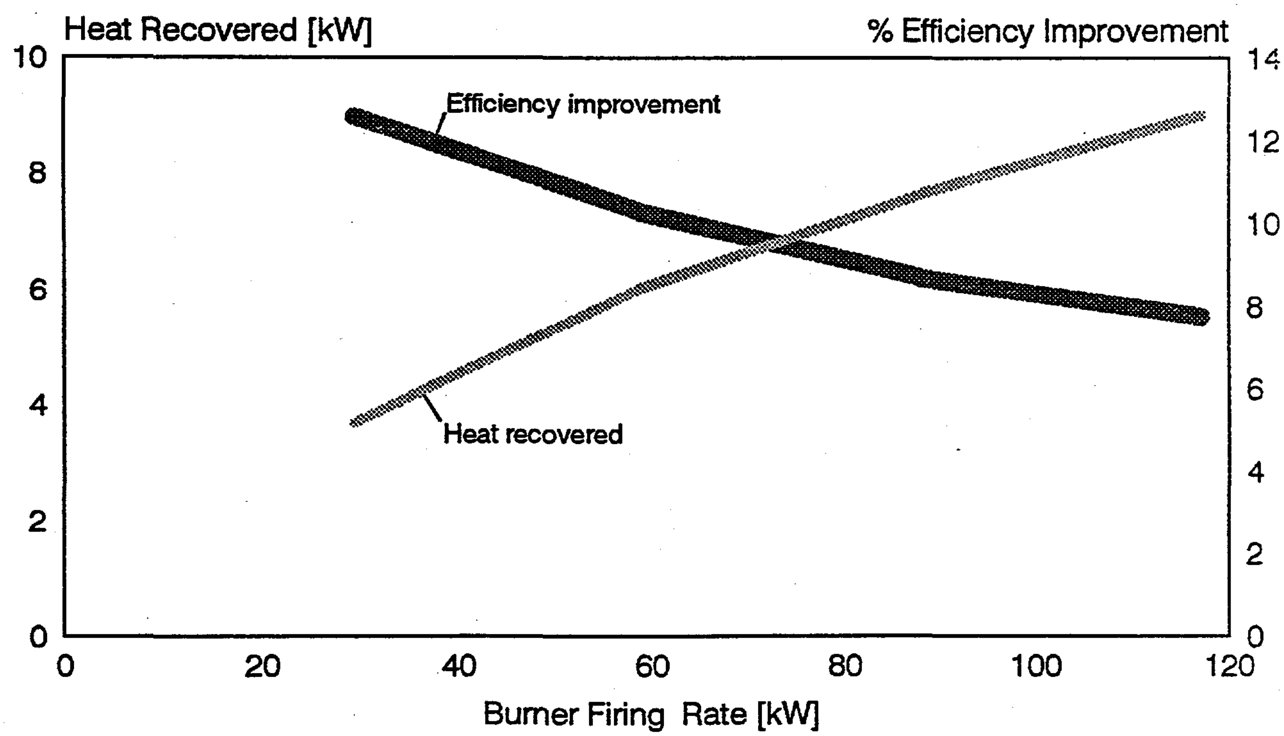

Figure 3-14. Economizer heat recovery and system efficiency improvement due to economizer heat recovery. Load Effect. Water-cooled condensing economizer.

In Appendix B, where the methodology for predicting particle capture is described, inertial impaction of flyash on the tubes is shown to be the most significant removal mechanism. Impaction, which is strongly dependent upon gas velocity, would certainly be affected by load. Figure 3-16 shows the predicted particle removal/particle diameter curves for 5 different loads. As discussed above, the highest pressure drop assumed for full load operation of this economizer is 2000 Pa (8 in of water). In Figure 3-16, one additionally higher pressure drop assumption has been added simply to illustrate the results which might be attained if a higher maximum pressure drop were accepted. Figure 3-16 clearly shows that inertial particle removal across the economizer would be worse at low loads than at high loads.

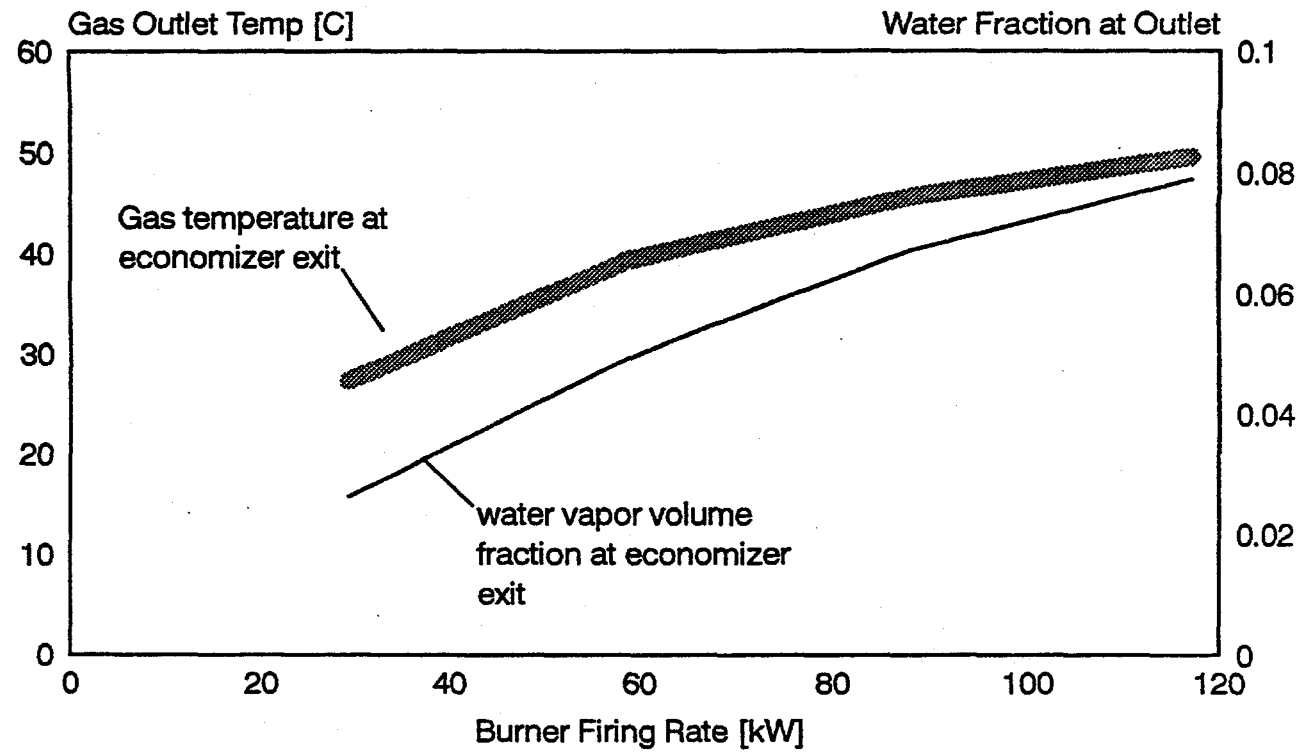

Figure 3-15. Economizer outlet gas temperature and water fraction as a function of boiler firing rate.

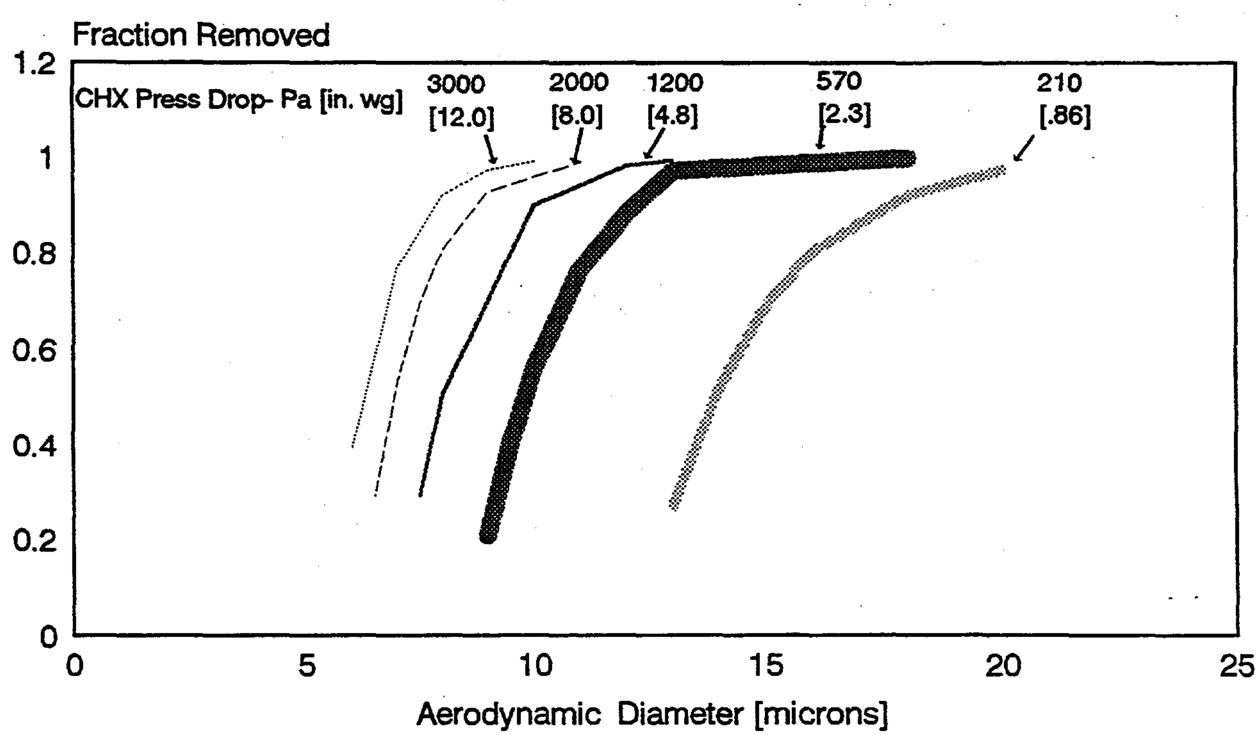

Figure 3-16. Effect of load on predicted particle removal by impaction / particle size curves for water-cooled condensing economizer.

SECTION SECTION 4. SUMMARY OF EXPERIMENTAL RESULTS

As discussed in Section 2, the data was generated in two ways, with oil-firing and with coal slurry firing. In the oil-fired case the performance of both air- and water-cooled heat exchangers were evaluated. Only the water-cooled economizer was used in the experiments with the coal slurry firing because of its greater applications.

AIR-COOLED ECONOMIZER; OIL-FIRING WITH SPRAY-DRIED FLYASH

The results of the tests conducted with the air-cooled heat exchanger are shown in Table 4-1 at firing rates of 62 kW (212,000 Btu/hr) and 82 kW (280,000 Btu/hr). All of the tests with the air-cooled economizer were done with oil-firing and spray-dried flyash as described in Section 2. For some of these tests additional water was sprayed directly into the combustion chamber flame zone to increase flue gas water content at the economizer inlet. These include "runs" 1,5,6, and 7. When additional water is sprayed into the combustion chamber the temperature of the flue gas entering the economizer decreases. The relationship between the amount of water added in this way and the economizer inlet gas temperature is not, however, generally in agreement with the expected relationship with water added in a presaturator section just upstream of the economizer (see Figure 3-7). This is because spraying the water directly in the combustion zone changes gas flow rates and heat transfer patterns within the boiler. In this experiment the water was added directly in the combustion chamber rather than in a presaturator section between the boiler and the condensing economizer to assure complete evaporation. The particulate concentrations at the inlet and outlet of the condensing economizer were measured and the overall particulate removal efficiencies across the economizer was determined to vary from 42% to 66%. Up to 5% by volume of the moisture in the flue gas was condensed. Based on this data there is little correlation between particulate removal efficiency and economizer inlet water vapor content. With the method developed under this study to predict thermal performance the calculated flue gas temperatures and the moisture content at the outlet could be compared with the measurements. In most cases, they were consistent.

WATER-COOLED ECONOMIZER; OIL-FIRING WITH SPRAY-DRIED FLYASH

Initial tests with the water-cooled condensing economizer were all done with oil-firing, spray-dried flyash and burner firing rate fixed at 62 kW (212,000 Btu/hr). The only parameter changed in this series of tests was the amount of water added to the combustion products using water sprays within the combustion chamber. Results are listed in Table 4-2. For all tests, total pressure drop across the economizer was 600 Pa (2.4 in. of water). At a firing rate of 62 kW (212,000 Btu/hr) the flue gas oxygen content was typically 8%. The particulate loading at the inlet of the economizer varied over a wide range of concentrations because different spray methods and conditions for the ash stream were being evaluated as part of these tests. For runs 1 through 5, simultaneous measurements of particulate concentration were made upstream and downstream of the economizer using two EPA-5 Trains. For runs 6 through 10, the cascade impactor was used at the economizer outlet providing both size distribution and total dust loading (by summing stages). During runs 6 through 10, the EPA-5 train was still used at the economizer inlet to provide a simultaneous inlet loading measurement. The different amounts of water sprayed into the combustion chamber for the tests listed in Table 4-2 give a wide range for the water vapor content at the economizer inlet. A comparison of the inlet water vapor contents and the particulate removal efficiencies again shows no consistent relationship. Using the water sprays to greatly increase inlet flue gas, water vapor content did not provide an increase in particle removal.

Using the measured particle outlet size distribution at various levels of flue gas water vapor content at the economizer inlet and separate measurements of particle size distribution at the inlet and outlet, it is possible to construct removal efficiency/particle size curves. Typical particle size distribution measured at the economizer inlet is shown in Figure 4-1. Figure 4-2 shows the particulate removal as a function of particle size. The variable parameter used in this figure is the part of the flue gas condensed; for example when the highest level of water vapor was added to the flue gas, the inlet water vapor content was 20%, and the outlet was 9%, giving 11% for the part of the flue gas condensed. As a general remark, the economizer was found to be quite effective in removing particles over 5 microns and the effects of water condensation rate were found to be small.

Table 4-3 lists the results of an additional series of tests in which water was sprayed directly on the first row of tubes on top of the condensing economizer. These tests were again done at a burner firing-rate of 62 kW (212,000 Btu/hr) and flyash was spray-dried in the back of the combustion chamber. The water spray rate at the economizer inlet was 13.2 liters/hr (3.5 gal/hr), which is more than adequate to saturate the flue gas. For these tests an additional location was added for particulate measurements - downstream of the induced draft fan. The measurements were made at the fan discharge with the expectation that the fan might provide some additional inertial removal of wet particulate at the economizer discharge. This removed particulate would drain out from the fan housing and duct walls along with the condensate. Results listed in Table 4-3 and a comparison with Table 4-2 show that removal efficiency across the economizer is improved with the addition of the water sprays.

The induced draft fan downstream of the economizer is effective in removing about 50% of the remaining particulates and boasts total system removal to 97-98%. Some additional tests were also done in which water sprays were located at the inlet to the fan, essentially spraying at the impeller eye. The rationale for these tests was that the water sprays may add impaction sites and improve overall particulate capture. These limited tests were not successful. The added sprays created problems in handling the runoff water really did not improve particle capture possibly because of re-entrainment caused by the spray.

Table 4-4 lists the results which were done at higher firing rates - 82 kW (280,000 Btu/hr), again with oil-firing and spray-dried flyash. The higher firing rate gives higher economizer pressure drop - 1150 Pa (4.6 in. of water) requiring a change to a higher static pressure induced draft fan. Particulate removal of 96% to 98% across the economizer and over 98% considering the additional removal across the induced draft fan were acheived. In these tests, no additional water sprays were used either in the combustion chamber or within the economizer. Generally the removal efficiency at the higher load without water sprays directly on the economizer tubes is similar to results at lower loads with water sprays (Table 4-3). This is significantly higher than that of prior results on Tables 4-2 and 4-3 at only 600 Pa (2.4 in of water) pressure drop without water sprays.

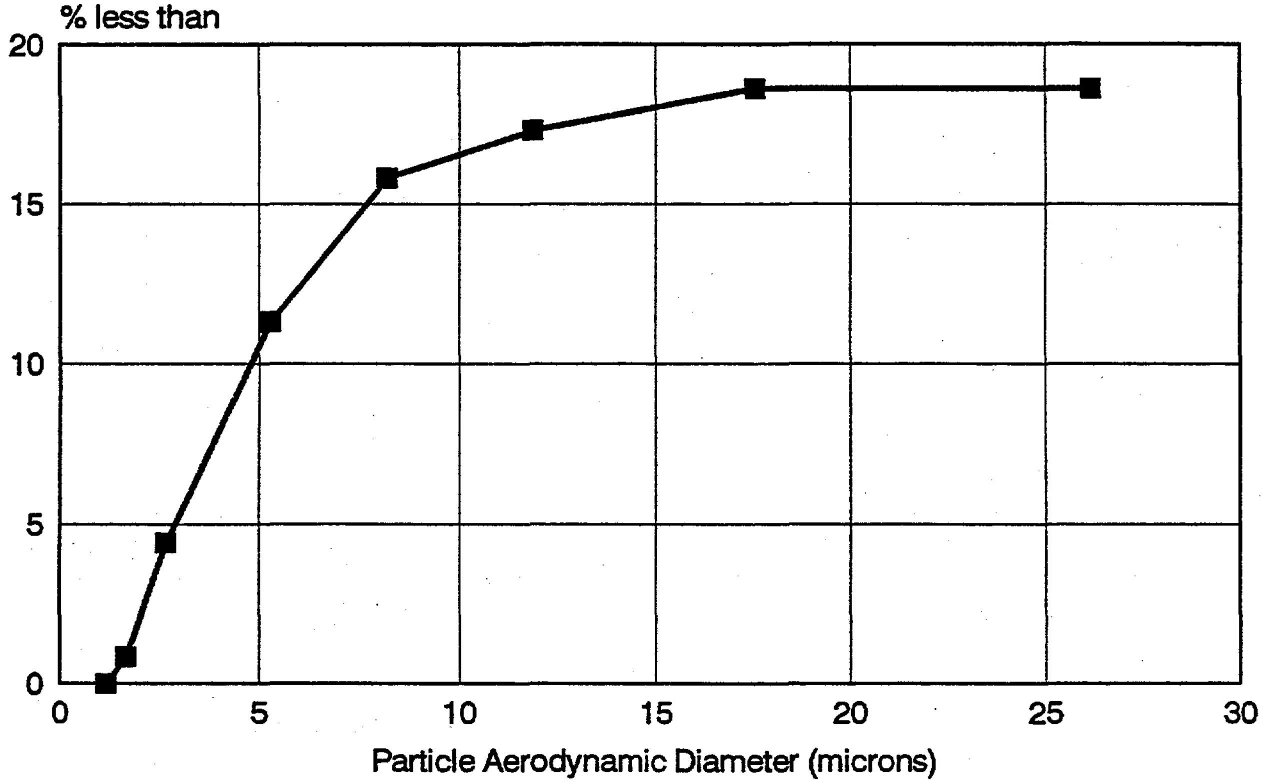

Figure 4-1. Typical size distribution at the economizer inlet. Spray-dried flyash.

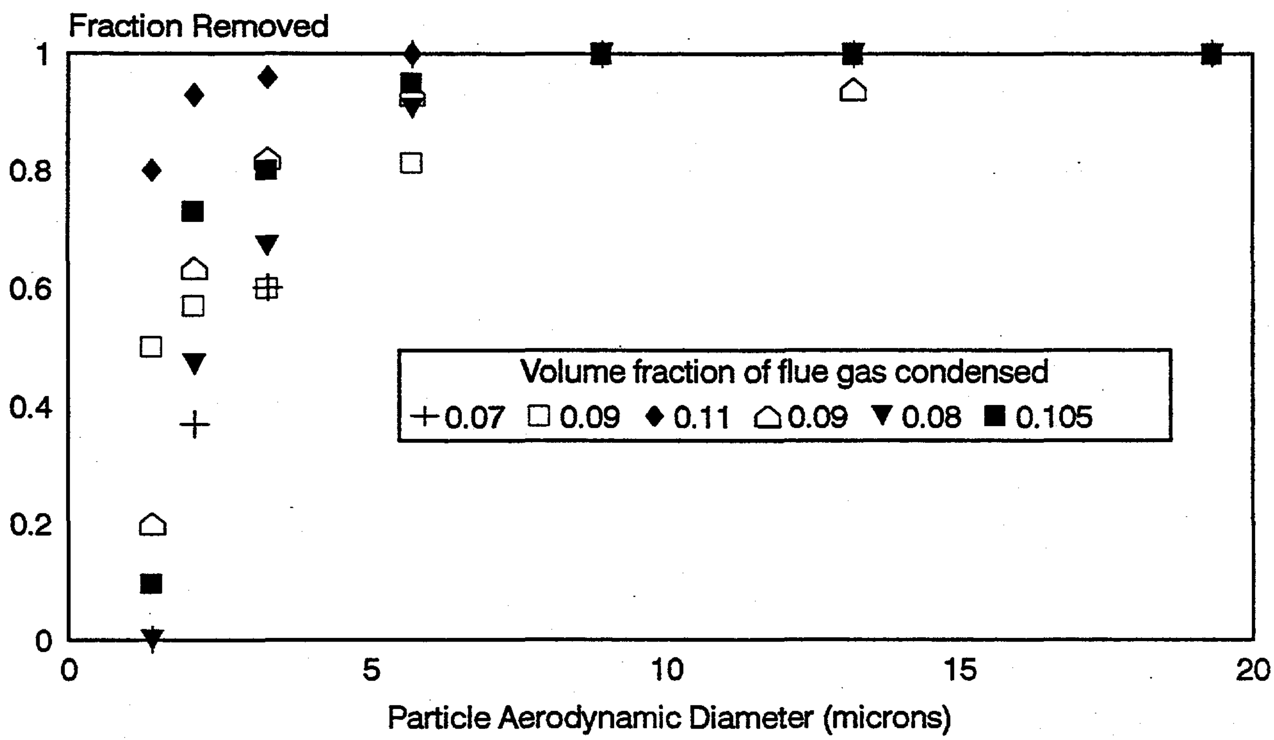

Figure 4-2. Particulate flyash removal efficiency as a function of particle size. Measured results with water-cooled condensing economizer with spray-dried flyash.

COAL SLURRY FIRING

In most of these tests coal-slurry was burned with the support of the independent oil-firing system integrated with the slurry burner. The firing rate of oil was maintained at 30 kW while the slurry rate was adjustable. For the first three runs the pressure drop across the economizer was kept at about 1150 Pa (4.6 in. of water), and other operating conditions such as water sprays to the top of the heat exchanger, and slurry feedrate were adjusted for evaluation. Some of the data obtained are shown in Table 4-5. In run 2 the use of water sprays directly on the economizer tubes significantly cooled the flue gas at the inlet of the economizer. Although it improved particle collection, it reduced the heat transfer performance of the economizer. In one of the tests the oil-firing system was turned off completely so that only the slurry was used. In this case, hot combustion air was provided to the burner head with the integral kerosene fired heater (see Section 2, Figure 2-2). In run 4 the firing rate of the slurry was calculated based on the weight of slurry used over a period of time. Although the feed rate was set in advance to a lower value than in other tests, this calculated value may be underestimated due to possible plugging of the nozzle at the end of the sampling run.

The removal efficiency measured with or without oil-cofiring ranged from 85% to 92%. Again, with water sprays to the top of the condensing heat exchanger marked improvements were observed with removal of 94%. For these tests the water vapor content could not be measured accurately using the EPA-5 train impingers due to heavy particulate loading in the flue gas and the necessity for shorter sampling times as a result. The particle size distribution for the particulates in the flue gas during coal slurry firing is shown in Figure 4-3. Measurements made during two separate runs are presented. Comparison with the size distribution of the flyash spray-dried into the flue gas (see Figure 4-1) shows somewhat larger particles in the case of the coal slurry firing.

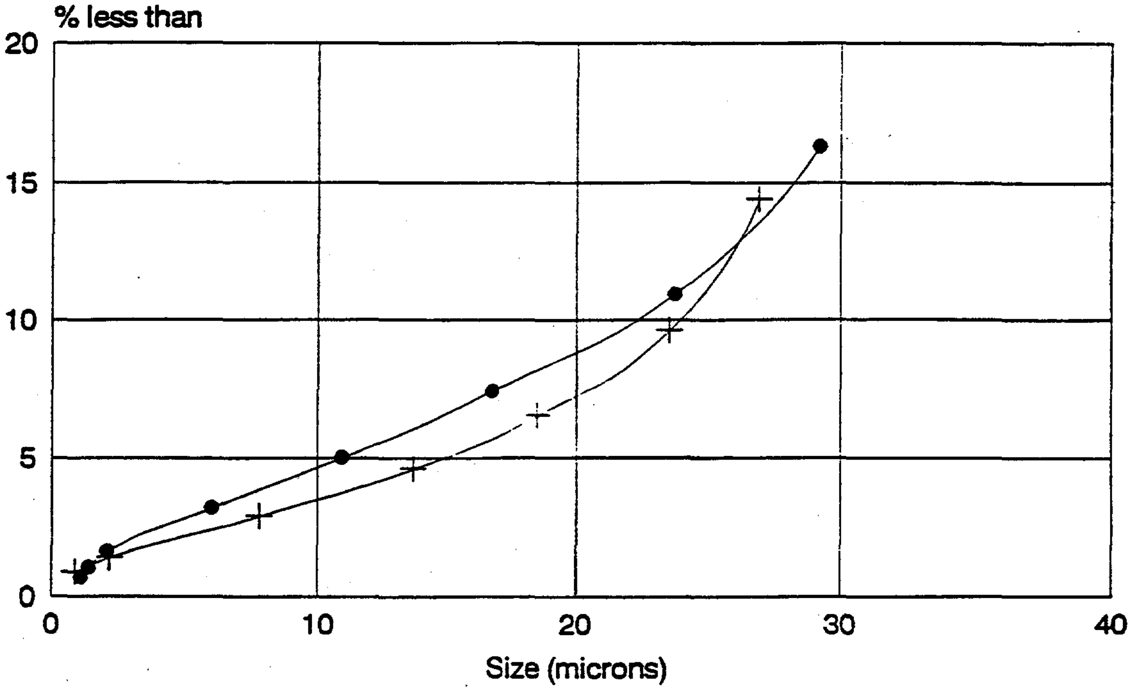

Figure 4-3. Typical flyash size distribution at the economizer inlet with coal water slurry firing.

Run	1	2	3	4	5	6	7	8	9
Firing rate, kW	82	82	82	62	62	62	62	62	62
(Btu/hr)	*(280,046)*	*(280,046)*	*(280,046)*	*(211,742)*	*(211,742)*	*(211,742)*	*(211,742)*	*(211,742)*	*(211,742)*
Ash in slurry, wt %	10	10	10	10	10	10	10	10	10
Slurry spray rate, cm3/s	1	1	1	1.08	1.05	1.05	1.05	1.37	1.26
Particulate concentration at the inlet of economizer, gm/dscm	0.706	0.847	0.777	0.742	0.6	0.494	0.512	1.45	1.59
Particulate concentration at the outlet of economizer, gm/dscm	0.388	0.494	0.332	0.247	0.212	0.212	0.282	0.586	0.561
Particulate removal efficiency across the economizer, %	45	42	57	66	63	55	45	60	65
Measured water vapor content at the inlet of economizer, %	24	15	14	ND	22	22	20	14	13
Measure water vapor content at the outlet of economizer, %	22	10	11	ND	20	17	15	12	12
Predicted water vapor content at the outlet of economizer, %	20.4	14.3	13	ND	18.9	16.5	14.7	12.1	
Measured flue gas temperature at the inlet of economizer, C (F)	177 (351)	191 (376)	193 (379)	127 (261)	143 (289)	81 (178)	81 (178)	127 (261)	133 (271)
Measured flue gas temperature at the outlet of economizer, C (F)	74 (165)	71 (160)	74 (165)	54 (129)	66 (151)	56 (133)	52 (126)	53 (127)	71 (160)
Predicted flue gas temperature at the outlet of economizer, C (F)	77 (171)	71 (160)	69 (156)	ND	71 (160)	61 (142)	58 (136)	60 (140)	
ND - No Data									

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Table 4-1. Summary of Results of Tests with Air-cooled Condensing Economizer

						Data for Runs 6-10 includes particle size distribution at the outlet of the condensing economizer				
Run	1	2	3	4	5	6	7	8	9	10
Firing rate, kW	62	62	62	62	62	62	62	62	62	62
(Btu/hr)	*(211,742)*	*(211,742)*	*(211,742)*	*(211,742)*	*(211,742)*	*(211,742)*	*(211,742)*	*(211,742)*	*(211,742)*	*(211,742)*
Particulate concentration at the inlet of economizer, gm/dscm	0.625	0.516	0.176	0.607	0.171	0.259	0.335	0.376	0.171	0.186
Particulate concentration at the oulet of economizer, gm/dscm	0.066	0.058	0.0413	0.061	0.014	0.019	0.0294	0.04	0.014	0.032
Particulate removal efficiency across the economizer, %	89	89	76	90	92	92.7	91	89	92	83
Measured water vapor content at the inlet of economizer, %	11	19	11	12	15	10	14	15	15	20
Measured water vapor content at the outlet of economizer, %	5.9	7	7	6	7	3	5	6	7	9
Predicted water vapor content at the outlet of economizer, %	5.1	7.6	5.1	5.5	6.9	4.5	6.4	6.8	6.9	9.8
Measured flue gas temperature at the inlet of economizer, C (F)	146 (295)	97 (207)	162 (324)	166 (331)	174 (345)	168 (334)	170 (338)	169 (336)	174 (345)	172 (342)
Measured flue gas temperature at the outlet of economizer, C (F)	45 (113)	44 (111)	43 (109)	44 (111)	46 (115)	44 (111)	45 (113)	47 (117)	46 (115)	50 (122)
Predicted flue gas temperature at the outlet of economizer, C (F)	39 (102)	46 (115)	39 (102)	41 (105)	45 (113)	38 (100)	43 (110)	44 (112)	45 (113)	51 (124)

Table 4-2. Summary of Results of Tests with Water-cooled Condensing Economizer. All Tests with Oil-firing, Spray-dried Flyash, Firing Rate of 62 kW (212,000 Btu/hr), Additional Water Sprays in Combustion Chamber.

Run	11	12
Firing rate, kW	62	62
(Btu/hr)	(211,742)	(211,742)
Particulate concentration at the inlet of economizer, gm/dscm	0.376	0.206
Particulate concentration at the outlet of condensing economizer, gm/dscm	0.0167	0.013
Particulate removal efficiency across economizer, %	95	93
Particulate concentration at the outlet of induced draft fan, gm/dscm	0.0056	0.006
Overall particulate removal efficiency across system	98	97
Measured water vapor content at inlet of economizer, %	9	7
Measured water vapor content at outlet of economizer, %	5	5
Measured flue gas temperature at the outlet of economizer, C (F)	39 (102)	36 (97)

Table 4-3. Summary of Results of Tests with Water-cooled Condensing Economizer. All Tests with Oil-firing, Spray-dried Flyash, Firing Rate of 62 kW (212,000 Btu/hr), Water Sprays Directly on First Row of Economizer Tubes.

Run	13	14
Firing rate, kW	82	82
(Btu/hr)	(280,046)	(280,046)
Particulate concentration at the inlet of economizer, gm/dscm	0.544	0.378
Particulate concentration at the outlet of economizer, gm/dscm	0.01	0.014
Particulate removal efficiency across economizer, %	98	96
Particulate concentration at the outlet of induced draft fan, gm/dscm	0.007	0.006
Overall particulate removal efficiency across system, %	98.6	98
Measure water vapor content at the inlet of economizer, %	11	10
Measure water vapor content at the outlet of economizer, %	5	5
Measured flue gas temperature at the inlet of economizer, C (F)	205 (401)	205 (401)
Measured flue gas temperature at the outlet of economizer, C (F)	51 (124)	52 (126)

Table 4-4. Summary of Results of Tests with Water-cooled Condensing Economizer. All Tests with Oil-firing, Spray-dried Flyash, Firing Rate of 82 kW (280,000 Btu/hr), No Additional Water Sprays Used.

		Water sprays at top of economizer	No oil-cofiring	
Run	1	2	3	4
Firing rate of coal slurry, kW	66	66	72	24
(Btu/hr)	*(225,403)*	*(225,403)*	*(245,894)*	*(81,965)*
Particulate concentration at the inlet of economizer, gm/dscm	5.08	3.34	9.23	1.89
Particulate concentration at the outlet of economizer, gm/dscm	0.777	0.184	0.72	0.205
Particulate removal efficiency across economizer, %	85	94	92	89
Measured flue gas temperature at the inlet of economizer, C (F)	203 (397)	63 (145)	235 (455)	249 (480)
Measured flue gas temperature at the outlet of economizer, C (F)	68 (154)	50 (122)	69 (156)	65 (149)
Flue gas oxygen content, %O2	10.2	ND	3.9	5.9

Table 4-5. Summary of Results of Tests with Water-cooled Condensing Economizer. All Tests with Coal-Water Slurry Firing, in Some Cases with Oil-cofiring.

SECTION 5. DISCUSSION

Results of the experimental work generally show that inertial capture of particulate is the most important removal mechanism at work for the size particulate examined here. Specifically, this is supported by: 1) the shape of the removal fraction/particle size curves (Figure 4-2), 2) the observed higher removal in the higher pressure drop, higher gas velocity water-cooled economizer relative to the air-cooled economizer, and 3) increased particle capture with increasing burner firing-rate and economizer pressure drop. The relative importance of inertial mechanisms is certainly consistant with the predictions presented in Appendix B and Section 3. However, the actual particulate removal performance of the economizers has been found to be better than predicted. Several reasons are possible for this including: errors in the prediction caused by extending isolated cylinder impaction results to the tube bank arrangement, radiative cooling of particulates leading to local condensation growth, and impaction of particulates on condensate drops falling from economizer tubes or introduced by the direct water sprays. It is this final possible mechanism which the authors find most interesting and which is developed further in this section.

To estimate particle inertial capture on drops of condensate, the drop size, drop average local number density, and relative velocity between the gas and the condensate drop must be known in addition to local gas properties. Most of this is not known and here some very rough approximations are made simply to examine the possibility that this mechanism might be important. For these calculations, it has been assumed that all drops are 5 mm in diameter, that 6% of the heat exchanger minimum flow area is effectively blocked by the drops and the drops are essentially stationary relative to the gas velocity. This last assumption implies the drops have recently fallen from a tube. Only the water-cooled economizer has been considered. The assumption of a 6% blockage fraction is equivalent to one 5mm drop near each tube in the water-cooled condensing economizer tested here. For these calculations, single drop collection efficiencies are related to local Stokes number using a relation due to Calvert [9]. Local gas properties are based on our routine for predicting heat transfer conditions within the economizer (see Appendix A).

Figure 5-1 provides a comparison of the predicted capture on the drops, the predicted capture by inertial impaction on the heat exchanger tubes and the increased removal efficiencies. All of these calculations were done assuming a burner firing rate of 62 kW (212,000 Btu/hr). These results show that, at least under the admittedly rough approximations made here, inertial capture on the drops can be a very significant factor in overall particle removal in condensing economizers. The high gas velocities through the condensing economizer can provide reasonable single drop capture efficiencies. (For example, the single drop collection efficiencies for 5 micron particles is 7.8%.) The removal efficiency across each row is only 6% (the assumed blockage fraction) of the single drop collection efficiency, giving very low per-row removal. When this removal, however, is taken cumulatively over 80 rows significant overall economizer removal is realized.

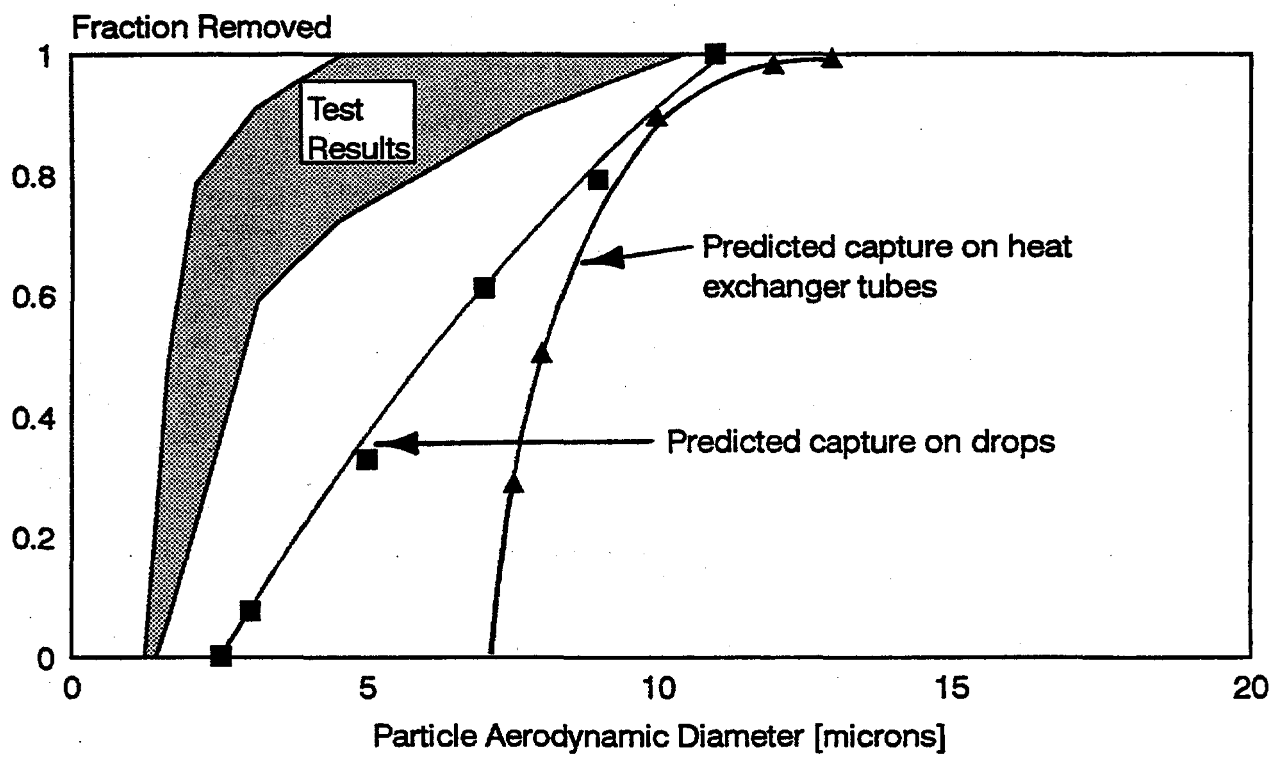

Figure 5-1. Prediction of particulate capture on condensate drops within the condensing economizer. Comparison with both predicted capture on heat exchanger tubes and observed results.

The observation that small impactors (drops in the case above) located within the high velocity areas of the economizer can significantly affect particle removal leads to a concept for improving particle capture in condensing economizers. The idea involves the addition of small diameter rods or fibers within the tube bank to act as impaction sites. These "seeded" impactors would collect particles which would then be rinsed away by draining condensate. In one approach to this, analyzed below, the added impactors are inserted perpendicular to the tube axes. Figure 5-2 shows one possible arrangement in which the impactors are woven around the tubes. An advantage of this approach is that the impactors cross the planes of highest velocity within the economizer (see Appendix A). An alternative, possibly simpler configuration, is shown in Figure 5-3. Here the impactor fibers are positioned along the bottom of diagonal rows. An advantage of this configuration is that condensate falling from the bottom of the tubes would likely flow along and rinse the added impactors.

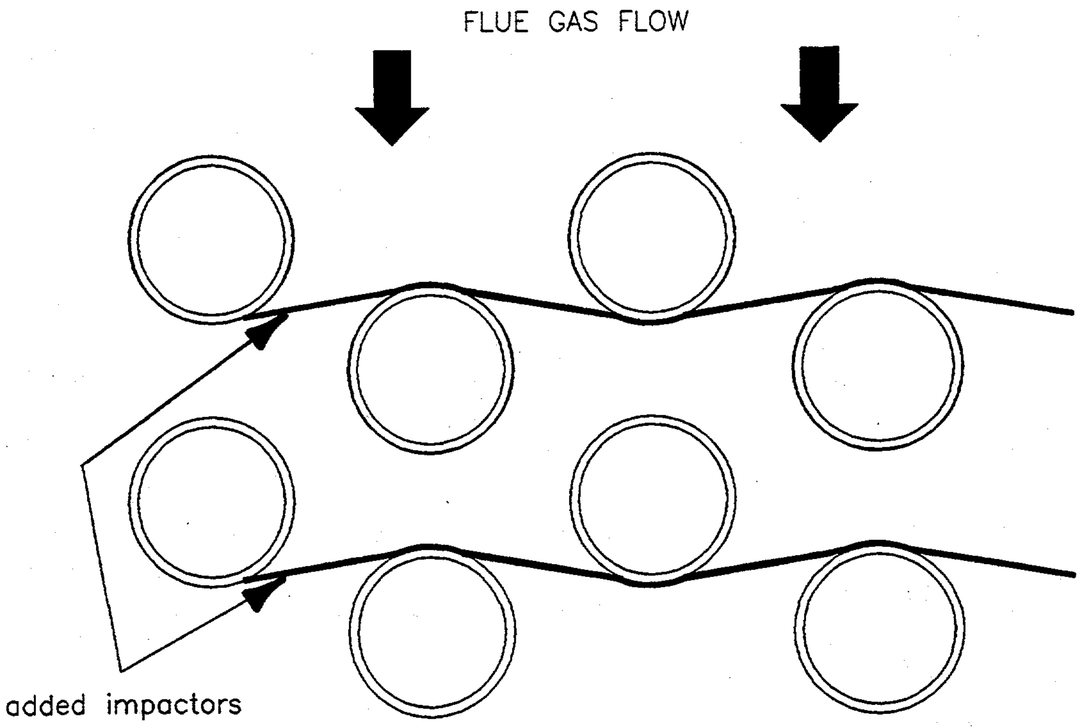

Figure 5-2. Illustration of the addition of impaction surfaces to the condensing economizer. Impactors woven around tubes.

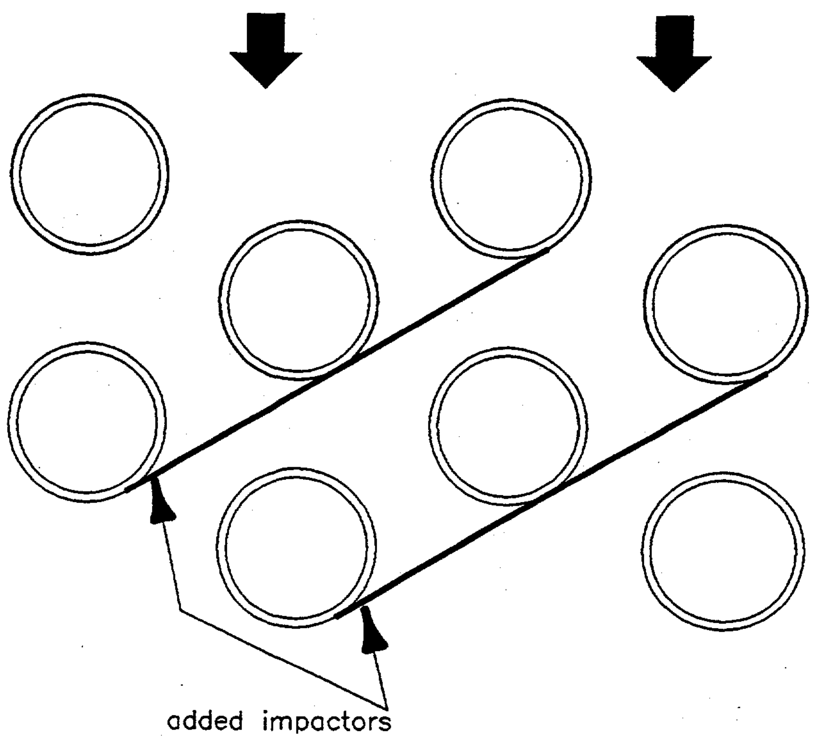

Figure 5-3. Illustration of the addition of impaction surfaces to the condensing economizer. Impactors positioned along bottom of diagonal rows.

In analyzing the expected capture on the small rods or fibers, the same approach to estimating capture efficiency as used for the cylindrical tubes themselves was again used (see Appendix B). Pressure drop due to the added impactors was done by calculating drag force on the added rods or fibers and then balancing this against pressure drop. In Figure 5-4, predicted capture on fibers of a range of diameters is shown using the assumption that 20% of the flow area of each row is blocked by the collectors. This figure shows that really significant removal could be achieved by the added impactors and it also illustrates the benefits of using as small a diameter impactor as practically possible.

The heat transfer predictions made for the economizers included in this study have shown that condensation may only be occurring in the later portions of the economizer tube bank. In this case, it might be best to add the small impactors only in the second half of the economizer. An important assumption made in estimating particle removal on the added impactors is that the sticking coefficient is 1 - if the particle hits the impactor it is removed. While this seems very reasonable for wet conditions, it may not be reasonable in the top half of the economizer where condensation is not occurring. Also, any particles which do collect on added impactors in the first (non-condensing) half of the economizer would not be removed by draining condensate. Figure 5-5 again shows predicted capture on added impactors with a range of diameters but in this case the impactors have only been added the second half of the economizer.

Figure 5-6 shows the effect of area blockage on particle removal with impactors of fixed diameter (1mm) added throughout the economizer. Figure 5-7 shows the same situation with the impactors added only in the second half of the economizer.

An important consideration in the addition of small impactors as suggested here is the increased pressure drop. Results of predictions for this are shown in Figure 5-8 with 1mm impactors. These pressure drops were calculated by assuming that the pressure drop in each row open area which has an added impactor balances the drag force on the impactor cylinder.

As the diameter of the impactors decrease, their effectiveness in removing particles increases. The lower particle limit on size may be set by considerations of mechanical strength. In addition, very small impactors in the condensing section are likely to become wet and have larger effective diameters in any case.

To illustrate one case - which in the authors' opinion may be practical - it might be assumed that 1mm impactors are added only to the second half of a condensing economizer with a 20% blockage fraction. For the case considered here, the overall particle removal efficiency due to just the added impactors is estimated to be 98%. This removal might be increased further by inertial impaction on condensate water drops. The increased pressure drop due to the added impactors would be about 800 Pa [3.2 in. of water].

The concept of adding the impactors within the gas flow passages of the condensing economizer could be considered as simply the integration of a downstream mist eliminator with the economizer. Advantages of this integration are: 1) within the economizer impactor surfaces will be continuously

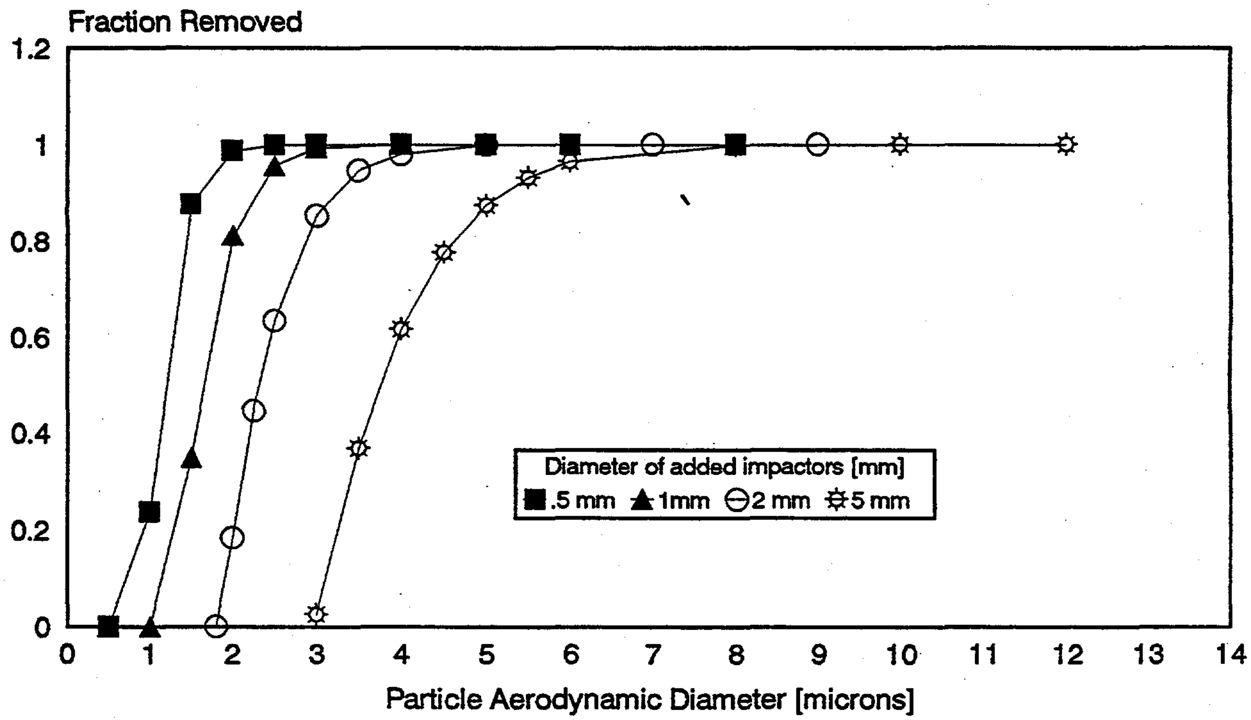

Figure 5-4. Illustration of particle capture by added impactors in condensing economizer. Effect of impactor diameter. Assuming 20% blockage of minimum flow area by cylindrical impactors throughout the economizer.

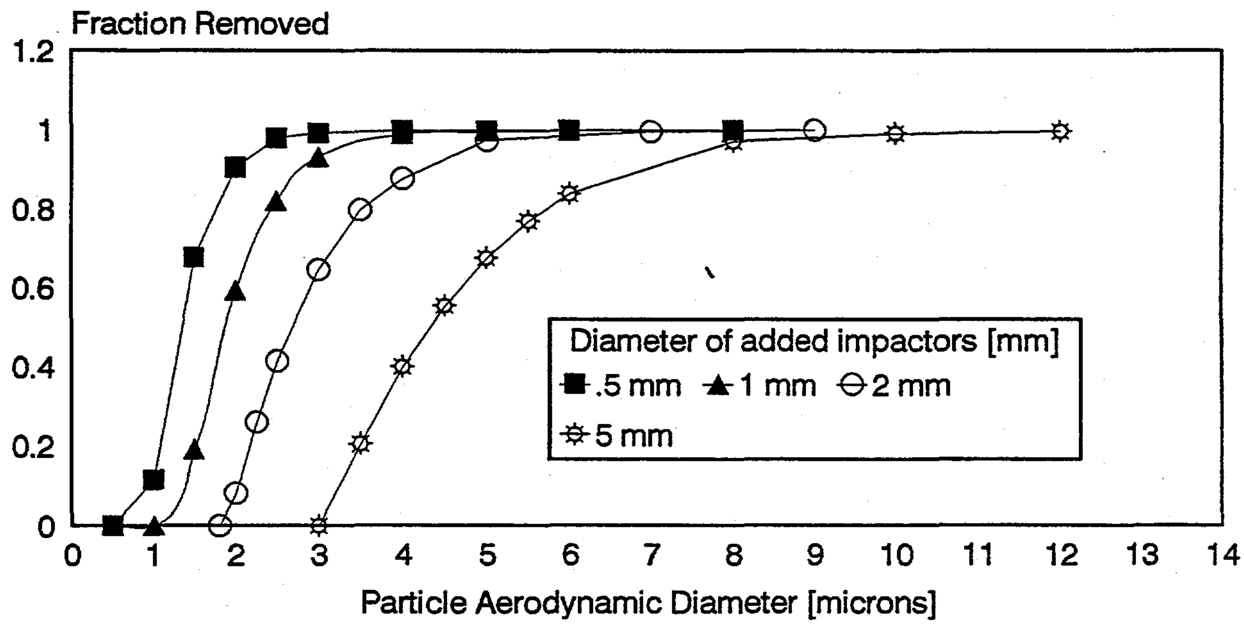

Figure 5-5. Illustration of particle capture by added impactors in condensing economizer. Effect of impactor diameter. Assuming 20% blockage of minimum flow area by cylindrical impactors only placed in second half of the economizer.

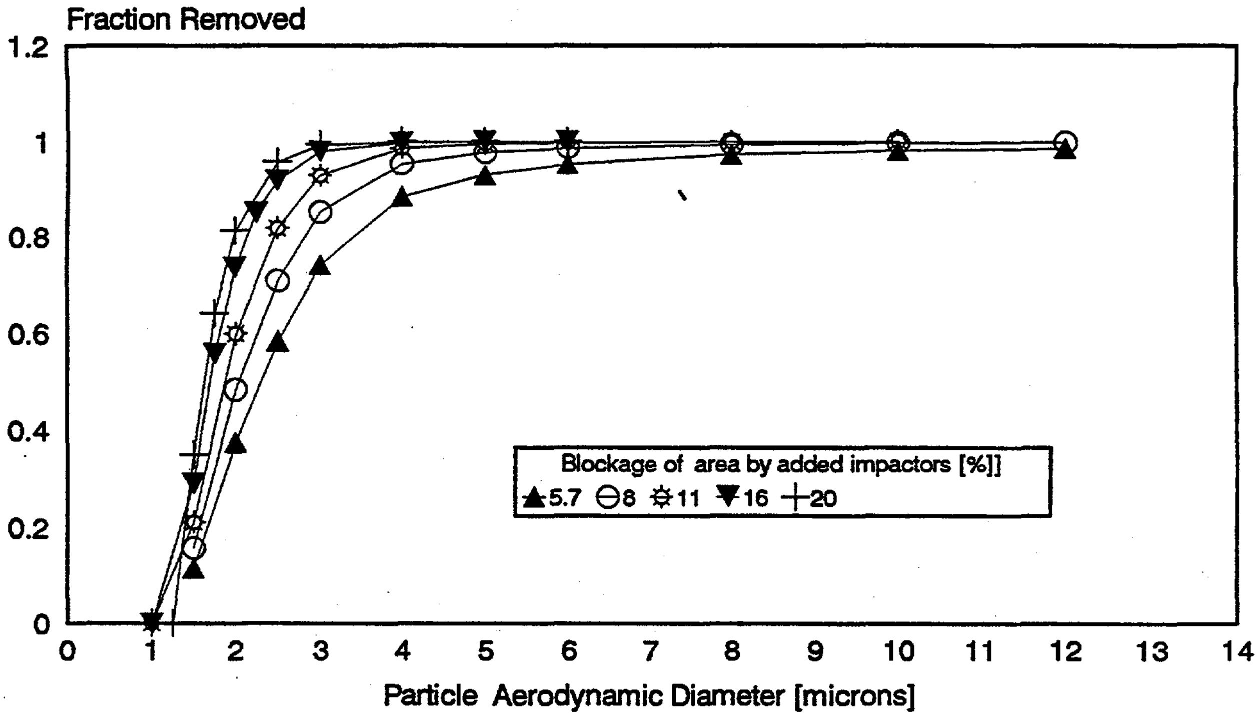

Figure 5-6. Illustration of particle capture by added impactors in condensing economizer. Effect of area blockcage. Assuming placement of 1 mm cylindrical impactors throughout the economizer.

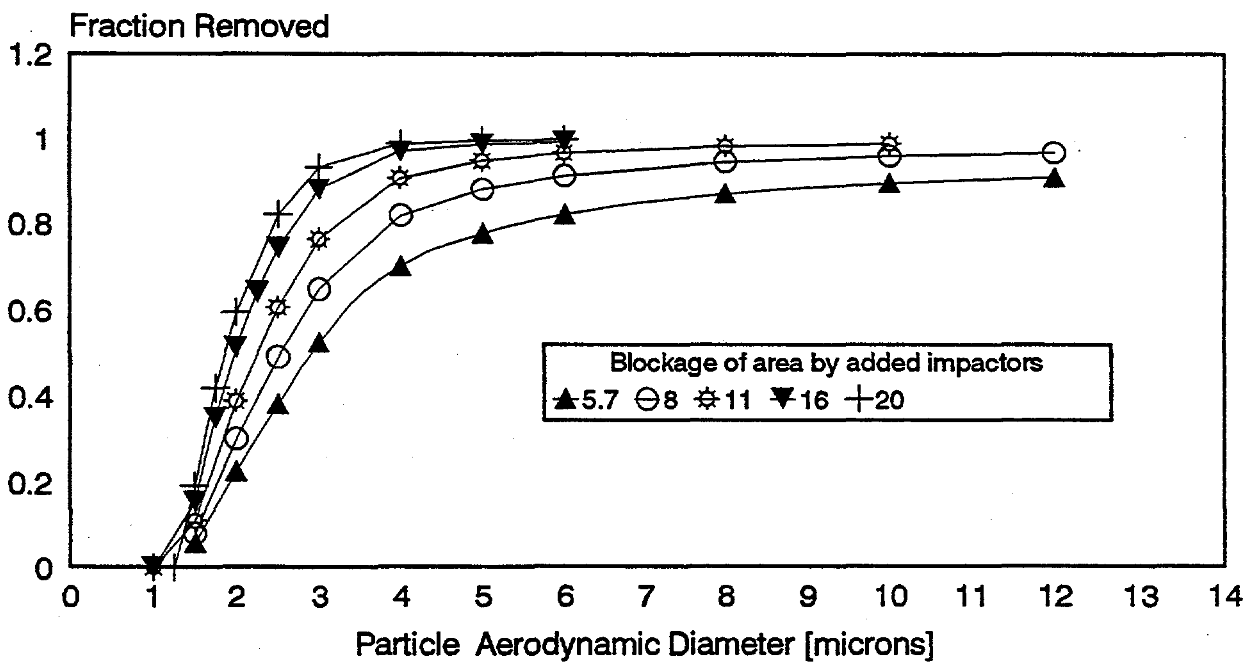

Figure 5-7. Illustration of particle capture by added impactors in condensing economizer. Effect of area blockcage. Assuming placement of 1mm cylindrical impactors only in second half of economizer.

rinsed by condensing water, avoiding pluggage and the need for a mist eliminator washing system, and 2) putting the impactors directly within the economizer takes advantage of the high velocities within the tube bank to achieve high impactor target efficiencies.

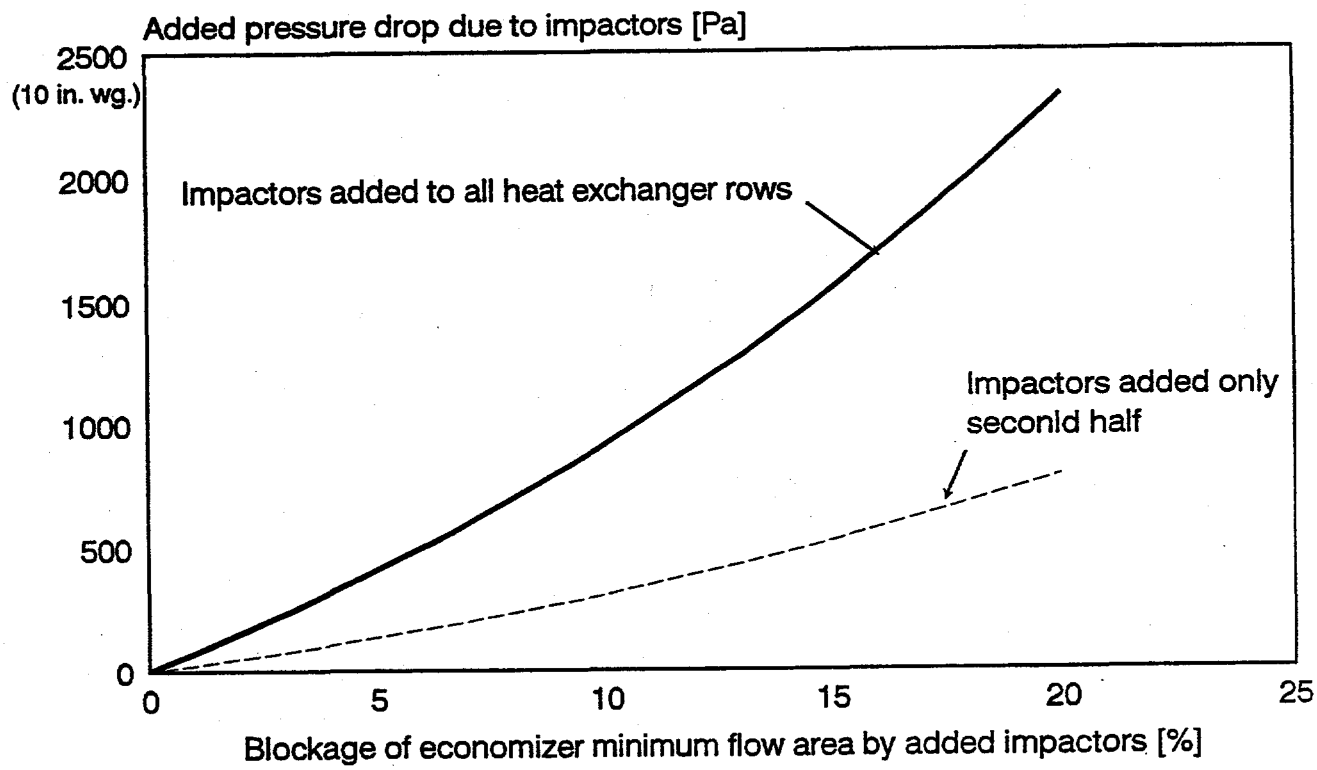

Figure 5-8. Effect of added impactors in condensing economizer on the pressure drop. At condition evaluated the pressure drop across the economizer without the added impactors is 1244 Pa [5 inches of water].

Perhaps the most important question which can be addressed by the results of this work is, the basic feasibility of using condensing economizers as the sole particle removal device in small coal-water slurry systems. These results have shown that high particle removal efficiencies clearly can be attained in condensing economizers and removal efficiencies can best be enhanced by factors which improve inertial impaction. The highest removal efficiencies were obtained with water sprays directly on the economizer tubes. In many applications, however, this would greatly reduce heat recovery and largely defeat the purpose of the economizer. (It might have a "compromise" situation in which water sprays are used only in the second half.) The concept of the added, small diameter impactors presented above could provide the high removal efficiencies observed with the water sprays (and probably better) without the negative impacts on the economizer heat recovery.

Since inertial impaction has been found to be the dominant particle collection mechanism any practical use must consider the possibility that collection efficiciency will decrease at low boiler loads. Several approaches could be considered for managing this situation. Condensing economizers could be installed in modules operating in parallel with independent induced draft fans. At low loads one or more modules could be closed off to maintain high gas flow in the remaining economizer(s). Recycle of flue gas from the economizer discharge to the economizer inlet could be used with increasing gas recycle as load decreases. Finally, water sprays might be considered for use only at low loads. The second and third options would reduce heat transfer effectiveness of the condensing economizer and this must be evalated in specific applications.

SECTION 6. CONCLUSIONS AND RECOMMENDATIONS

The following conclusions can be made based upon the studies conducted during this project:

- Condensing economizers can be used to control particulate emissions from coal water slurry-fired boilers. Particulate removal efficiencies as high as 98% were achieved in this study.

- For flyash typical of coal combustion, inertial impaction is the most important mechanism for capture in a condensing economizer. An important effect of this result is that removal efficiency is load dependent. Increasing firing rate increases gas flow, economizer pressure drop, and particle removal efficiency.

- Use of water sprays directly on the economizer tube bank increases particle collection efficiency. The improvement probably results from inertial impaction on water drops within the economizer.

- Increasing flue gas water vapor content upstream of the economizer increases water condensation rates, decreases flue gas inlet temperature, decreases total heat transfer rate, and has no significant effect on the particle removal efficiency. Calculations have shown that under conditions of the highest temperature difference between the flue gas and the tube surface, thermophoresis may contribute to capturing some of the smallest particles. Addition of water to the flue gas reduces the potential contribution of thermophoresis.

- A concept for adding small impaction surfaces throughout the condensing portion of the economizer has been presented. This can lead to improved particulate capture with only a modest increase in pressure drop.

- Conditions in the condensing economizers studied here really do not favor fog formation and particle growth. Particle growth which might have been caused by the presence of supersaturated conditions in the gas phase would improve downstream inertial collection.

For future development work and possibly commercial application, the following recommendations are made:

- Experimental verification should be done of the improved performance predicted with added small diameter impactors in the condensing part of the economizer.

- In commercial applications, consideration should be given to low load conditions and modular economizer design or some of the other approaches discussed in Section 5 to maintain high removal efficiency at low gas flow rates.

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APPENDIX A - METHOD OF CALCULATING THERMAL PERFORMANCE AND PRESSURE DROP

This Appendix describes the method used to calculate the latent and sensible heat transfer rates, the water vapor condensation rates, temperature profiles, and pressure drop through the heat exchangers. For heat transfer the method developed by Colburn and Hougen [1,2,3] has been used. This general method for evaluating the condensation of a single vapor from a non-condensible gas has been specifically applied to flue gas condensing economizers by Razgatis, et al. [4]. The analysis summarized here specifically applies to the Teflon-coated tube arrangements used in this project.

The situation analyzed is illustrated below. Flue gas passes around the heat exchanger tube and the cooling fluid (air or water) flows through the center. If the tube surface temperature is below the dew point for the flue gas water vapor, a condensate film will form as shown.

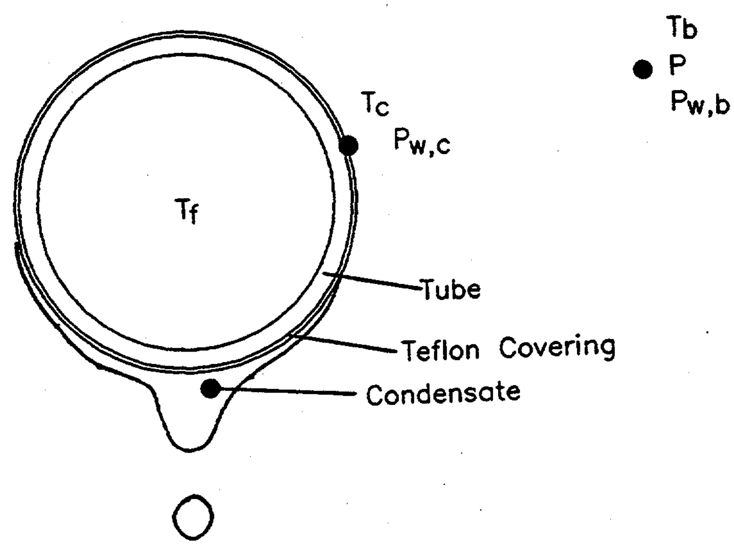

Figure A-1. Condensation on Heat Exchanger Tube

For the economizers used in this project the thickness of the Teflon covering was .381 mm (.015 inches). In the case of the water-cooled economizer the covering accounts for about 25% of the overall heat transfer resistance. In the air-cooled case the much higher resistance on the inside of the tubes leads to a smaller relative importance of the covering. A heat balance is prepared where heat flow from the flue gas to the condensate film (or, if the tube surface temperature is above the water vapor dewpoint, the tube surface temperature) 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 to the Tube Surface	=	Heat Transfer to the Cooling Fluid

or:

$$h_o(T_g - T_c) + h_{fg} K_m P_g \ln\left[\frac{P - P_{w,c}}{P - P_{w,b}}\right] = h_i(T_c - T_f) \quad \text{(A1)}$$

where:

h_i = heat transfer coefficient between film and cooling fluid. Includes inside convective coefficient and thermal resistance of the tube and Teflon covering.
h_o = outside convection coefficient
h_{fg} = water vapor latent heat of vaporization
K_m = water vapor mass transfer coefficient
P = total pressure (roughly atmospheric)
$P_{w,c}$ = water vapor partial pressure (saturation) at the condensate temperature
$P_{w,b}$ = water vapor partial pressure in the bulk flue gas
T_b = flue gas bulk temperature
T_c = condensate film temperature
T_f = cooling fluid temperature

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

$$\ln\left[\frac{P_{w,c}}{P_c}\right]=A_1-\frac{A_2}{T_c-A_3} \tag{A2}$$

where

P_c	= critical pressure
A_1, A_2, A_3,	= constants dependent upon units (see [5])

The outside film coefficient (h_o) is dependent upon gas properties, geometry and gas flow rate. Correlations by Kays and London [6] for flows normal to staggered tube banks have been used (see also [4]). With gas flowing through a bank of staggered tubes there is an entrance effect which leads to a lower outside film coefficient in the beginning rows. This effect has been included in the performance calculation method used here (see [4] and [6]). The water vapor mass transfer coefficient has been calculated by analogy with the outside convection coefficient [2]. The inside film coefficient has been calculated using standard relations. For the water-cooled case, this is very straightforward and the simple "Dittus-Boelter" correlation was used [7]. In this case great accuracy is not required because the heat transfer resistance provided by the inside film coefficient is negligibly small compared to the overall resistance. With the air-cooled exchanger, this situation is complicated by a transition Reynolds number (in our experimental arrangement), high resistance due to the internal film coefficient, and a very short tube length. 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 effect of the part of the tube outside (not in contact with the flue gas) was analyzed and these tube extensions were found to contribute significantly to the effective internal heat transfer resistance, essentially acting as fins. The situation is illustrated in Figure A-2. (Note: see Section 2 of report for full description of economizers) These extensions are particularly important in the case of the air-cooled exchanger, where forced convection from the inside of these tube ends (Fig. A-2 "conv 2" and natural convection from the outside of these ends (Fig. A-2 "conv 3") add 50-70% to the heat transfer rate. These effects were included by increasing the effective internal convective heat transfer coefficient.

Calculations are done on each section of the heat exchanger assuming constant properties across the section. For the results shown in this report, each section included two rows of tubes in the case of the air-cooled economizer and four rows in the water-cooled case. Within each section equations A1 and A2 were solved simultaneously by iteration. Calculations with more and less rows in each section did not indicate a significant difference in the overall heat exchanger performance.

Figures A-3 to A-7 illustrate typical results of a heat transfer analysis done for the water-cooled economizer. For this example it was assumed that the boiler fired a coal-water slurry with a coal concentration of 65% and a total heat input rate of 88 kW (300,000 Btu/hr). The overall excess air level was taken as 30%, the temperature of the flue gas entering the economizer was 175 C (347 F), and no additional water sprays were used either upstream or within the economizer.

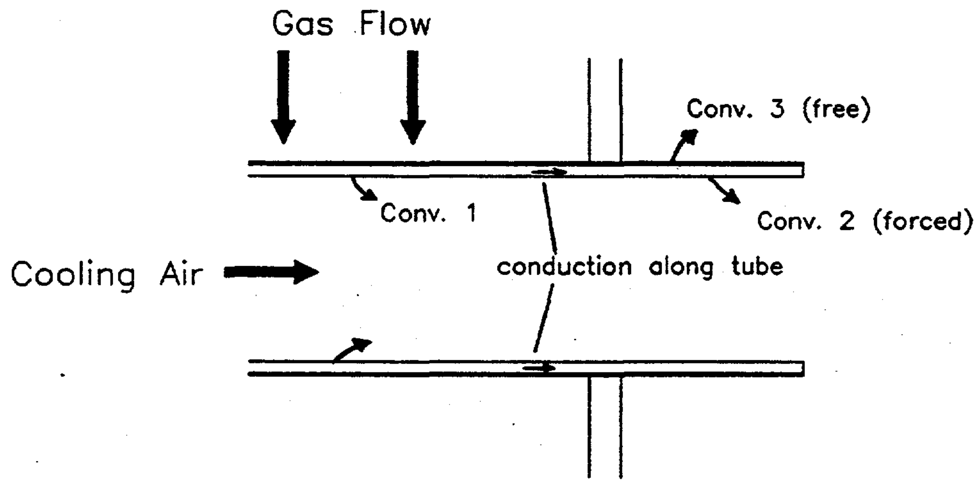

Figure A-2. Air-cooled condensing economizer. Illustration of the effect of tube ends on convective heat transfer.

Figure A-3 shows the profiles for flue gas temperature and tube outer surface temperature. The water vapor dewpoint here is about 48 C (119 F). For the first 25 rows of the economizer the surface temperature is above this and the gas is simply being cooled without condensation. At about row 25 condensation begins in this example and it can be noted that the bulk flue gas temperature is still well above the water vapor saturation temperature at this location. At this point latent heat transfer begins (Figure A-4) and the mole fraction of water vapor in the flue gas begins to decrease (Figure A-5). Figure A-6 shows the flue gas maximum velocity through the economizer and the gas Reynolds number based on this velocity. The maximum velocity is based on the minimum flow area within the heat exchanger and the location of this minimum area is shown in Figure A-8. The Reynolds number is based on the velocity at this minimum area and the hydraulic diameter of these areas. Figure A-7 shows the outside convective heat transfer coefficient through the economizer.

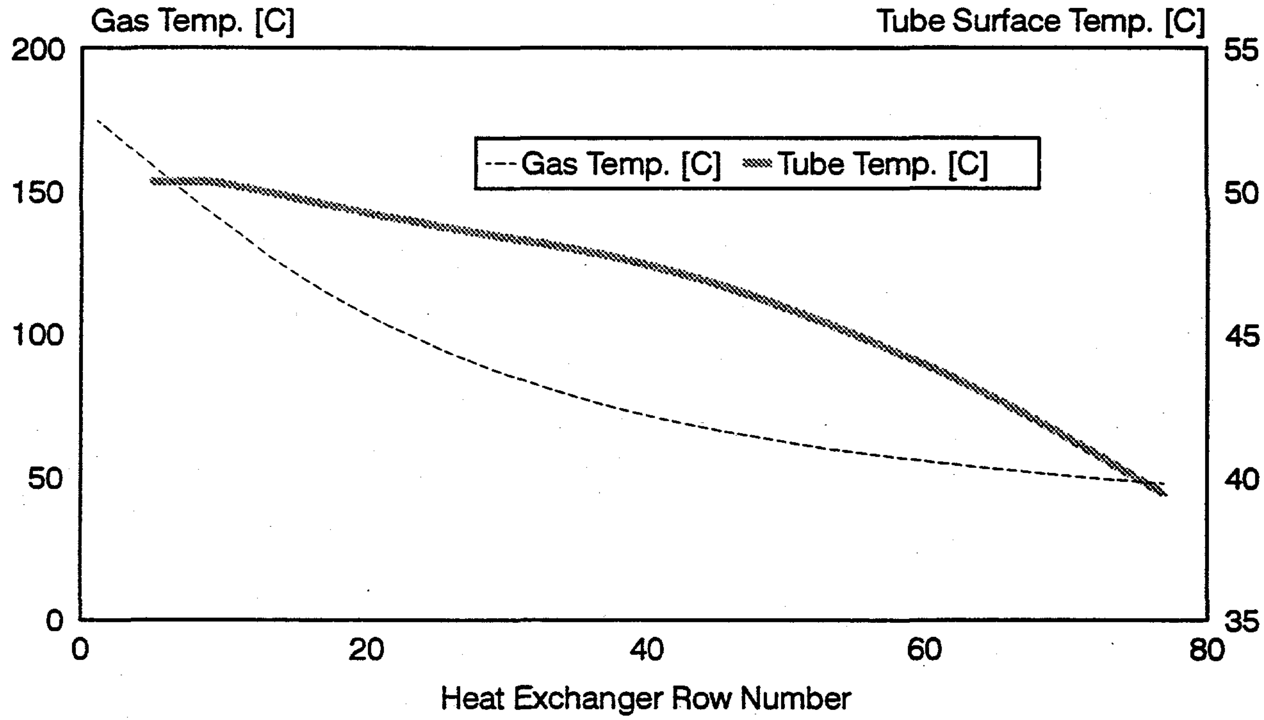

Figure A-3. Profile of flue gas temperature and tube condensate film or metal surface temperature through water-cooled condensing economizer. Results of calculations done for example case.

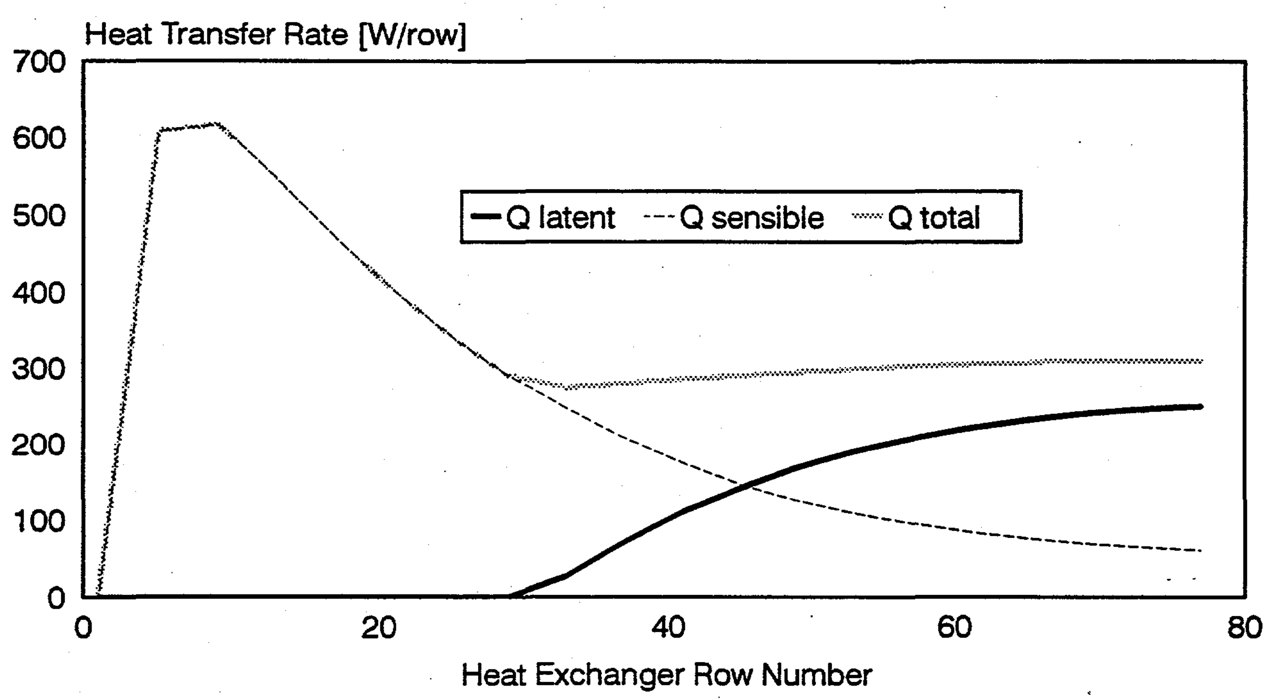

Figure A-4. Profile of local heat transfer rates - latent, sensible and total. Results of calculations done for example case.

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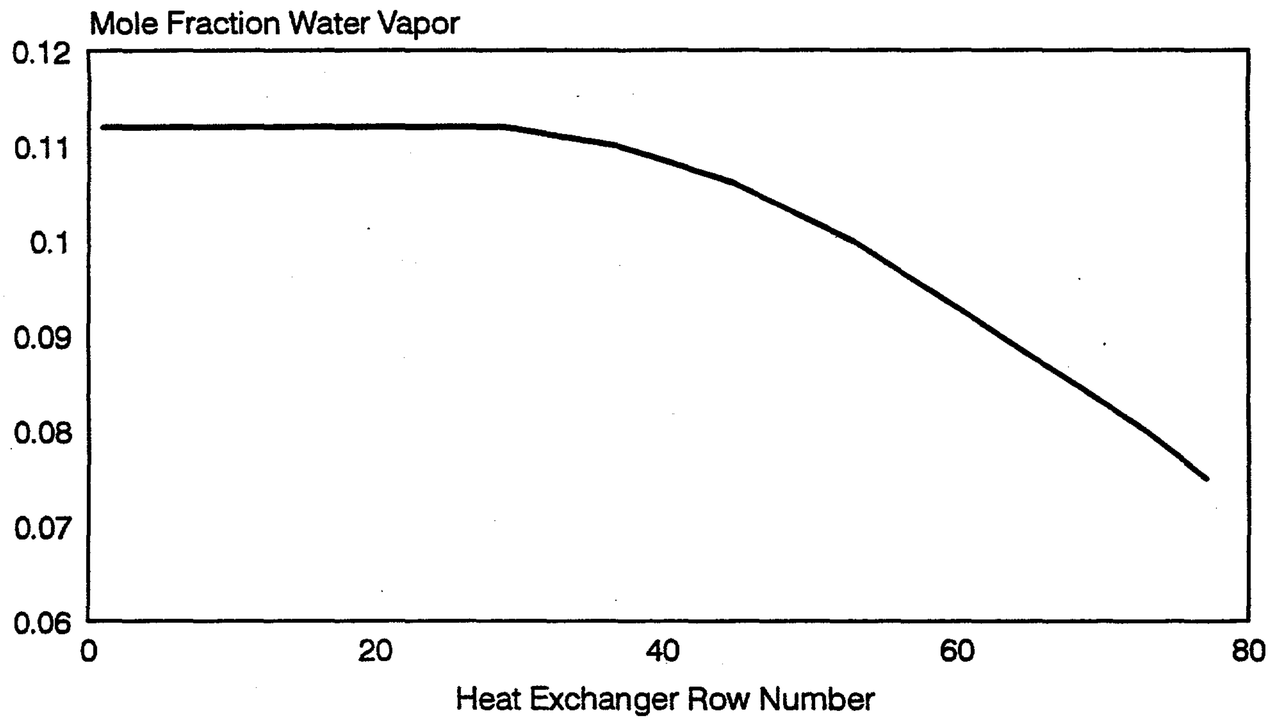

Figure A-5. Profile of flue gas water vapor content through water-cooled condensing economizer. Results of calculations done for example case.

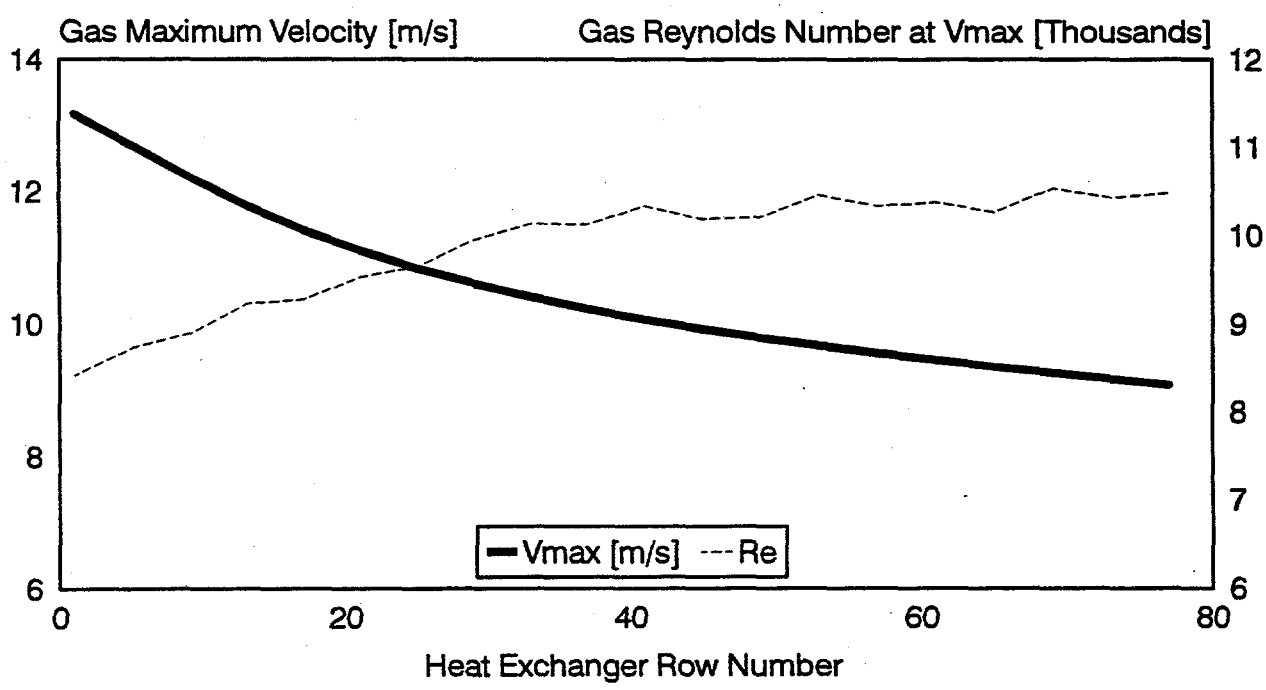

Figure A-6. Profile of maximum flue gas velocity and Reynolds number based on maximum velocity through water-cooled condensing economizer. Results of calculations done for example case.

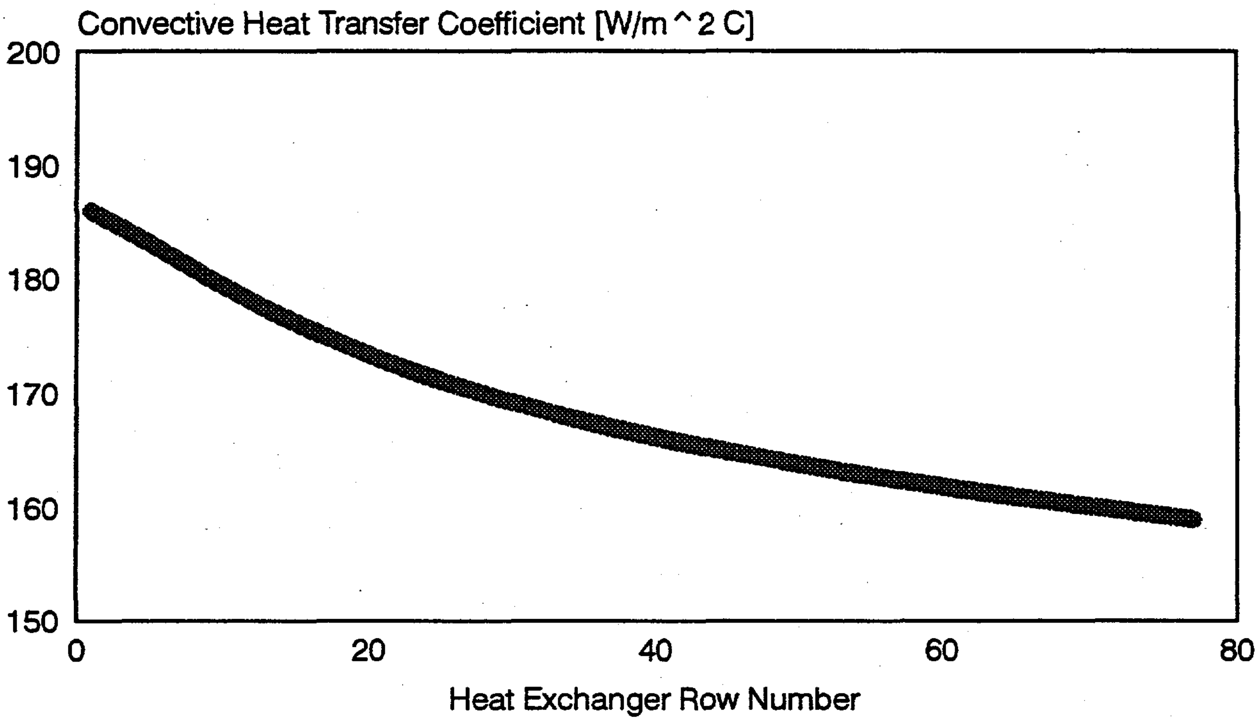

Figure A-7. Profile of convective coefficient for heat transfer between the flue gas and the outer tube surface through water-cooled condensing economizer. Results of calculations done for example case.

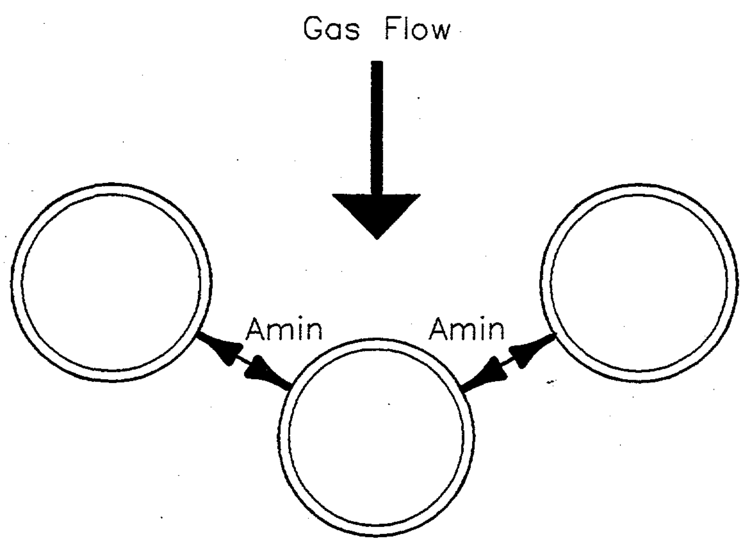

Figure A-8. Illustration of the location of the gas-side minimum flow area in the water-cooled condensing economizer.

For calculating flue gas pressure drop through the condensing economizer a correlation presented by Kays and London [6] has been used. The pressure drop due to friction is:

$$\Delta P = \frac{G^2}{2 \cdot g} \cdot f \cdot \frac{A}{A_c} \cdot v_m \tag{A3}$$

where:
- A = total heat exchanger surface area
- A_c = minimum free flow area
- f = friction factor
- g = gravitational constant
- G = flue gas mass flow/minimum free flow area
- v_m = flue gas specific volume.

The friction factor in equation A3 is, by the correlation used:

$$f = C_f \cdot Re^{-.18} \tag{A4}$$

where: C_f = a geometric constant based on the tube geometry (for water-cooled economizer C_f=0.47)
Re = Reynolds number based on minimum free flow area (Figure A-8)

In addition to friction pressure drop the pressure drop due to entrance and exit effects and velocity change across the economizer must also be considered. These effects combined were found to be fairly small for the arrangements used in this work - about 75 Pa (0.3 inches of water). The predicted and measured pressure drops in the economizer used here are discussed in the main part of this report. For the water-cooled economizer the pressure drop ranged from 622 to 1245 Pa (2.5 to 5.0 inches of water). At full load a pressure drop of 1245 Pa (5 inches of water) across an industrial condensing economizer is fairly typical. In some cases much higher pressure drops (to 2490 Pa (10 inches of water)) are used.

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APPENDIX B-METHOD OF ESTIMATING PARTICLE CAPTURE ACROSS THE CONDENSING ECONOMIZER

As flue gas flows through a condensing economizer particles can be captured on the tube outer surfaces and removed along with the condensate. This appendix describes methods to estimate the capture efficiency for particles. Mechanisms which can affect particle capture and which are discussed in this section include inertial impaction of particles on the tubes, interception, diffusion of particles to the tube surface, thermophoretic deposition (motion of particles in a temperature gradient), and particle growth due to direct condensation of water vapor on the particles. The growth of particles by this last mechanism can lead to capture on the tubes downstream by inertial impaction. Each of these is discussed individually below. Single tube and single row capture are first estimated and then this is extended to the entire heat exchanger.

INERTIAL IMPACTION

As flue gas flows around a tube, larger and heavier particles have too much momentum to "make the turn" and hit the tube surface. If the tube surface is wet with condensate, particles hitting the surface will stick and be washed off with the condensate. In estimating particle capture by inertial impaction results of a numerical study by Wessel and Righi have been used [1]. In this study finite difference methods were used to develop general correlations for particle capture for isolated cylinders in cross flow. Among other results in this work are included target efficiency which is defined as the fraction of particles within an inlet flow area defined by the projected area of the tube which impact on the tube surface. The target efficiency is a function of the Stokes and Reynolds numbers defined by:

$$\text{Stokes number} \quad Stk = \frac{\rho_p \cdot d_p^{\,2} \cdot U_o}{9 \cdot \mu_g \cdot d_t} \tag{B1}$$

and

$$\text{Reynolds number} \quad Re = \frac{\rho_g \cdot d_p \cdot U_o}{\mu_g} \tag{B2}$$

where: d_p = particle diameter
- d_t = tube diameter
- U_o = free stream gas velocity approaching isolated tube
- ρ_g = flue gas density
- ρ_p = density of flyash particle
- μ_g = flue gas viscosity

In correlating the results Wessel and Righi combined the Stokes and Reynolds number into an "effective" Stokes number, Stk_e (after Israel and Rosner [2]). The effective Stokes number is:

$$Stk_e = Stk \cdot \psi(Re) \tag{B3}$$

where: ψ = non-Stokes particle drag correction factor (see Ref. 1)

For the application considered in this project Stke is typically in the 0.125 to 0.5 range and here the isolated cylinder target efficiency is given by [1]:

$$_1 \cdot \ln(8 \cdot Stk_e) + \beta_2 \cdot (Stk_e - 0.125) + \beta_3 \cdot (Stk_e - 0. \tag{B4}$$

where: β_1 = 0.01978749
β_2 = 0.5136545
β_3 = -0.0482858
η_t = single cylinder target efficiency

The correlation used here is based on an isolated cylinder in a uniform flow at U_o. To apply this to the condensing economizer requires the selection of a velocity to consider as the equivalent of U_o. For this purpose the mean gas velocity at the minimum flow area of the row immediately preceding the target row has been used. This was illustrated in Appendix A.

For a row of tubes (row number "i") the removal efficiency for a particle of a given diameter is:

$$\eta_i = f_b \cdot \eta_{ti} \tag{B5}$$

where: f_b = fraction of heat exchanger open cross section area blocked by tubes in the row
η_{ti}= single cylinder target efficiency in row i
η_i = fraction of particles entering row i which are captured in row i

In a general row, i, the fraction of the particles which are in the flue gas at the economizer inlet and which are removed across row i is less than η_i because some of these particles are removed in rows upstream of i. The fraction of these entering particles which are removed in section i is:

$$\eta_{pi} = \eta_i \cdot \left(1 - \sum_{k=1}^{i-1} \eta_{pk}\right) \tag{B6}$$

where: η_{pi} = fraction of particles entering economizer which are removed in row i.

The fraction of entering particles removed through row i is simply the summation of η_{pi} from row 1 to row i.

INTERCEPTION

Particles which do not directly impact on the tube surface but which come closer to the surface of the economizer tubes than one particle radii (from the particle center) are captured by interception. For the particle and tube sizes considered here this effect is very small and can be neglected. To demonstrate this a solution in Flagen and Seinfeld [3] for laminar flow through filters beds can be used. The Kuwabara solution [4] for flow through such beds is used and the single cylinder collection efficiency for interception is:

$$\eta_{int} = 2 \cdot A_f \cdot \left(\frac{d_p}{d_t}\right)^2 \qquad \text{(B7)}$$

where: η_{int} = single cylinder interception collection efficiency

A_f = a parameter which depends on the fraction of the heat exchanger volume blocked by tubes (α):

$$A_f = 2 \cdot \left(\alpha - \frac{3}{4} - \frac{\alpha^2}{4} - \frac{1}{2} \cdot \ln\alpha\right) \qquad \text{(B8)}$$

d_p = particle diameter

d_t = tube diameter

For particle diameters ranging from 0.1 to 10 microns the single cylinder collection efficiency by interception estimated in this way is many orders of magnitude smaller than the collection efficiency by impaction.

DIFFUSION

In the turbulent flow through the economizer tube banks particles can move by molecular diffusion through the boundary layer to the tube surface. In this section an estimate of the rate of particle capture by diffusion (particle mass transfer) is presented using analogy with heat transfer. To do this it is first necessary to estimate particle diffusivity and the Stokes-Einstein relation from kinetic theory is used for this:

$$D = \frac{k \cdot T \cdot C_c}{3 \cdot \pi \cdot \mu \cdot C_c \cdot d_p} \tag{B9}$$

where:

C_c = slip correction factor
D = particle diffusivity
k = Boltzmann's constant
T = flue gas temperature

The diffusivity of particles is much lower than for diffusing molecules (water vapor for example) and diffusivity decreases with increasing particle size. Collection by diffusion is generally only significant for the smallest particles. Table B-1, below, lists particle diffusivities calculated for gas temperature of 100 C (225 F).

Table B-1.Particle Diffusivities

Particle Size (microns)	Diffusivity (m^2 / s)
.1	8.15×10^{-10}
.5	7.27×10^{-11}
1	3.07×10^{-11}
5	5.23×10^{-12}
7	3.69×10^{-12}

Kays and London [5] present correlations for heat transfer in flow through the banks in the form:

$$St \cdot Pr^{2/3} = Ch \cdot Re_f^{-0.4} \tag{B10}$$

where:

Ch =	a geometry constant (=.70 for our water-cooled economizer)
Pr =	Prandlt number, $\frac{\mu_g C_p}{k}$
C_p =	flue gas specific heat
k =	flue gas conductivity
Re_f =	Reynolds number for gas flow, $\frac{4r_h G}{\mu_g}$
r_h =	hydraulic radius of minimum flow area (= 1.03 cm, for water-cooled economizer)
G =	flue gas mass flow/minimum flow area
St =	heat transfer Stanton number, $\frac{h}{GC_p}$
h =	convective heat transfer coefficient

In estimating particle diffusion rates by analogy, equation B10 is again used but the heat transfer Stanton number is replaced by a mass transfer Stanton number and the Prandlt number is replaced by the Schmidt number as:

$$St_m \cdot Sc^{2/3} = Ch \cdot Re_f^{-0.4} \tag{B11}$$

where:

St_m = mass transfer Stanton number, $\frac{g}{G}$

g = mass transfer coefficient (used below)

Sc = Schmidt number, $\frac{\mu_g}{\rho_g D}$

(other variables defined earlier)

Using Equation B9, St_m and g can be estimated for a given gas temperature and flue gas mass flow rate. Given g, the particle deposition rate by diffusion can be calculated as:

$$\dot{m}_p = g \cdot m_p \tag{B12}$$

where:

m_p = mass fraction of particles in the flue gas in a given size range
$\dot{m}_p$ = particle deposition rate per unit area

If, as will be shown, the total fraction of particles removed by diffusion through the economizer is small then the mass fraction of particles in the flue gas at any point in the economizer can be taken as the same as at the economizer inlet. The mass flow rate of particles, in a given size range, through the economizer is simply: $M \bullet m_p$, where M = flue gas mass flow rate.

Overall removal efficiency through the economizer is the ratio of removal rate to mass flow rate of particles entering the economizer as:

$$\eta_d = \frac{100 \cdot g \cdot A}{M} \tag{B13}$$

where: A = total economizer surface area
η_d = collection efficiency by diffusion

To illustrate the magnitude of the diffusion effect, removal efficiencies are listed in Table B-2, below, for typical conditions in the water-cooled economizer. These removal efficiencies are clearly extremely small and could be completely ignored if they were not used as part of the procedure for estimating collection by thermophoresis, below.

Table B-2 Example Values of Particle Removal Efficiency by Diffusion in Water-Cooled Condensing Economizer (Flue Gas Temperature 107C [225F] and Flue Gas Mass Flow Rate: 159 kg/hr [350 lbs/hr])

Particle Size (microns)	Removal Efficiency (%)
.1	0.22
.5	6.6×10^{-2}
1	2.5×10^{-2}
5	7.8×10^{-3}
7	6.2×10^{-3}

THERMOPHORESIS

Thermophoresis is the motion of a particle in a gas resulting from a temperature gradient. Rosner [6] has shown that thermophoresis can be the most significant force leading to deposition of sub-micron particles in flue gas. The thermophoretic drift velocity for a particle in a temperature gradient can be expressed:

$$v = (\alpha_T D)_p \cdot \left(-\frac{\nabla T}{T}\right) \tag{B14}$$

where: $(\alpha_T D)_p$ = thermophoretic diffusivity
∇T = temperature gradient
T = average gas temperature

For particle larger than the mean free path of the flue gas the thermophoretic diffusivity can be expressed [7] as:

$$(\alpha_T D)_p = \frac{3 \cdot \mu_g \cdot C_c \cdot H}{2 \cdot \rho_g} \tag{B15}$$

In this expression H is:

$$H = \frac{1}{1+\frac{6\cdot\lambda}{d_p}} \cdot \frac{\frac{k_g}{k_p}+4.4\cdot\frac{\lambda}{d}}{1+2\cdot\frac{k_g}{k_p}+\frac{\lambda}{d_p}} \tag{B16}$$

where: λ = mean free path at bulk conditions
k_g = flue gas thermal conductivity
k_p = ash particle thermal conductivity

Table B-3, below, lists typical values for thermophoretic diffusivity at a flue gas temperature of 93 C (200 F).

Table B-3. Thermophoretic Particle Diffusivities.

Particle Size (microns)	Thermophoretic Diffusivity (m^2/s)
.1	$7.4 * 10^{-6}$
.5	$6.9 * 10^{-6}$
1	$5.7 * 10^{-6}$
5	$2.1 * 10^{-6}$
7	$1.6 * 10^{-6}$

Following the method of Rosner [6], particle capture due to thermophoresis can be expressed as a factor which multiplies capture by diffusion alone. This factor can be expressed:

$$F = \frac{Pe_s}{1-Pe_s} \tag{B17}$$

F = factor by which thermophoresis augments capture of particles by diffusion
Pe_s = Peclet number for thermophoresis

The Peclet number here can be estimated as:

$$Pe_s = (\alpha_T Le)_p \cdot \frac{Sc_p^{2/3}}{Pr^{2/3}} \cdot \left(\frac{T_b - T_c}{T_c}\right) \tag{B18}$$

where:
- T_b = bulk flue gas temperature (absolute)
- T_c = temperature of tube or condensate film surface (absolute)
- Sc_p = Schmidt number, the ratio of the flue gas kinematic viscosity to the particle diffusivity (see Table B-1)
- $(\alpha_T Le)_p$ = particle Lewis number for thermophoresis, the ratio of the particle thermophoretic diffusivity (see Table B-2) to the thermal (heat) diffusivity of the flue gas.

Using equations B-17 and B-18, the thermophoretic diffusivities of Table B-2, the particle diffusivities of Table B-1, the expected particle removal efficiency has been estimated for three values of the mean temperature difference between the flue gas and the heat exchanger surface. Results, listed in Table B-4 below show that significant removal efficiencies, by thermophoresis, can only be realized with high values of flue-gas to surface temperature difference.

Table B-4. Estimated Overall Particle Capture Efficiency by Thermophoretically Enhanced Diffusion in Water-Cooled Condensing Economizer

Particle Size (microns)	Removal Efficiency (%) Average Temperature Difference Between Gas and Surface (C):		
	20	50	100
.1	3.9	9.6	19.
.5	5.4	13.	27.
1	2.9	7.3	15.
5	1.1	2.7	5.4
7	0.8	2.0	4.1

DIRECT CONDENSATION OF WATER VAPOR ON PARTICLES

One additional mechanism which might be considered in condensing economizers is the direct condensation of water vapor on the particles, increasing their aerodynamic diameter, making them easier to collect by inertia. In fact this mechanism has been the basis of some types of particle collection devices which saturate the flue gas upstream of a condenser section. Direct steam injection into polluted air streams has also been used to enlarge particle in the steam jet [8]. In a condensing economizer this mechanism requires that some part of the flue gas between the bulk conditions and the surface conditions be supersaturated. The degree of supersaturation required to achieve condensation on the particles depends upon the affinity of the surface to water. Hydrophilic particles would require a lower level of supersaturation than hydrophobic particles.

The degree of supersaturation which might be present in the gas phase depends upon the flue gas state and the surface temperature. Figure B-1 illustrates a set of conditions which might exist at one point in a condensing economizer and which would not favor supersaturation. The bulk flue gas is at condition 1 where the flue gas mole fraction of water vapor is about 10%, which is a very typical level in the absence of water sprays. Point 2 represents the tube surface temperature and the mole fraction of water vapor corresponding to saturation at that temperature. Point 3 represents the saturation temperature corresponding to the water vapor content of the flue gas; it is the "dewpoint" of the flue gas. Since the surface temperature is below the saturation temperature water vapor will condense on the tube even though the bulk gas temperature remains well above the dewpoint. As a very rough approximation the state of the flue gas through the boundry layer can be represented by a straight line connecting points 1 and 2. Since this line does not pass below the saturation curve, the presence of a supersaturated region is unlikely.

Figure B-2 illustrates a case in which condensation of water vapor within the flue gas is much more likely. The flue gas in this case has a much greater water vapor content and is at the saturation temperature. The surface temperature is considerably colder than the case considered above and the approximate state line through the boundry layer passes below the saturation curve.

Figure B-3 shows the trend in the bulk flue gas temperature as well as the surface temperature through a condensing economizer for an example case. This represents the water-cooled economizer studied in this project and these points are based on results of heat transfer calculations (see Appendix A). For this example no additional water was added to the flue gas. Generally the conditions are not very favorable for the existance of a large supersaturated zone in the boundry layer. Under some conditions flue gas water vapor content was increased to as much as 22% (volume basis) but even under this condition it does not appear likely that the flue gas becomes supersaturated to a great degree. Still higher flue gas moisture contents and/or still lower surface temperatures might be effective in promoting condensation of water vapor on the particles in the flue gas.

SUMMARY

The presentation in this Appendix has shown the importance of inertial impaction in the capture of flyash particles through a condensing economizer and also the relatively weak contribution made by other mechanisms under the conditions studied here. Based on the review of these other mechanisms operating conditions which could improve capture can be identified. For example very low tube surface temperatures and hot, saturated flue gas would lead to a more significant role for condensation of water vapor on the particles with better downstream collection by inertial impaction. High levels of temperature difference between the gas and the tube surface would improve the role of thermophoresis. Considerations of the overall heat balance of the plant and the utility of the heat recovered would really deterimine the practicability of such conditions.

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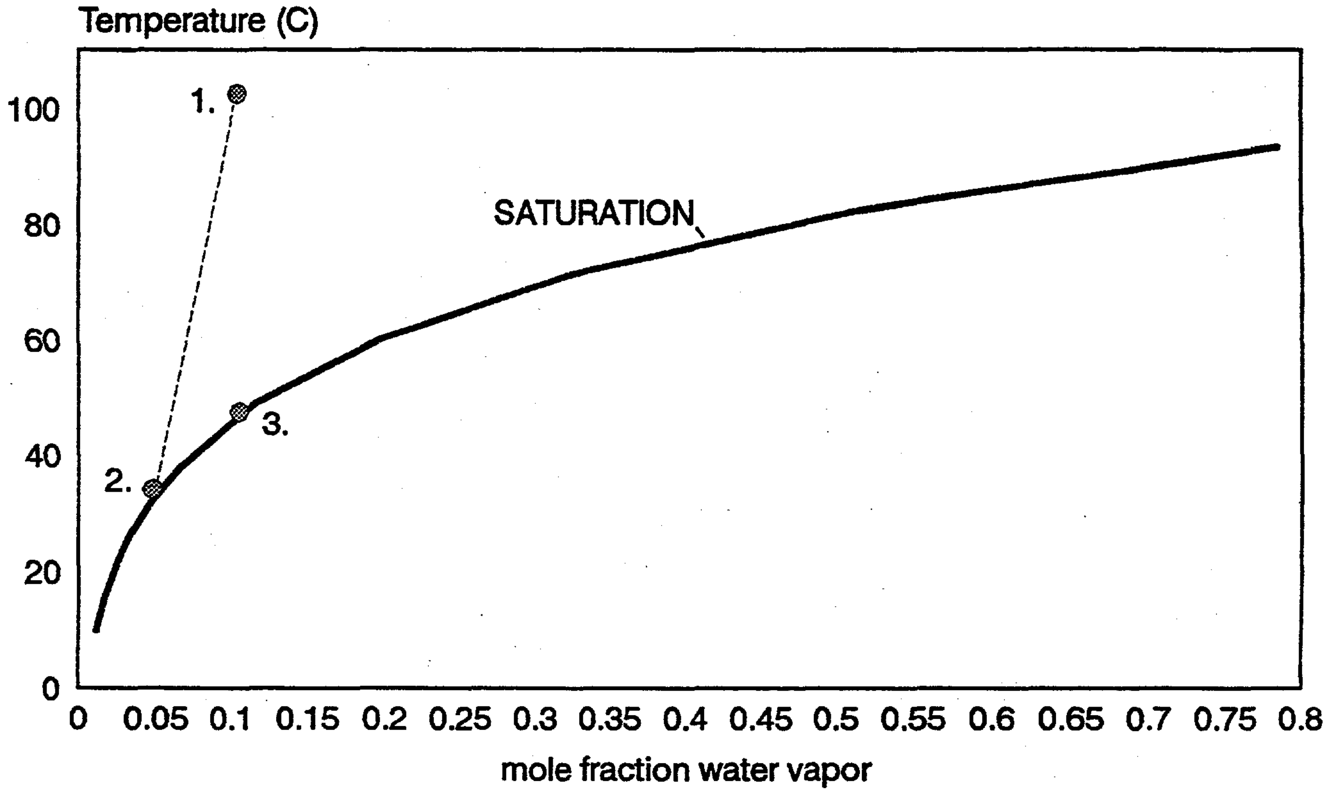

Figure B-1. Illustration of the conditions in economizer which would not favor condenstion of water vapor on particles. 1-bulk flue gas state, 2. tube surface, 3. saturation at bulk flue gas mole fraction water vapor.

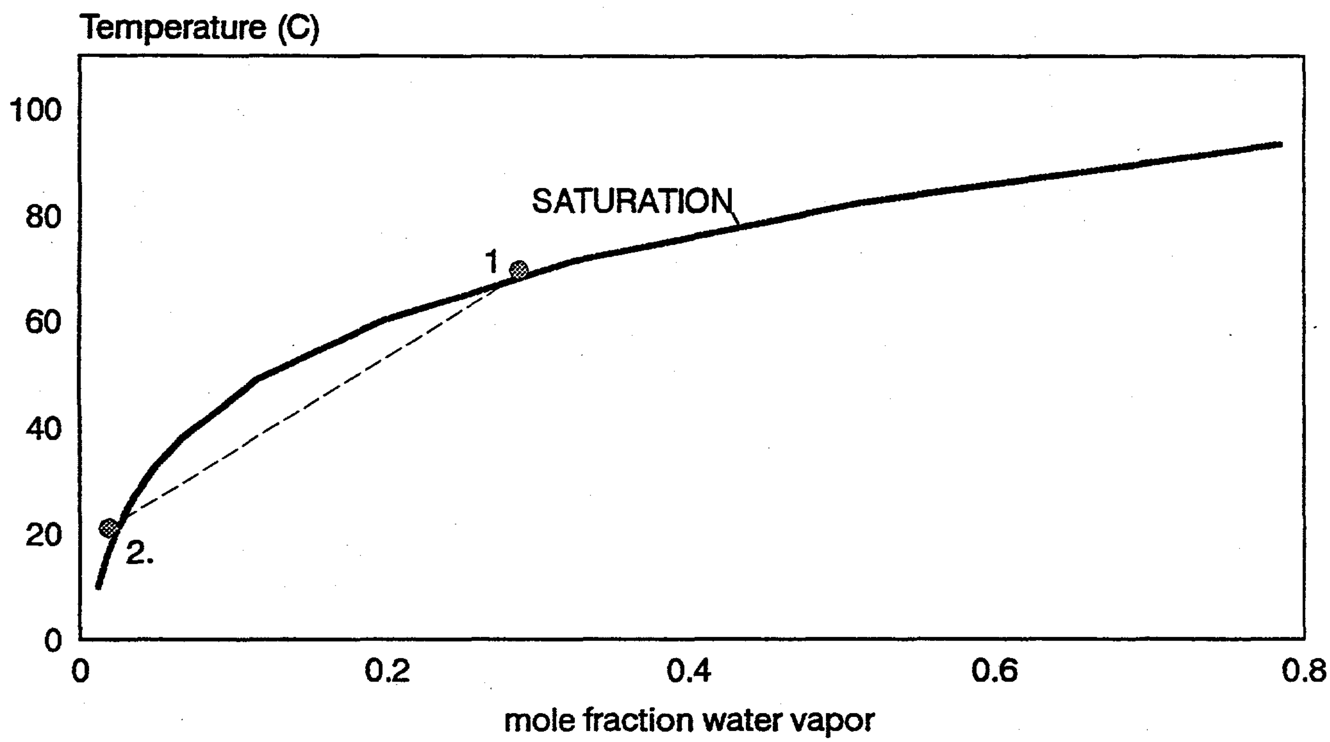

Figure B-2. Illustration of conditions in condensing economizer which would favor condensation of water vapor on particles. 1-bulk flue gas state, 2-tube surface.

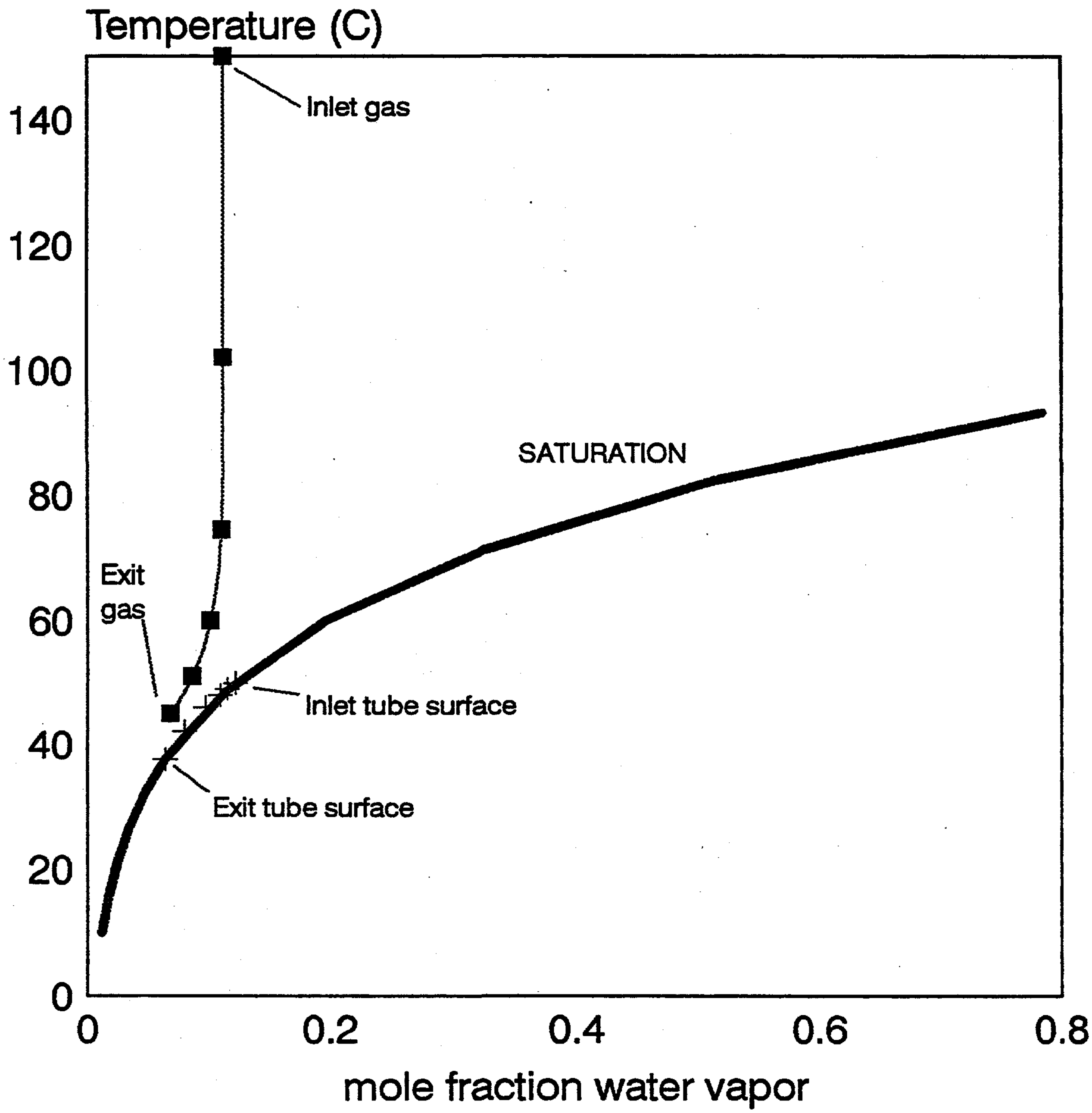

Figure B-3. Illustration of the change in temperature and water vapor content through the economizer. Example results of calculations. "Surface" points represent surface temperature and equilibrium water vapor content at surface temperature.