MILLIKEN CLEAN COAL TECHNOLOGY DEMONSTRATION PROJECT

HEAT PIPE PERFORMANCE – FINAL REPORT

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LIST OF ABBREVIATIONS

ABB API	ABB Air Preheater, Inc.
acfm	Actual Cubic Feet Per Minute
ASTM	American Society for Testing and Materials
Avg	Average
Btu	British Thermal Unit
CCT-IV	Clean Coal Technologies IV
CML	CAPCIS March Limited
CO_2	Carbon Dioxide
CONSOL R&D	CONSOL Inc., Research & Development Department
CS	Carbon Steel
dB	Decibel
EF	Degrees Fahrenheit
ft or ‘	Feet
Dia.	Diameter
Diff	Difference
DOE	Department of Energy
DP	Differential Pressure
DPU	Data Processing Unit
ECTC	Environmental Control Technology Center
ESEERCO	Empire State Electric Energy Research Corporation
emf	Electro Motive Force
EPA	Environmental Protection Agency
EPRI	Electric Power Research Institute
ESP	Electrostatic Precipitator
FD	Forced Draft
FG	Flue Gas
FGD	Flue Gas Desulfurization
ft	Feet
ft^3	Cubic Feet
gpm	Gallons Per Minute
H_2	Hydrogen
hp	Horse Power
hr	Hour
Hz	Hertz
ID	Inside Diameter
ID	Induced Draft (when referring to a fan)
in. or “	Inches
in. WC	Inches Water Column
KW	Kilowatts
KWh	Kilowatt hour
LACR	Low Alloy Corrosion Resistant
lb/hr	Pounds per Hour

LIST OF ABBREVIATIONS (Cont.)

Max	Maximum
MgO	Magnesium Oxide
$Mg(OH)_2$	Magnesium Hydroxide
mils/yr	0.001 inches per year
min.	Minute or Minimum
MM	Million
Moly	Molybdenum
MW	Megawatts
MW_{net}	Net Megawatts
N_2	Nitrogen
NH_4HSO_4	Ammonium Bisulfate
NO_x	Nitrogen Oxides
NYSEG	New York State Electric & Gas Corporation
NYSERDA	New York State Energy Research and Development Authority
O_2	Oxygen
OD	Outside Diameter
P	Pressure
PA	Primary Air
PF	Power Factor
PFG	Primary Flue Gas
ppm	Parts per Million
Pri.	Primary
psig	Pounds Per Square Inch Gage
PTC	Performance Test Code
rpm	Revolutions Per Minute
SA	Secondary Air
SCR	Selective Catalytic Reduction
S-H-U	Saarberg-Hölter Umwelttechnik
SNCR	Selective Non-Catalytic Reduction
SO_2	Sulfur Dioxide
SO_3	Sulfur Trioxide
scfm	Standard Cubic Feet per Minute at 60°F and 1 Atm.
SS	Stainless Steel
TC	Thermocouple
TEFC	Totally Enclosed Fan Cooled
V	Volt
W	Watt
WDPF	Westinghouse Distributed Processing Family
wt	Weight

ABSTRACT

A main feature of the Clean Coal Technologies IV Demonstration program at New York State Gas and Electric Company's Milliken Station is demonstration of retrofit SO_2 and NO_x control systems which have minimum impact on the overall plant heat rate. As part of the new system design, the original rotary regenerative air heaters on the Unit 2 boiler were replaced with heat pipe air heaters. Use of the heat pipe air heaters offered the potential of improving plant heat rate by: (1) allowing operation at lower flue gas outlet temperatures than the existing air heaters, and (2) elimination of combustion air leakage within the air heater which reduces the overall system fan power requirements. This report provides a comprehensive review of the air heater performance during the first 3.3 years of operation. Major areas covered include: mechanical design, materials selection, equipment layout, performance testing and performance, operating problems and solutions, and performance benefits.

1.0 EXECUTIVE SUMMARY

1.1 Program Goals and Results

A main goal of the Department of Energy's (DOE) Clean Coal Technologies IV test program at the New York State Gas & Electric Company's (NYSEG) Milliken Station is to demonstrate overall pollution abatement with increased energy efficiency. To reduce plant air emissions, SO_2 and NO_x control systems were retrofitted on both the Unit 1 and Unit 2 boilers. Innovative technologies, such as the use of heat pipe air heaters on the Unit 2 boiler were incorporated into the design to lessen the impact of the new emission control systems on the overall plant heat rate. The heat pipe air heaters were designed and manufactured by ABB Air Preheater Inc. (ABB API) of Wellsville, New York. Expected benefits of replacing the two original Ljungstrom® regenerative air heaters on the Unit 2 boiler with the heat pipes included: (1) higher heat recovery by allowing operation at a lower effective flue gas outlet temperature than the original air heaters, and (2) reduction in the overall boiler and flue gas desulfurization (FGD) system fan power requirements by elimination of the air leakage inherent in the design and operation of Ljungstrom air heaters.

Detailed tests and analyses indicate that the thermal performance of the heat pipes is about the same as the original air heaters. The goal of a 20°F reduction in the effective air heater flue gas outlet temperature was not achieved. However, the use of the heat pipe exchangers successfully reduced air heater leakage to near zero levels. This is improving the boiler heat rate by greatly reducing the fan power requirements for the system. At full boiler load, the fan power savings comparing Unit 2 with Unit 1 averaged 778 KW or about 0.49% of the gross load.

Cold-end fouling of the heat pipes is the main operating concern. The fouling reduces the thermal performance and increases the gas side pressure drops with time. Normally, the heat pipes must be washed every six months to remove cold-end deposits. Based on the most recent operations, there are indications that the period between washes at the Milliken Station can be extended by limiting the minimum boiler low load to about 80 MW. This practice helps to avoid excessively low cold-end temperatures at lower loads which increase fouling.

1.2 Initial Performance Problems and Solutions

The heat pipe air heaters were put into service in December 1994. The initial operations indicated that performance was significantly below design. The cause was traced to problems with the inlet air flow distribution to the heat pipes and to the use of impure naphthalene heat transfer fluid in some of the high temperature tubes. The naphthalene problem was due to suppliers not meeting the ABB API purity specifications. Analysis of heat pipe tube contents indicated that naphthalene contaminants had decomposed forming mixtures of noncondensing gases composed of hydrogen, carbon dioxide, and ethylene. The noncondensing gases reduced the heat pipe thermal efficiency by blanketing heat transfer surface and by raising operating pressures and temperatures of individual heat pipes.

To solve the air flow distribution problem, perforated plates were installed at the discharges of the primary air and secondary air fans. Condenser end baffle plates were also installed within the heat pipes to force combustion air flows away from potentially non active heat transfer zones into active zones.

The decomposition of naphthalene contaminants is believed to be a one time occurrence. Therefore, to remove the noncondensing gases, ABB API installed fill nipple valves on all the naphthalene tubes. The heat pipe tubes were then re-evacuated under cold conditions and vented under hot conditions. After these changes were made, performance tests were conducted during May 1996 and November 1996. The tests demonstrated that the heat pipes were meeting the design pressure drops and that the total air leakage into the flue gas side of the air heaters was low, averaging 3.0 wt. % and 1.6 wt. % of the inlet flue gas flow for the 2A and 2B heat pipes, respectively. The heat pipes were, however, designed to have zero percent air to flue gas leakage. Since the construction is all welded, it is unlikely that the combustion air is leaking into the lower pressure flue gas section. Rather, air infiltration at man way door seals and at sootblower wall penetrations is mainly responsible for the very small measured leak rate. For practical purposes, the heat pipes are zero leak air heaters and are considered to have met this design guarantee.

1.3 Heat Pipe Air Heater Thermal Performance

The ASME Code procedure for testing air heaters was followed to provide a consistent evaluation method agreed upon by both the purchaser and supplier. The thermal performance of the heat pipes, while reasonably good, did not meet the design guarantees. For the May 1996 tests, the totally corrected flue gas outlet temperature for the 2A heat pipe was 17°F-18°F above the 253°F design temperature and for the 2B heat pipe was 12°F above the design. For the November 1996 performance tests, the differences were slightly higher at 20°F-23°F for the 2A heat pipe and 15°F-16°F for the 2B heat pipe. Based on an analysis done by CONSOL R&D, the uncertainty in these results is ±4.4°F. These results mean that the desired thermal performance improvement of 0.5% was not achieved. This is based on a typical boiler efficiency improvement of 1% for every 35°F reduction in the flue gas outlet temperature (no leak condition) from an air heater. However, an energy loss to stack comparison indicates that the clean condition heat pipe thermal performance is equal to and no worse than the performance of the original Ljungstrom air heaters.

1.4 Measured Benefits of Reduced Leakage

Although the thermal performance of the new heat pipe air heaters was not better than the replaced Ljungstrom units, the use of the heat pipes provided considerable improvement in fan power requirements. This is shown by direct comparison of the Unit 1 and 2 operating results for similar conditions of boiler excess air and gross load. Such a comparison is justified since Milliken Units 1 and 2 are identical except for the use of Ljungstrom air heaters with hot primary air fans in Unit 1 and heat pipe air heaters with cold primary air fans in Unit 2. At 100 MW and 160 MW gross load, the Unit 2 combined power requirements for the primary air (PA), forced draft air (FD), and induced draft (ID) fans, averaged 0.67MW (900hp) and 0.78MW (1050hp) less than for Unit 1, respectively. Most of the power savings can be attributed to the lower combustion air and flue gas flows for the Unit 2 boiler due to the zero air leak operation of the heat pipe air heaters. The differences represent considerable power cost savings for the zero leak heat pipe system. Assuming incremental costs of 2.3¢/KWh and a 65% plant capacity factor, the 25-year life cycle power cost saving is estimated at $2.55MM. Actual power cost savings are likely to be greater since these results have not considered power reductions for the electrostatic precipitator and the FGD system with optimized pumping (i.e., headers removed from service to accommodate reduced flue gas flow).

1.5 Cold-End Fouling

The main operating problem experienced with the heat pipe air heaters was flue gas side fouling of the cold-end tube banks. As with other types of utility boiler air heaters (Ljungstrom® and tubular units), the heat pipe fouling was associated with sulfuric acid condensation on heat transfer surfaces which are below the acid dew point. Hard fly ash deposits formed on the heat pipe tubes and fins, reducing the thermal performance and increased the flue gas side pressure drop. The fouling was promoted by direct gas flow impact since the worst fouled areas were against the gas flow on the top side of the tubes. The fouling was localized and limited to the cold-end tube banks.

The Milliken heat pipes were designed with a triangular-pitch, staggered-tube bundle layout throughout. The design provides high heat transfer and is compact. However, the design makes the cold-end difficult to clean by conventional sootblowing when sticky cementitious ash deposits form. For close-packed tubes, the staggered layout quickly dissipates most of the sootblower jet energy within the first two tube rows. During the heat pipe test program, attempts were made to improve the on-line cleaning of the cold-end tube banks. An Infrafone® was installed on the 2A heat pipe and four sootblower lances in the 2B heat pipe were modified by replacing the standard Bergemann ½" cone nozzles with special **e**" venturi nozzles. The Infrafone is a device which uses high intensity, ultra low frequency sound for the on-line cleaning of equipment. Neither the Infrafone nor the modified sootblower lances appeared to provide any significant cold-end cleaning benefit over the existing sootblowers. The Infrafone operation was discontinued after more than 300 days of service due to vibration-caused damage to ductwork and equipment.

Cold-end deposits, while a nuisance and detrimental to plant performance, can be removed by periodic water washing. Unlike for the Unit 1 Ljungstrom air heaters, which can be washed with the boiler on-line at low load, cleaning of the Unit 2 heat pipes requires that the boiler be shut down. This is because the heat pipes require some manual cleaning. At Milliken, the heat pipe air heaters are water washed approximately every six months. The best technique is to use a combination of deluge washing using the internal water spray headers with the air sootblowers in operation and manual washing with small low pressure hand lances to clean areas missed by the deluge washing.

The heat pipe performance results for the most recent six-month operating period (October 31, 1997 to April 24, 1998) indicate that it may be possible to extend the period between washes by limiting the minimum boiler load to about 80 MW, maintaining flue gas flow balance between the air heaters, and by bypassing some secondary air at off-peak load conditions. These adjustments help to prevent operation of the cold-end heat pipes at excessively low temperatures. During the last six-month operating period, the full load flue gas side pressure drops increased only about 1.0 in. WC compared to the normal 3-5 in. WC increase.

1.6 Conclusions and Recommendations

The ABB API heat pipe air heaters at Milliken are providing significant boiler operational benefits through elimination of air leakage associated with the originally installed air heaters. The combined power for the PA, FD, and ID fans in Unit 1 is typically more than 1,000 hp greater than that for Unit 2 under full boiler load conditions. These results indicate that the use of heat pipe air heaters in coal-fired boiler applications can provide significant capital and operating cost benefits, particularly for new plants where advantage can be taken of the zero air leakage design to reduce downstream equipment sizes for ID fans, particulate collectors, FGD scrubber systems, and stacks. The Milliken Station experience showed that after the naphthalene contamination problem was corrected, the operation of the heat pipes was trouble free for all but the cold-end tube modules. In order for the heat pipe air heaters to meet their full potential, progress must be made to improve the on-line cleaning of the cold-end sections; otherwise, the units should be operated with higher flue gas outlet temperatures above the acid dewpoint to avoid cold-end fouling. Possible actions to improve cold-end cleaning and reduce fouling include:

1. Relocate some of the upper level sootblowers to increase the number of sootblowers around the cold-end modules. This would increase the sootblower coverage. Inspections of the heat pipes have shown that the upper level sootblowers are probably not necessary since tube metal temperatures are above the acid dew point and the fly ash does not stick to the tubes.
2. Split the eight tube row deep cold-end module into two four-tube row deep modules with a level of sootblowers between. This would improve cleaning by reducing the required blowing penetration for the sootblowers.

3. Replace the staggered tube layout cold-end module with an in-line tube layout. This would help to provide deeper penetration of the sootblower jets but would require more tubes than the current staggered arrangement.

4. Replace the finned tube cold-end module with a smooth tube module. A no-fin design would require more tubes since the heat transfer per tube would be reduced but cleaning should be easier since there would be less support for deposit adherence.

5. Change the orientation of the sootblowers from perpendicular to the tubes to parallel with the tubes. This would help increase sootblower penetration by providing better alignment of the sootblower jet with the flow channels through the tube bank.

6. Reduce the flue gas SO_3 level to the heat pipe air heaters by injecting additives such as $Mg(OH)_2$ or MgO into the boiler. Reducing the flue gas SO_3 level would decrease the acid dew point and allow lower temperature operation without condensation. This form of SO_3 control is now used mainly in oil-fired boilers and several companies can supply the reagents. However, tests are recommended for this option to determine the cost/benefits and to establish the impact, if any, on the ESP particulate collector and the FGD scrubber system.

Recommendations one and six are the easiest to achieve at the Milliken Station. Because of access limitations around the heat pipe air heaters, the other recommendations are likely to be difficult to implement and costly. Recommendations two through five are better suited for consideration in a new system design.

There is a concern that the heat pipe thermal performance may be slowly degrading due to loss of naphthalene. This is due to the installation of purge valves on the fill stems of all naphthalene heat pipe tubes. The valves were installed to vent noncondensing gases which were generated by small amounts of naphthalene contaminants. After purging the gases, the valves were closed, capped and left in place. This provides the ability to again vent the tubes if additional decomposition were to occur but presents a continuing potential for naphthalene leakage past valve stem seals. Normally, the fill tubes are crimped shut and the ends seal welded to prevent any possible fluid loss. Plant personnel have periodically used a photo ionization detector to check for heat pipe condenser end naphthalene leaks when the heat pipes were in operation. The checks have shown varying levels of naphthalene at the test ports. The last check done in December 1997 showed a steep decline in naphthalene levels at all test ports. This likely means that the leaking tubes are now empty. For the Milliken heat pipe installation, periodic naphthalene leak checks will continue to be necessary to determine if additional tubes begin to leak. If this occurs, it may be necessary to remove the fill stem valves, refill the empty tubes and then to crimp and seal weld the fill stems.

Finally, the Milliken Station heat pipe air heater experience has pointed out the need for better quality control of the heat transfer fluids used in the fabrication. Fluid purity is critical if good performance and long-term operability are to be achieved. It is recommended that both the vendor and purchaser confirm the purity of each chemical batch. This would provide a double check and help to insure against noncondensable gas generation from contaminants.

2.0 INTRODUCTION

2.1 Use of Air Heaters in Utility Boilers

The hot flue gases from coal-fired electric utility boilers contain significant amounts of thermal energy. At 650°F, the sensible heat of the flue gas leaving a boiler economizer is typically about 15 percent of the fuel energy. Common practice is to recover most of this energy by preheating the combustion air in recuperative or regenerative heat exchangers.

In a recuperative heat exchanger, the flue gas and air streams are separated by the heat transfer surface. Heat energy from the flue gas is transferred directly across the heat transfer surface to the air. Tubular air heaters (Figure 1) in which the hot flue gases pass through metal tubes with air passing around the outside of the tubes are the most common type of recuperative heat exchanger used. These units provide a passive operating design with no moving parts and, when new, can have a low or zero leakage between the air and flue gas sides. The units are, however, physically large as compared to other types of air heaters and are prone to cold-end corrosion and fouling if tube wall temperatures drop below the acid dew point of the flue gas. With time, air leakage increases as more and more tubes corrode through. With acid condensation, serious fouling can occur due to the formation of sticky fly ash/acid poultices. Poultice formation can plug tubes. For the remaining open tubes, this increases outlet flue gas temperatures and flue gas side pressure drops.

The rotating wheel (rotor) Ljungstrom type exchanger is the most common type of regenerative air heater used by utilities. In these units, heat is transferred indirectly from the hot flue gases to the cooler combustion air through an intermediate medium, in this case, a basketed rotor containing many corrugated metal plates (Figure 2). The corrugations separate the plates and provide a torturous path for gas or air to flow to improve the heat transfer. The rotor continuously turns through the flue gas and air streams. The metal plates in the rotor baskets absorb sensible heat from the flue gas as the rotor turns through the flue gas side of the exchanger. This heat is transferred to

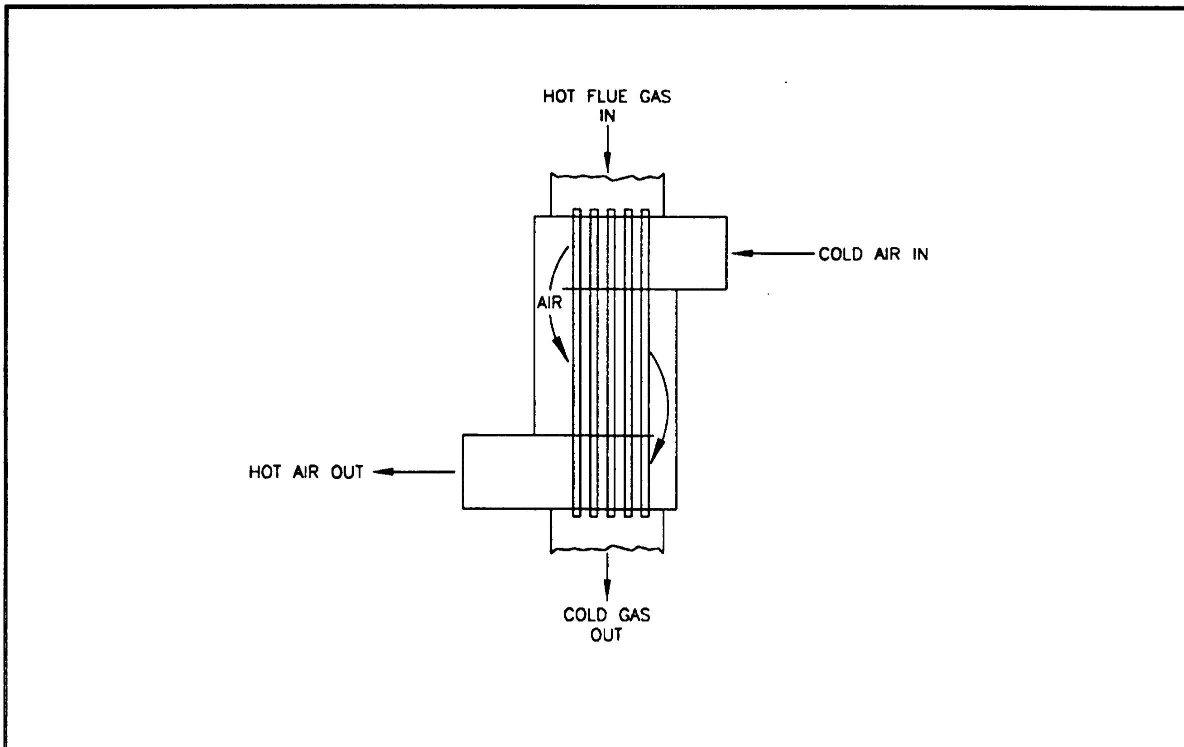

Figure 1. Tubular recuperative air heater, three air pass - counter flow unit.

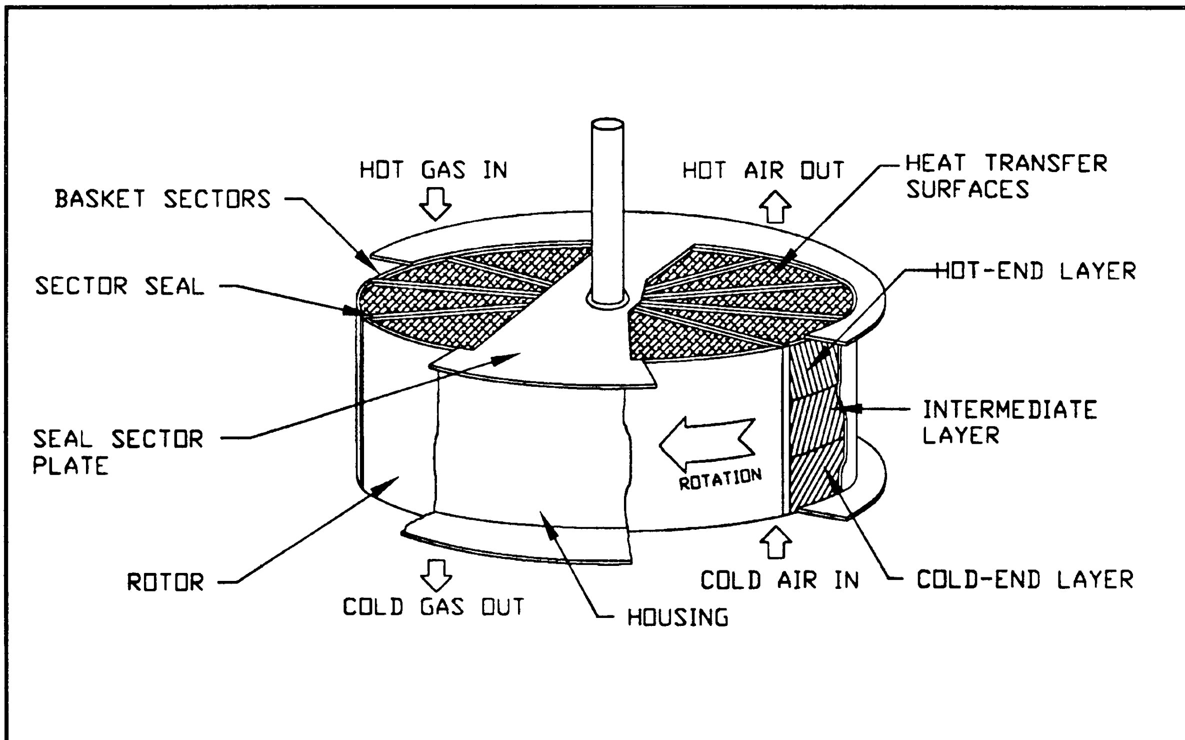

Figure 2. Rotating basket (Ljungstrom® type) regenerative air heater.

the air as the hot plates rotate through the air side. The design is compact and provides efficient heat transfer.

Although there are seals around the rotor, leakage from the combustion air side of the exchanger into the flue gas side is perhaps the biggest problem with the design. The leakage occurs in three areas: across the radial seals, in the clearance between the rotor and the metal case, and by entrainment from the basket gas passages as the baskets rotate from the air side into the flue gas side. When new, the air leakage may be as low as 5 percent to 10 percent of the incoming flue gas flow, depending on air heater size and air-to-gas pressure differentials. As the seals wear and the air heater ages,
the leakages often increase. If periodic maintenance and replacement of parts are not performed, leakage can increase substantially. The air leakage increases the forced draft fan power consumption since the leaked air bypasses the combustion step and more combustion air must be supplied. The induced draft fan power also increases since the flue gas flow out of the air heater increases by the amount of air leakage. Additionally, the air leakage reduces plant thermal efficiency since less heat is transferred to the combustion air, and increases maintenance on the air heater due to the need to replace or adjust worn seals.

2.2 General Description Heat Pipe Air Heaters

The heat pipe is a new heat exchanger design which can be used for utility air heaters. The heat pipe design has the potential to eliminate many of the problems associated with the tubular and Ljungstrom air heater designs and to operate at somewhat lower flue gas outlet temperatures which would improve overall plant heat rates. Heat pipe air heaters operate as regenerative exchangers in which heat from the hot flue gases is indirectly transferred to the cold air by means of a working fluid. The operation is illustrated in Figure 3. The heat pipe tubes are partially filled with a heat transfer working fluid. The heat pipe tube is sealed under high vacuum to insure that the only gas inside the tube is the working fluid vapor. Passing hot flue gases over the lower end of the tube causes the working fluid to boil and the vapors to flow to the cold end of the tube. Cold air flowing over the top of the tube condenses the vapors releasing latent heat which heats the air. Since the heat pipes are mounted at a slight angle from horizontal (five degrees for the Milliken units), the condensed liquid flows back by gravity to the evaporator end of the pipe to repeat the cycle. Wall grooves or wicks are sometimes used inside the heat pipe tubes to improve wall wetting and heat transfer.

Inside a heat pipe, heat is transferred by boiling and condensing heat transfer mechanisms. For these mechanisms, heat transfer can proceed at extremely high rates as compared to conduction and/or convection. Because of this, a heat pipe can transfer several thousand times the amount of heat energy as solid copper for a given temperature difference. Due to the high internal heat transfer rates, individual heat pipes operate essentially isothermally with very small temperature differences between the hot and cold ends. This aids in achieving uniform outlet temperatures for heated and cooled process streams.

Depending upon the application, many different materials can be used as working fluids including: liquefied gases, water, hydrocarbons, chlorofluorocarbons, and liquid metals. The working fluid must be operated below its critical temperature, must be compatible with the tube wall material, and must be stable and not decompose under operating conditions. For the Milliken air heater design, naphthalene was selected for the high temperature sections and toluene used in the intermediate and cold-end sections.

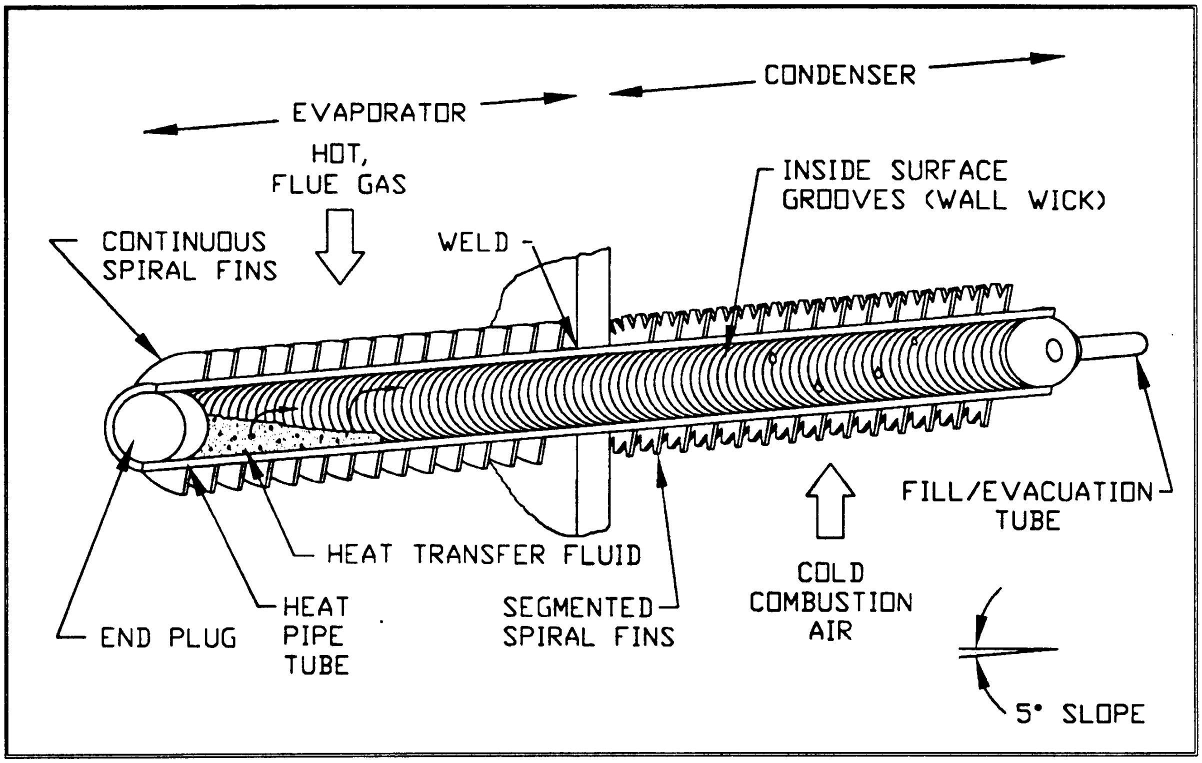

Figure 3. Heat pipe cross section.

A full-scale heat pipe air heater consists basically of two ducts with a common wall. Individual heat pipe tubes extend through the common wall across both ducts (Figure 4). Hot flue gases flow through one duct while cold combustion air flows through the other duct. The tubes are usually seal welded or gasketed in some fashion at the common wall to prevent air leakage between the flue gas and air sections. The ends of the tubes are free to expand or contract as necessary within the duct casing. By extending the individual tube surface through the use of fins, compact units can be designed.

The main advantages of the heat pipe air heater design over tubular designs are: compactness, a lower potential for air leak development, and uniform temperature distribution across the heat transfer zone. The common wall divider between the flue gas and air sections is made of a thick, heavy metal plate which is unlikely to corrode through over the life of the unit. Additionally, each heat pipe tube provides a double barrier against air leakage. Should a tube become penetrated on the flue gas side due to corrosion, the tube would loose the charge of working fluid and become inactive. Air would not however flow into the flue gas section unless the air end of the tube also became punctured, an unlikely event. Finally, because each heat pipe operates isothermally along its length, the outlet temperatures for both the heated and cooled streams can be controlled more exactly and uniformly. This potentially could benefit utility air heaters by eliminating flue gas side cold spot areas and allowing operation at lower outlet temperatures due to tighter control of the cold-end heat transfer surface metal temperatures.

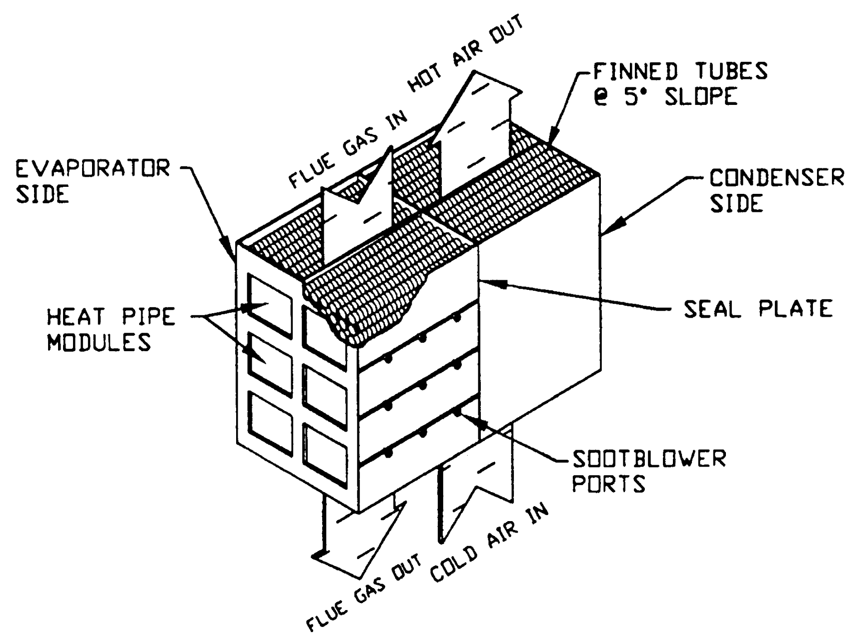

Figure 4. Typical heat pipe air heater construction.

When compared with the Ljungstrom type regenerative air heaters, the heat pipe air heater design provides the following actual and potential advantages:

1. Zero air leakage – reduces PA, FD, and ID fan power and for boilers with wet FGD systems, reagent slurry pumping power requirements.
2. Lower capital costs in new plants for downstream equipment (fans, particulate collectors, scrubbers, and stacks) as a consequence of zero air leakage.
3. Better air heater outlet temperature control allowing for more heat recovery and lower flue gas outlet temperatures.
4. Passive design with no moving parts to operate or maintain.
5. Lower erosion due to low flue gas velocities.
6. Lower flue gas and air side pressure drops which reduce the fan power.
7. Potential for improved ESP operation due to more uniform inlet flue gas exit temperature profile.
8. Improved flue gas flow balance control for boilers with multiple air heaters due to elimination of air leakage.

3.0 FULL-SCALE HEAT PIPE DESIGN

Milliken Unit 2 was originally constructed in 1958. The boiler is a reheat, tangentially-fired pulverized bituminous coal unit designed by Combustion Engineering. It has been upward rerated to a design point of 150 MWnet. Originally, Unit 2 was equipped with two vertical flow Ljungstrom® air heaters. As part of the Milliken CCT-IV demonstration program, the Unit 2 Ljungstrom units were replaced with two vertical flow heat pipes to help overcome some of the boiler heat rate decline expected with the concomitant installation of low NO_x burners and an FGD system.

3.1 Mechanical Design

The design of the individual heat pipe air heaters is summarized in Table 1. The general construction is schematically shown in Figure 5 for the Unit 2A heat pipe. As indicated by the insert in Figure 5, the Unit 2B heat pipe is constructed as a mirror image of Unit 2A. Each heat pipe contains 12 (three horizontal - four vertical) shop-fabricated heat transfer modules which are field assembled. The modules are 100 percent seal welded to eliminate air leakage into the flue gas from the ambient environment or across the division wall between the air and flue gas sections. The box-shaped modules sit on duct transition sections which tilt the tubes five degrees above horizontal.

The heat pipe tubes are 2" outside diameter (OD), 0.095" wall, and approximately 35.5 feet long. To achieve maximum heat transfer and compactness of design, the tubes are finned on both the flue gas and air sides and the tube rows are arranged on a 3.75" center-to-center staggered triangular pitch. On the flue gas side, 3/4" high continuous spiral fins (three per inch) are used. On the air side, 3/4" high segmented fins (seven per inch) are used. The fins are attached to the tubes by a high frequency resistance welding process. The tube and fin materials are carbon steel (CS) in areas where operating temperatures are above 300°F, and a low-alloy corrosion resistant (LACR) material (CorTen A) in areas below 300°F. Some T11, a low-carbon, low-alloy (1-1/4 chrome -1/2 Moly), CS tube material is used in the highest operating temperature areas to reduce the potential of working fluid breakdown. The heat pipe casing is ASTM A36 mild CS since all flue gas side parts are expected to be at temperatures above the acid dew point.

The heat pipe tubes are fixed only at the division wall. This allows the tubes to expand or contract as necessary. On the air side, the tubes expand within the exchanger case since the tubes are hotter than the combustion air being heated. On the flue gas side, the tubes contract within the case since the tubes are colder than the flue gas being cooled. A tube sheet is used to support the tube ends on the air side; while on the flue gas side, the lower tube ends are supported by short cylinder sleeves that are welded to the module walls.

The flue gas section casing is designed for -35 in. WC pressure. The design pressures for the primary air and secondary air sections are +60 in. WC and +35 in. WC, respectively.

Table 1

Heat Pipe Air Heater Design Summary

New York State Gas and Electric Milliken Station -- Unit 2

Manufacturer	ABB Air Preheater Inc.	
Model	303.8-408-36-DV	
Number of Heat Pipe Air Heaters	2	
Number of Tube Modules/Air Heater	3 Horizontal/4 Vertical	
Module Slope	5°	
<u>Tubes</u>		
Number	2,880/ Heat Pipe	
Diameter	2.0 in.	
Wall	0.095 in.	
Pitch (triangular)	3.75 in.	
Material	**Primary Sections**	**Secondary Sections**
	7 rows T11 CS	1 row T11 CS
	21 rows A-178A CS	27 rows A-178A CS
	8 rows A-618 CorTen A	8 rows A-618 CorTen A
Working Fluid	21 rows Naphthalene	14 rows Naphthalene
	15 rows Toluene	22 rows Toluene
<u>Fins</u>	**Flue Gas Side**	**Air Side**
Type	Continuous Spiral	Segmented Spiral
Attachment	Welded	Welded
Height	0.75 in.	0.75 in.
Thickness	0.059 in.	0.036 in.
Density	3 per in.	7 per in.
Material	28 rows A-178A CS	28 rows A-178A CS
	8 rows A-618 CorTen A	8 rows A-618 CorTen A

Design Performance (ea.)	**Flue Gas Side (Combined)**	**Air Side**	
		Primary	**Secondary**
Inlet Flow	750,000 lb/hr	62,500 lb/hr	562,500 lb/hr
Inlet Temperature	680 °F	80 °F	80 °F
Outlet Temperature	253 °F	650°F	617 °F
Specific Heat	0.260 Btu/lb-°F	0.247 Btu/lb-°F	0.247 Btu/lb-°F
Duty	83.3 MM Btu/hr	8.8 MM Btu/hr	74.5 MM Btu/hr
Minimum Cold Tube Temperature		221 °F	170 °F
Guaranteed Pressure Drop	3.65 in. WC	3.60 in. WC	5.35 in. WC

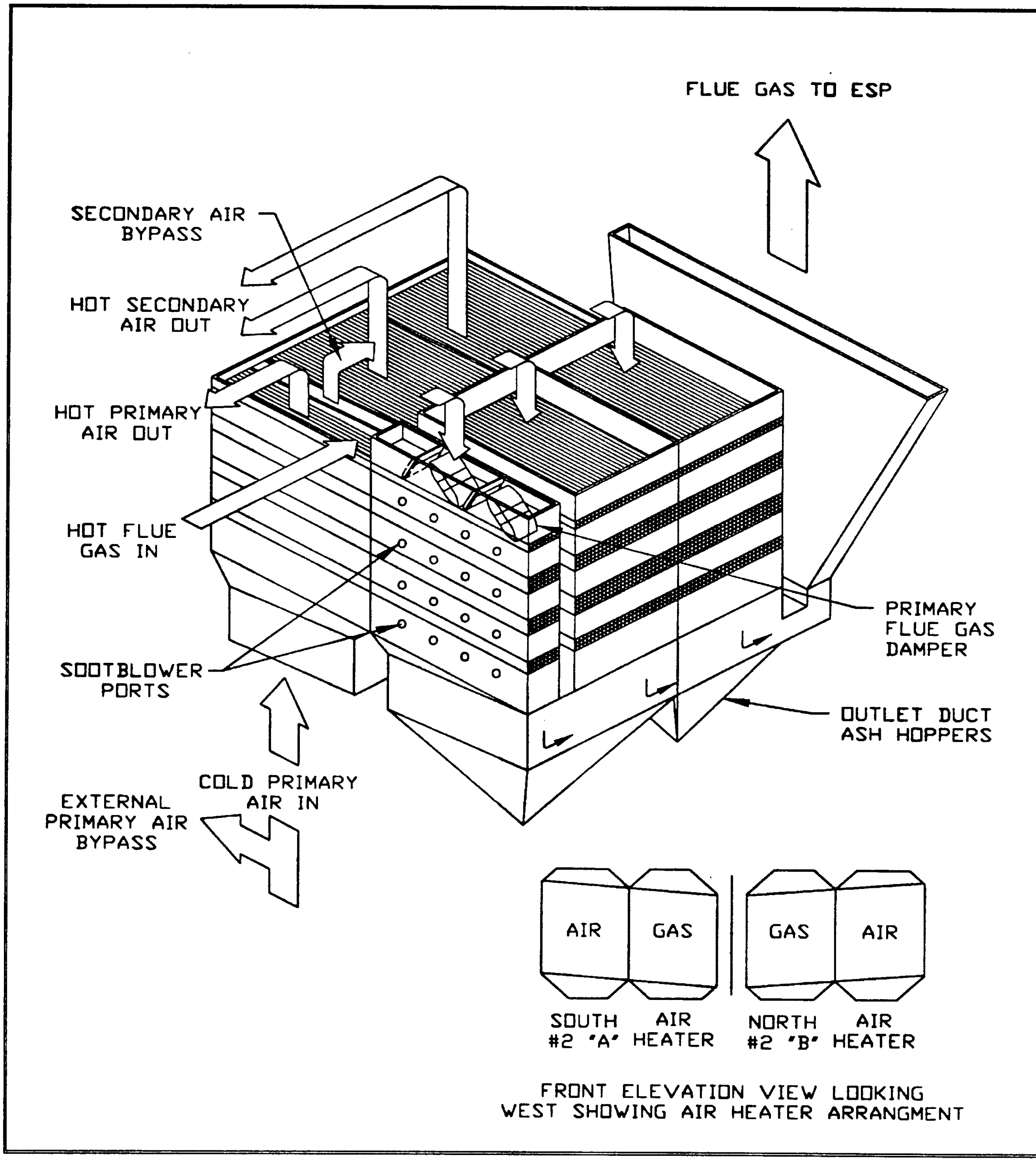

Figure 5. Construction of Milliken heat pipe air heaters.

Each heat pipe exchanger weighs approximately 960,000 lbs. The exchangers are each supported on eight legs (not shown in Figure 5) and are each anchored by one leg to the floor. The other seven support legs distribute the load and are free to move on sliding plate foot bearings. Four of the support legs are on guided foot bearings while the other three are on unguided free foot bearings. This accommodates lateral movement due to thermal expansion or mechanical stress. The tops of the heat exchangers are free to move and expand both vertically and horizontally. This movement is accommodated through the use of expansion joints on all ducts attaching to the air heaters.

Each heat pipe exchanger is designed to heat both primary and secondary air streams in separate sections. This provides added flexibility for coal drying and in achieving maximum heat recoveries. Bypasses are provided for both the primary and secondary air streams. The primary air bypass is external to the heat pipe and is used to supply tempering air at the coal mills. The secondary air bypass is internal and an integral part of the heat pipe as shown in Figures 5 and 6. An electrically driven damper inside the heat pipe (Figure 6) is used to control the flow through the secondary air bypass. This bypass is used primarily to limit heat transfer from the flue gas section to avoid low cold-end tube temperatures which result in acid condensation. Under certain conditions, bypass control must be limited to prevent overheating of the hottest operating toluene filled tubes.

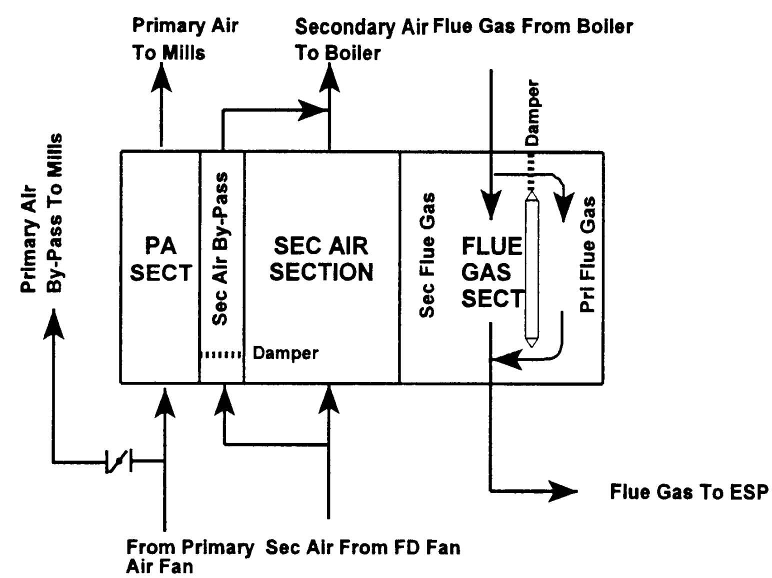

Figure 6. Heat pipe process flow streams.

Two ducts from the boiler supply hot flue gases to the heat pipe air heaters, one duct to each air heater. The flue gases approach the air heaters through horizontal ducts. A set of ladder vanes inside the ductwork hood, which is mounted on top of each heat pipe, redirects the flue gas flow vertically downward to the heat pipe tube banks. The tube banks are split into one primary and two secondary flue gas sections. Gas distributes to the sections based on the pressure drop through the sections. When the air heaters were originally installed, the primary flue gas sections did not have inlet dampers. NYSEG later installed louvered dampers to provide better temperature control and reduce the potential of overheating the hottest toluene filled tubes. Closing the dampers reduces the flow through the primary flue gas section and increases the flow through the secondary flue gas sections. However, primary flue gas flow adjustments have little effect on the overall secondary section flows since the primary gas flow is normally only about one-eighth the secondary flow.

On-line cleaning of the heat pipes is accomplished using sootblowers supplied with 150 psi air. There are 32 sootblowers, 16 on each air heater, which are located in lanes between the tube banks (Figure 5). The bottom three tube banks can be sootblown from both the top and bottom sides. There is provision to sootblow only the bottom of the top tube bank since the fly ash is dry at this location and little fouling is expected. Because of the large amount of sootblowing air required, a new air compressor was purchased and integrated into the existing plant air system. The new system uses a 3,000 acfm, 1,200 hp, inter-cooled, three stage Ingersoll Rand centrifugal compressor.

The sootblowers are partially retractable Bergemann units. The sootblowers have variable frequency gear motor drives which allow slower or faster blowing times depending upon the fouling conditions. When activated, the sootblower lances rotate in a helical fashion into the heat pipe to clean tube banks above and below the lance. Because of site access constraints, the sootblowers are equipped with ½ long retractable lance tubes. When fully retracted, the lance tubes extend half way across the heat pipe tube banks. The lances are equipped with two venturi nozzles at the tip end and two nozzles at the center. The two nozzles at each location are located 180° apart on the lance circumference. This design allows complete tube bank cleaning with a lance travel of one-half the cross section distance.

As shown in Figure 5, there are solids collection hoppers directly under the flue gas side tube banks. The hoppers collect fly ash and sootblowing deposits which drop from the downward flowing flue gases as the gases change direction and flow to the outlet duct. Periodically, the hoppers are pneumatically emptied using the existing boiler/ESP pressurized-ash conveying system.

3.2 Temperature Measurement

Temperature measurement is key to analyzing performance and controlling the heat pipe air heaters. On-line measurements, which are tied into the plant's computer data logging system, fall into three categories: (1) thermocouple (TC) arrays for measuring flue gas and air temperatures to and from the heat pipes, (2) internal temperatures of critical tubes, and (3) tube skin and flue gas temperatures in the coldest tube row of the cold-end module. Because of high emf and linearity, chromel/constantan Type E thermocouples are used throughout. The flue gas and air TC arrays are located in the ductwork close to the heat pipes. All array TCs are contained within thermowells. The TC arrays provide information for calculating average temperatures and allow analysis of thermal performance based on changes in temperature gradient spreads. The TC arrays around each heat pipe are listed in Table 2.

Each air heater is supplied with ten heat pipe tubes fitted with TCs within thermowells to measure temperatures in critical areas. Type E dual element (one active element plus spare) TCs are used. The thermowells are welded into evaporator end, end caps. This helps to insure accurate measurement of the tube operating temperature since the thermowells are surrounded by boiling liquid. Condenser end TCs are not used since inaccurate results would be obtained if non- condensable gases begin to buildup inside the heat pipes due to contamination or breakdown of the working fluids. The locations of heat pipes with thermowell TCs are indicated in Figure 7. There are three TCs in the hottest row of toluene filled tubes (Module C3 of the primary air heating section). To prevent working fluid breakdown, the temperature of the toluene tubes must be limited to a maximum 550°F. Plant operators, therefore, monitor these TCs to guide adjustment of the primary air rate through the air heater or the primary flue gas damper position. The other TCs are in areas where tube corrosion is a concern. Three TCs are located in the coldest CS heat pipes (last row of Modules C1, C2, and C3) and four TCs are located in the cold-end heat pipes (last row of Modules D1, D2, and D3).

Table 2

Milliken Heat Pipe Flue Gas and Air Stream TC Arrays

Stream	TC/Array
Flue Gas In	10
Flue Gas Out	6
Secondary Air Out	6
Secondary Air In	4
Primary Air In	2
Primary Air Out	2

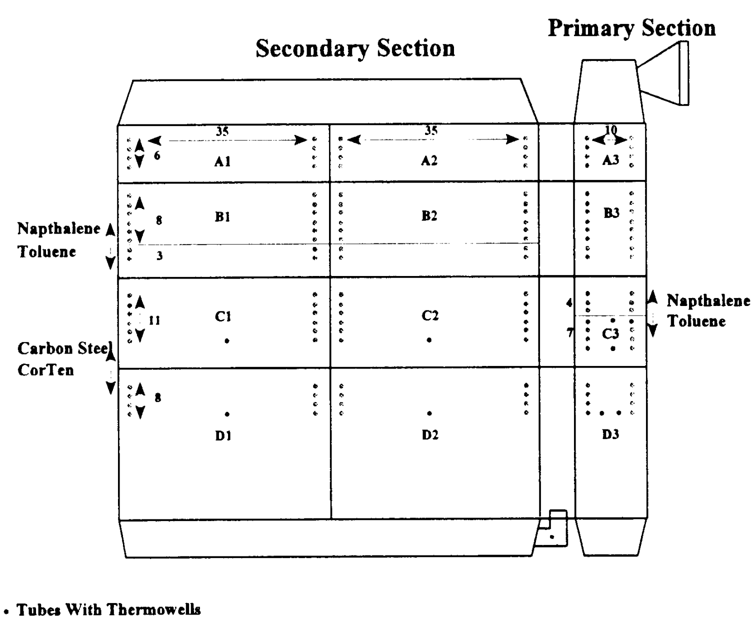

Figure 7. Tube layout and thermowell locations for the Milliken heat pipe air heaters.

After the air heaters were in operation, additional TC instrumentation was installed in the outlet cold-end modules to help monitor fouling behavior. Six heat pipe skin TCs were installed in each heat pipe air heater (Figure 8). This was done by grinding the fins off a small area of a tube and then gluing a sheathed TC to the tube using a heat transfer cement. For comparison, four flue gas TCs were located near the skin TCs; two gas TCs in the primary flue gas section and two gas TCs in the secondary flue gas section. Monitoring of these TCs provided an on-line means of estimating where fouling was most severe in each heat pipe.

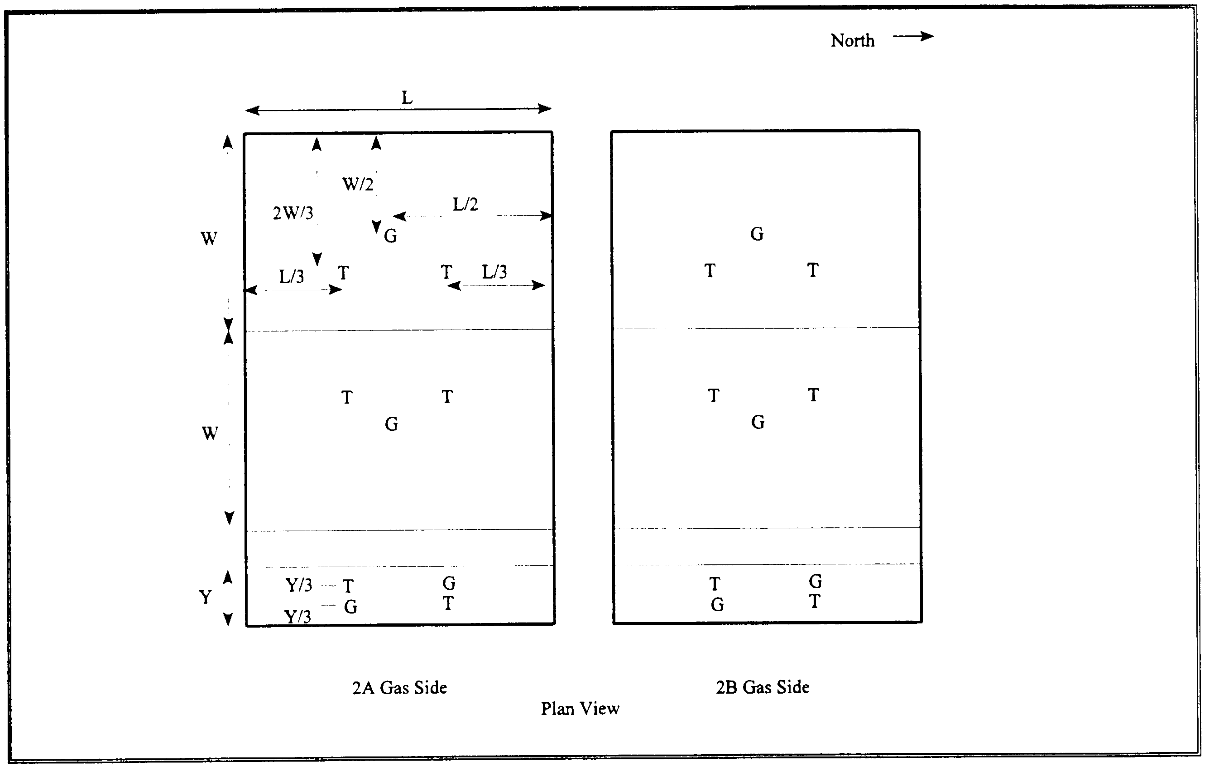

Figure 8. Cold-end tube (skin) and flue gas TC locations.

3.3 Staggered Tube Design

In 1992, when inclusion of a heat pipe was first proposed for the Milliken Unit 2 boiler, in-line tube arrangements were the common practice for finned tube air heaters. The in-line tube arrangement had proved to be a successful design which did not have problems with fly ash plugging or erosion. According to ABB API conventional sootblowing techniques were generally adequate to remove the slight plugging which occurred between tube rows. Staggered, finned tube air heaters were normally not specified for coal fired boilers because of concern for potential increased plugging and fouling of tubes and fins and expected difficulties in cleaning. However, there was great interest in developing a successful staggered design. For the same heat transfer requirements, a staggered tube arrangement can result in a cheaper more compact design with fewer tubes due to the increased heat transfer afforded by the tortuous flow path. For the Milliken system design, a 35% reduction in the number of heat pipe tubes was expected for the all staggered tube design over an all in-line tube design.

As part of a design effort, ABB API installed a pilot heat pipe air heater at Milliken on a flue gas slipstream. The pilot heat pipe had removable tubes which could be rearranged with in-line or staggered tube pitches. Between October 6 to 27, 1993, ABB API ran tests on the slipstream unit.

The results indicated that a staggered, spiral finned design was practical and could be operated on a pulverized coal boiler without plugging. The testing also indicated that conventional sootblowing would be effective in cleaning the tubes and fins.

After the ABB API test program was completed, NYSEG and CONSOL conducted separate tests to further establish operability of the staggered tube design in a boiler flue gas environment with and without ammonia slip from a NO_x removal process (Appendix A). As part of this testing, parametric performance and long term operability tests were conducted without ammonia slip between October 27 and December 13, 1993. Cold-end tube temperatures were controlled at nominally 170°F to simulate the operation of the commercial air heater with a 250°F flue gas outlet temperature. This insured that the cold-end tube metal temperatures were well below the flue gas acid dew point. The testing showed no fouling of the hot-end tube module and minor fouling of the cold-end tube module. The testing indicated that total flue gas side pressure drops might be expected to increase about 2.2 times the base drop over a six-month period. This was thought to be acceptable since the original pressure drop could be recovered by scheduled air heater washing every six months. The decision was made to install an all staggered tube design. As will be explained in a later section, all tube bank modules but the cold-end module proved to be readily cleanable by sootblowing. Cold-end fouling remains however a major operating problem for the current heat pipe design.

3.4 Materials Selection

Heat pipe material selection was based on the results of corrosion test programs conducted at the EPRI Environmental Control Technology Center (ECTC) and at the NYSEG Milliken Station. The tests were primarily directed at evaluating construction materials for the cold-end modules of the commercial-scale heat pipe air heaters. In the cold-end modules, temperatures drop low enough for the small amount of SO_3 contained in the flue gases to react with water vapor and begin condensing as sulfuric acid. This can lead to severe fouling as fly ash/acid poultices form and acid attacks the heat transfer surfaces. Since testing of the selective non-catalytic reduction (SNCR) NO_x removal process was originally proposed in the Milliken CCT-IV program, there were additional concerns for fouling and corrosion caused by the ammonia loss or slip from such processing. Leftover ammonia can lead to ammonium sulfate/bisulfate condensation in the air heater at temperatures higher than the SO_3 acid dew point. To address these issues, corrosion tests were initially conducted at the ECTC, and then later at the Milliken Station.

The ECTC test facilities include a selective catalytic reduction (SCR) NO_x removal pilot plant which has a small heat pipe heat exchanger. The exchanger is used to heat the 120°F SCR reactor feed gas (flue gas from the Kintigh Station FGD) using the 590°F reactor exit flue gas (Figure 9). This exchanger provided an ideal location for testing candidate materials of construction in a fly ash-free environment with or without ammonia present. As shown in Figure 9, test heat pipes made from CS, Cor-Ten® B, and AL-6XN® were installed (stacked vertically) at two locations. Since the cold-end module of the ECTC heat pipe contained 2,205 duplex stainless tubes with 409 stainless fins, information on these materials was also obtained. At the outlet of module 1 on the return side, the test heat pipes were exposed to the coldest cold-end flue gases with the highest potential for acid deposition. These heat pipes operated with metal surface temperatures between 150°F and 210°F which are well below the normal sulfuric acid dew point of 270°F. At the inlet to module 1 on the return side, the test heat pipes were in an area where ammonia sulfate/bisulfate fouling was expected based on the previous operating history of the exchanger. Here the heat pipes operated with metal surface temperatures between 260°F and 290°F.

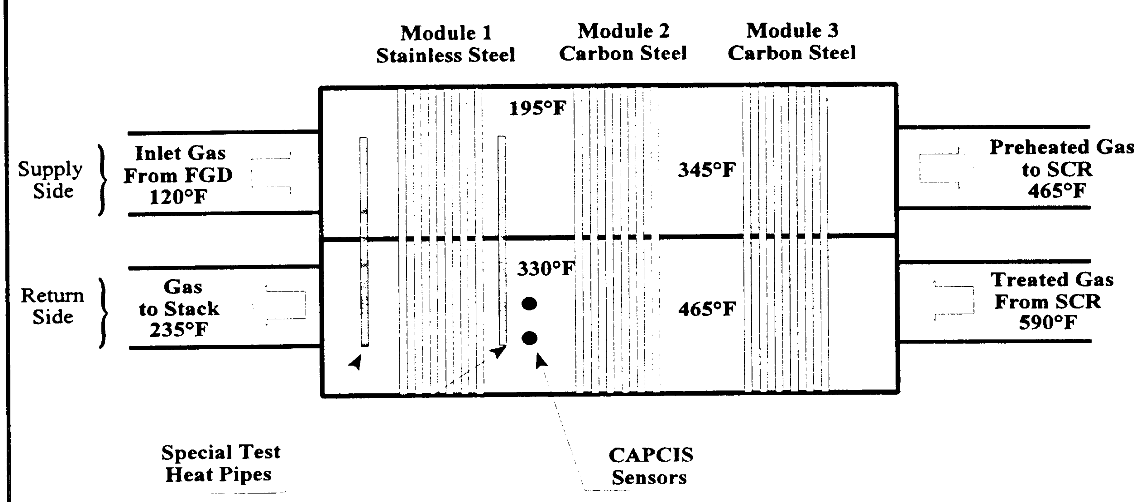

Figure 9. Plan view of ECTC heat pipe heat exchanger.

Although corrosion of the heat pipes was experienced on both the supply and return sides, only the return side corrosion will be discussed. This is the only area where the corrosion results are relevant to the design of the Milliken air heaters.

To monitor corrosion rates during individual tests, two electrochemical CAPCIS corrosion probes were installed on the return side between modules 1 and 2. One probe was made of SA-178A CS and the other from Cor-Ten B. The probe instrumentation incorporates the use of electrochemical impedance measurement (EIM), electrochemical potential noise (EPN), electrochemical current noise (ECN), and zero resistance ammetry (ZRA). Changes in these electrical responses are used to determine the corrosion rate in real time and can be used to determine the type of attack (uniform or localized). A description of the measurement techniques can be found in Reference 1.[1]

The ECTC test program consisted of operating the SCR heat pipe exchanger at design flow and temperature conditions with differing amounts of ammonia in the return side flue gas feed. To insure constant ammonia slip conditions, the SCR was operated at zero ammonia slip with the required amount of ammonia injected separately downstream of the SCR reactor but ahead of the heat pipe inlet. Three test conditions were established, zero ammonia slip, 1-2 ppm ammonia slip, and 4-5 ppm ammonia slip. After each test period, the unit was shut down for inspection and cleaning of the

the heat pipe exchanger. The test heat pipes were installed in the ECTC heat exchanger in November 1992 and removed in May 1993. The total operating exposure to a flue gas environment was 3,310 hours.[2] After removal, the test heat pipes and one original heat pipe from the heat exchanger were destructively tested by ABB API. The detailed results of the destructive testing can be found in Appendix B.

The general conclusion based on the destructive testing analysis was that none of the tested or original tube materials could provide a 20-year life for a cold-end tube bank for the ECTC heat pipe operating conditions and a standard tube wall thickness of 0.100". The AL-6XN was unsuitable since the material exhibited a marginal corrosion rate (5.3 mils/yr max.) at the location between modules 1 and 2, and localized pitting and cracking at the outlet of module 1 on the return side. At the module 1 outlet on the return side, the 409 SS fin material, CS, and Cor-Ten B all exhibited high corrosion rates of up to 17.5 mils/yr for the 409 SS, 42 mils/yr for the Cor-Ten B, and 77 mils/yr for the CS. The Cor-Ten B corrosion was relatively uniform as opposed to groove patterns associated with the corrosion of the CS. The groove patterns appeared to be due to liquid collecting on the tube surface with subsequent transport on and around the heat pipe. At the return side outlet of module 1, 2205 SS showed the lowest corrosion rates. However, the corrosion appeared to be flow related with the leading edge of the tubes showing more corrosion than the trailing edge, and there was evidence of anodic protection of the 2205 SS by the 409 SS fin material. For all materials, corrosion rates were lowest at the location between modules 1 and 2.

For the on-line CAPCIS corrosion probes, an internal air purge is used to control the sensing element temperature at a set temperature below the flowing flue gas temperature. This feature allows the determination of specific conditions where the rate of corrosion becomes problematic. During the ECTC test program, corrosion rates were measured for SA-178A CS and Cor-Ten B over a temperature range of about 100°F to 230°F[3]. The data show a variation in corrosion rate with temperature. At temperatures below the water dew point (~120°F for the flue gas from the FGD), both materials show high rates of corrosion. From this point, the rates initially decline with increasing temperature to a minimum, then increase with increasing temperature to a second maximum, and finally decline again. For the CS probe, the second maximum occurred at about 160°F regardless of the ammonia slip level while for the Cor-Ten B material, the second maximum appeared to shift to higher temperatures with increasing ammonia slip. At zero ammonia slip, CS corroded more rapidly than the Cor-Ten B. The presence of ammonia in the flue gases appears to reduce the corrosion rate for the CS (particularly at five ppm level) but increases the rate for the Cor-Ten B material.

The ECTC heat pipe environment is believed to be a worst-case test environment due to the high flue gas moisture (saturated with water at FGD outlet conditions) and the lack of any fly ash. High moisture levels increase the temperature at which SO_3 begins to condense and allows a more dilute, more corrosive acid to form. Operating without fly ash present, results in the tube and fin metal surfaces being the only sites on which condensed acid can collect and react. With fly ash present, some of the acid would be sequestered by absorption on the ash or neutralized by alkalinity in the ash.

Based on the above, a decision was made to continue material selection testing at the Milliken Station where tests could be conducted in a fly ash containing flue gas environment. Materials to be tested were: SA-178A CS, Cor-Ten A, and 2205 duplex SS. These materials were selected because of cost and availability advantages for CS, the well known greater corrosion resistance of Cor-Ten A over Cor-Ten B, and the superior performance shown by 2205 SS in the ECTC tests. For the Milliken tests, the Cor-Ten B CAPCIS corrosion probe was refurbished with Cor-Ten A sensing elements and Consol R&D fabricated three "simple" air-cooled corrosion probes made from 2205 SS (one probe) and Cor-Ten A (two probes). The simple corrosion probes were designed to simulate the operation of a heat pipe by maintaining the corrosion coupon metal temperature constant. This was accomplished using internal air purges. The simple probes had no electronic method for determining corrosion rates; rather the corrosion rates were determined by manual measurement of the probe outside diameters after exposure.

The Milliken corrosion testing was done in three stages. First, while the CAPCIS Cor-Ten B probe was being refurbished, the SA-178A CS CAPCIS probe was installed at the outlet of the Milliken Unit 2 ESP. In this location, the probe was exposed to a conventional flue gas environment but again without fly ash present. Over extended time periods, the probe was operated with sensing element temperatures of either 168°F (1,609 hours) or 231°F (1,501 hours). For the Milliken ESP outlet conditions, the electronically indicated corrosion rates were approximately 2 mils/year. This was confirmed by manual dimensional measurements which indicated somewhat lower average rates. These results indicated that CS was suitable for the ductwork and equipment downstream of the proposed air heater.

In the second stage of testing, the simple air-cooled corrosion probes were installed in the Unit 2 ESP inlet ductwork. At this location, the probes were exposed to a normal flue gas environment with fly ash present. The tests showed low corrosion rates (typically <3 mils/yr) for Cor-Ten A regardless of the average targeted operating skin temperature (i.e., 172°F, 192°F, or 202°F). However, the 2205 SS simple air-cooled probe showed severe pitting under fly ash scale buildups after only 832 hours of service at 170°F surface temperature. This resulted in the 2205 SS being eliminated from further consideration as a construction material.

The third stage of testing was to install the SA-178A CS and Cor-Ten A CAPCIS corrosion probes in the outlet duct of the ABB API slipstream heat pipe. As mentioned in Section 3.3, ABB API installed the slipstream heat pipe at Milliken to test the staggered tube design concept. After ABB API completed this testing, NYSEG and CONSOL R&D took over operation and installed a pilot SCR reactor ahead of the test heat pipe. This afforded heat pipe testing in a flue gas environment with fly ash and ammonia present. The test results indicated that the fly ash provided some protection against SO_3 and/or ammonium bisulfate (NH_4 HSO_4) attack. Overall corrosion rates for both metals were low, i.e., 2.9-3.5 mils/yr for CS at 176°F and < 2mils/yr for Cor-Ten A at 174°F. Corrosion rates did not appear to depend on the ammonia slip between one and 3.5 ppm slip. Based on these results and all the previous corrosion test work, the decision was made to use CS for heat pipes operating above 300°F skin temperature and Cor-Ten A for all heat pipes operating below 300°F skin temperature. The detailed test results for the Milliken corrosion program can be found in Appendix B.

3.5 Installation -- Equipment Layout

The main goal of the Unit 2 equipment design was to install SO_2 and NO_x control systems with minimum impact on the overall plant heat rate. Therefore, energy technologies such as the use of a heat pipe air heater were integrated into the plant design. The heat pipe was designed for a minimum 20°F decrease in the flue gas side air heater outlet temperature. This was expected to provide an approximate 0.5% improvement in heat rate. The no air leak feature of the heat pipe was expected to reduce air flows by about 16% and save approximately 337kW of fan power.

Because the Unit 2 air heater and coal mills were being replaced at the same time, there was an opportunity to reconsider the design of the primary air supply/coal mill circuit to further reduce power requirements. Two concepts were considered: the use of a single sector air heater coupled with four hot primary air fans (one fan to each mill), or, separation of the primary and secondary air heating sections and the use of two cold primary air fans (one supplying each air heater). The decision was made to install the cold primary air fan system since analysis of the concept indicated reduced construction and capital equipment costs, lower projected maintenance costs, and a 20 Btu/KWh power savings.

The overall process flow scheme is presented in Figures 10, 11, and 12. Figure 10 shows the flue gas loop with hot gas from the boiler economizer passing through the heat pipe. The hot flue gases heat the primary air and secondary air streams in separate compartments in the air heater. From the air heaters, the cooled flue gases then proceed on to the ESP particulate collectors, ID fans, FGD, and finally the stack.

Figure 11 shows the primary air circuit to the coal mills. High pressure primary air is supplied by a cold PA fan to the air heater. Heated primary air streams from the two air heaters combine in a common header which splits into four coal mill feed streams. Bypassed tempering air mixes with the heated air ahead of each mill. The flows of hot primary air and tempering air are blended as required by mill load and coal dryness.

The secondary air system is shown in Figure 12. A low pressure FD fan supplies the air to the secondary air heating section of the air heater. The heated air then flows to the boiler burners. Normally, all the required secondary air flow passes through the air heater. A bypass is provided to help control the flue gas outlet temperature. This prevents operating the cold-end heat pipe tubes at too low a temperature which would result in excessive fouling.

Figures 13 to 16 show the final equipment layout. Because the heat pipes were located under the precipitator, the old Ljungstrom air heaters were left in place. The inlet vertical ductwork to the Ljungstroms was disconnected and new horizontal ducts to the heat pipes installed as shown in Figure 13. There are ash hoppers under the flue gas sections of the air heaters to collect sootblown ash materials. The cooled flue gases leave the bottom west side of the air heater, travel vertically up to a crossover duct, which leads to another vertical flow duct to the precipitator entrance. Leaving the precipitator, the flue gases flow down to the ID fan at grade level.

A front, east side, view of the system is presented in Figure 14. The view shows the locations of the primary air fans, sootblowing air compressor, mill seal air fans and the 32 sootblowers.

Figure 15 is a northward facing view of the unit showing the primary and secondary air duct system. The view shows the PA and FD fans and the common motor. Placing the PA and FD fans directly below the heat pipes allows use of short expanding ducts between the fan discharges and the heat pipe connection flanges. An isometric view of the system is presented in Figure 16.

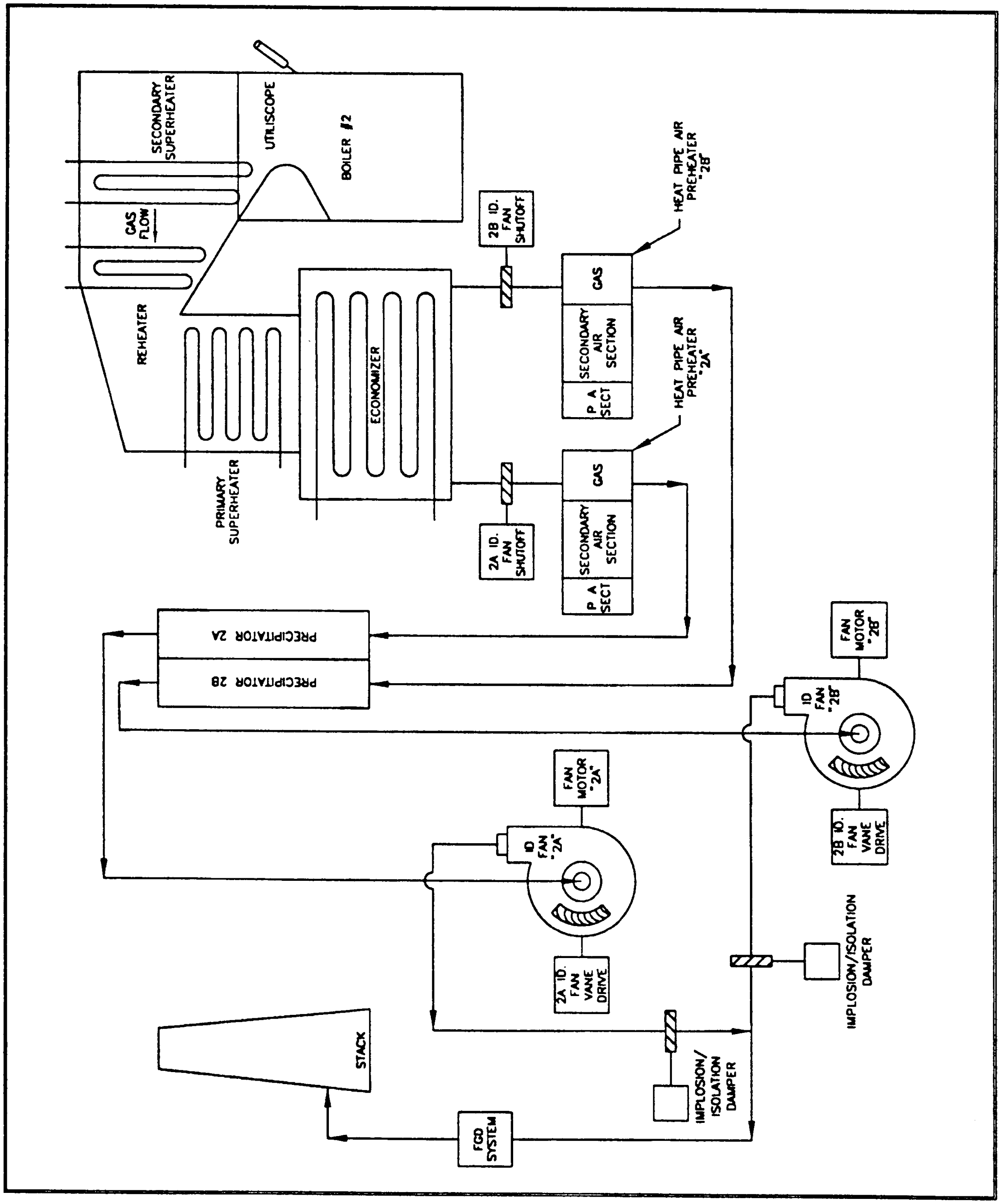

Figure 10. Unit 2 Induced draft system.

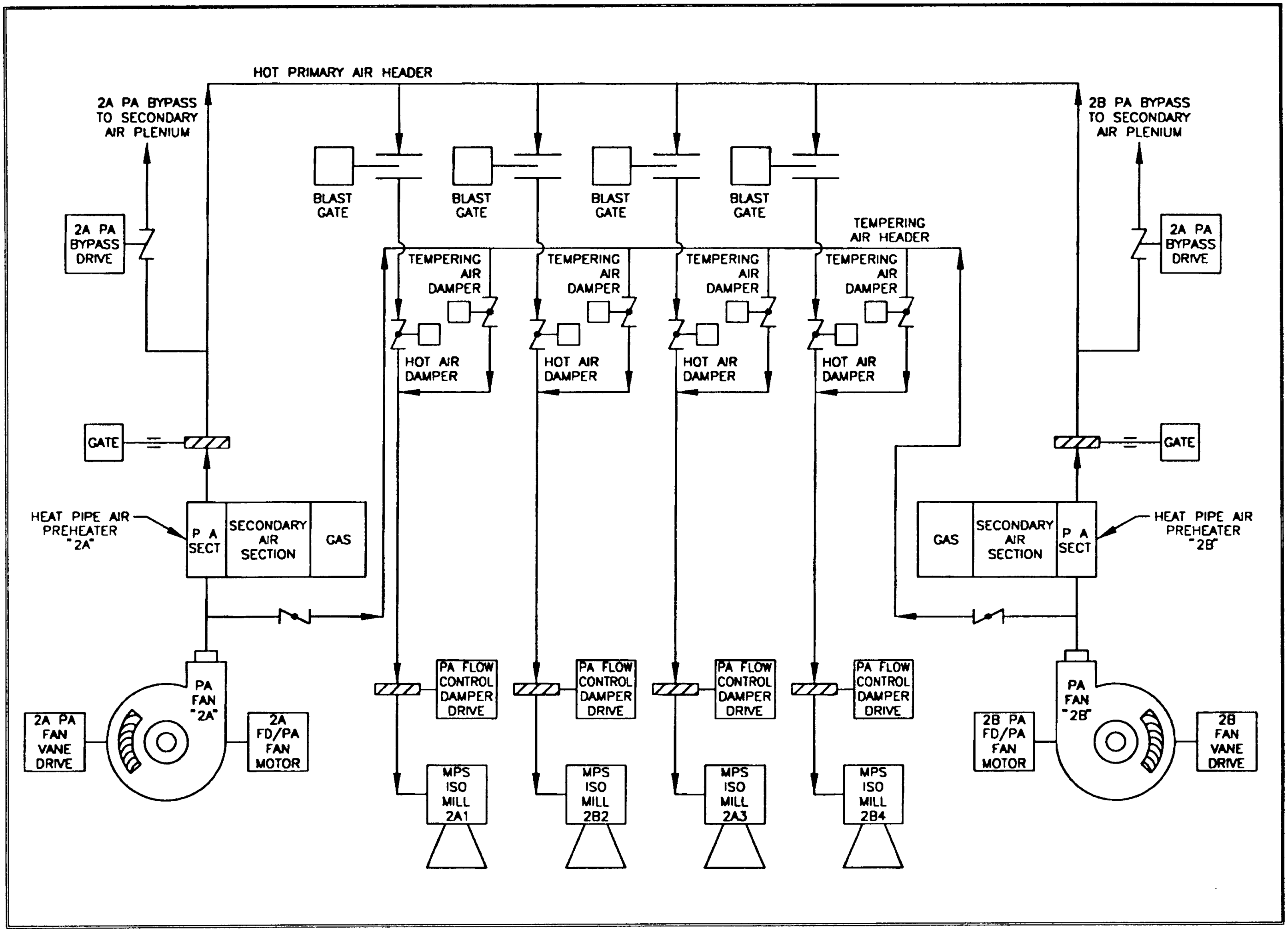

Figure 11. Unit 2 primary air system.

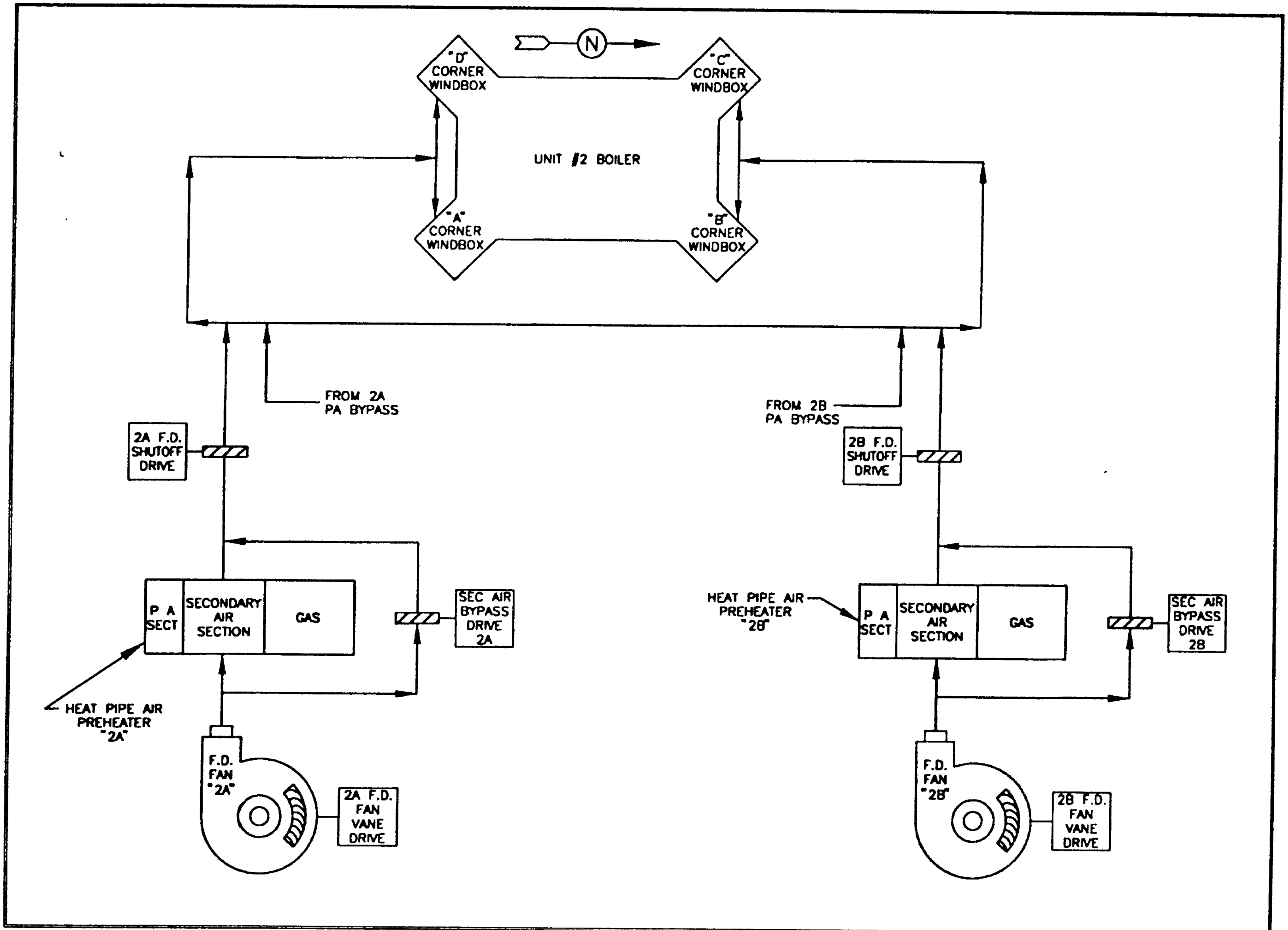

Figure 12. Unit 2 forced draft (secondary air) system.

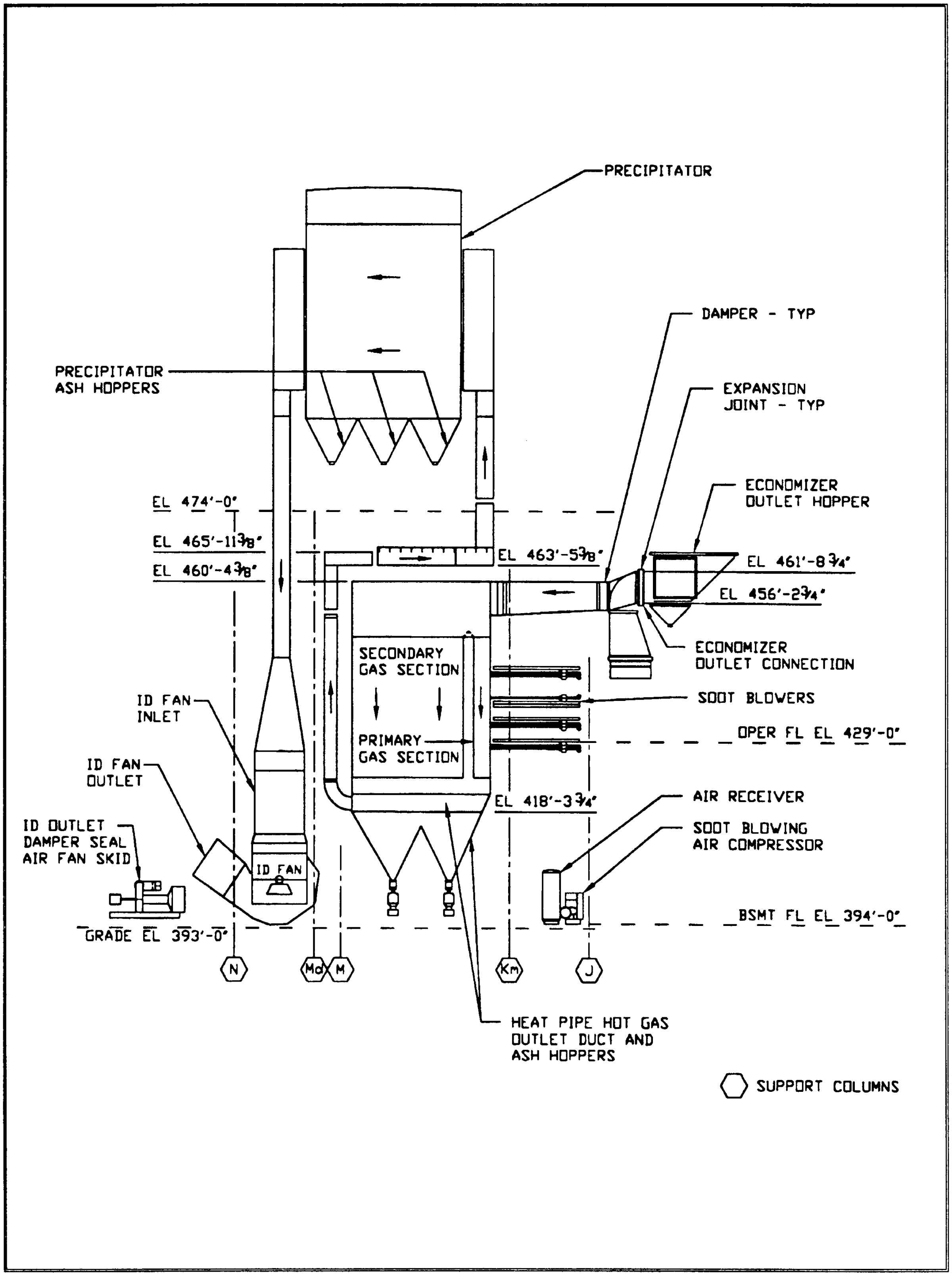

Figure 13. Heat pipe equipment layout -- flue gas section.

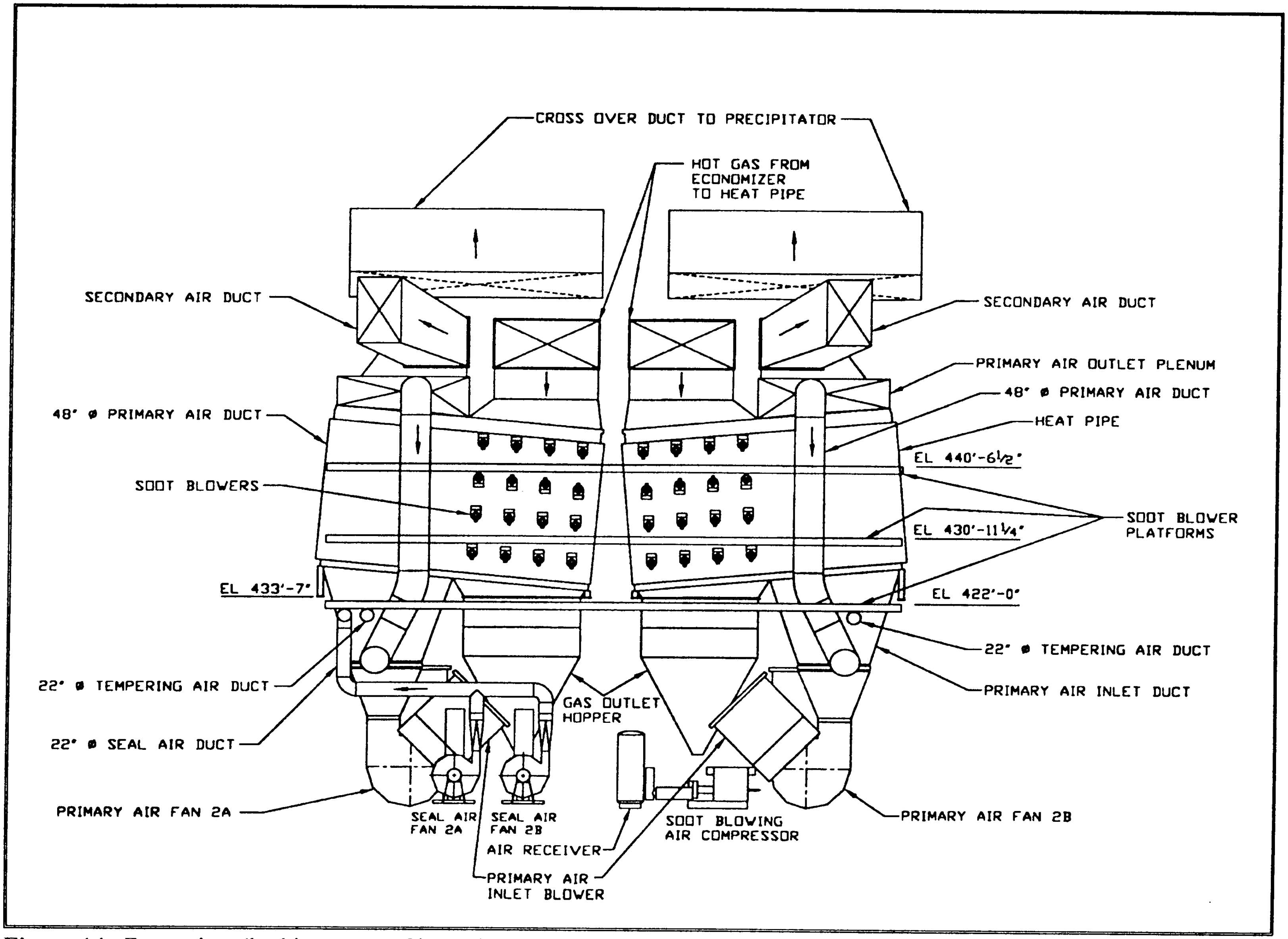

Figure 14. Front view (looking west) of heat pipe air heaters.

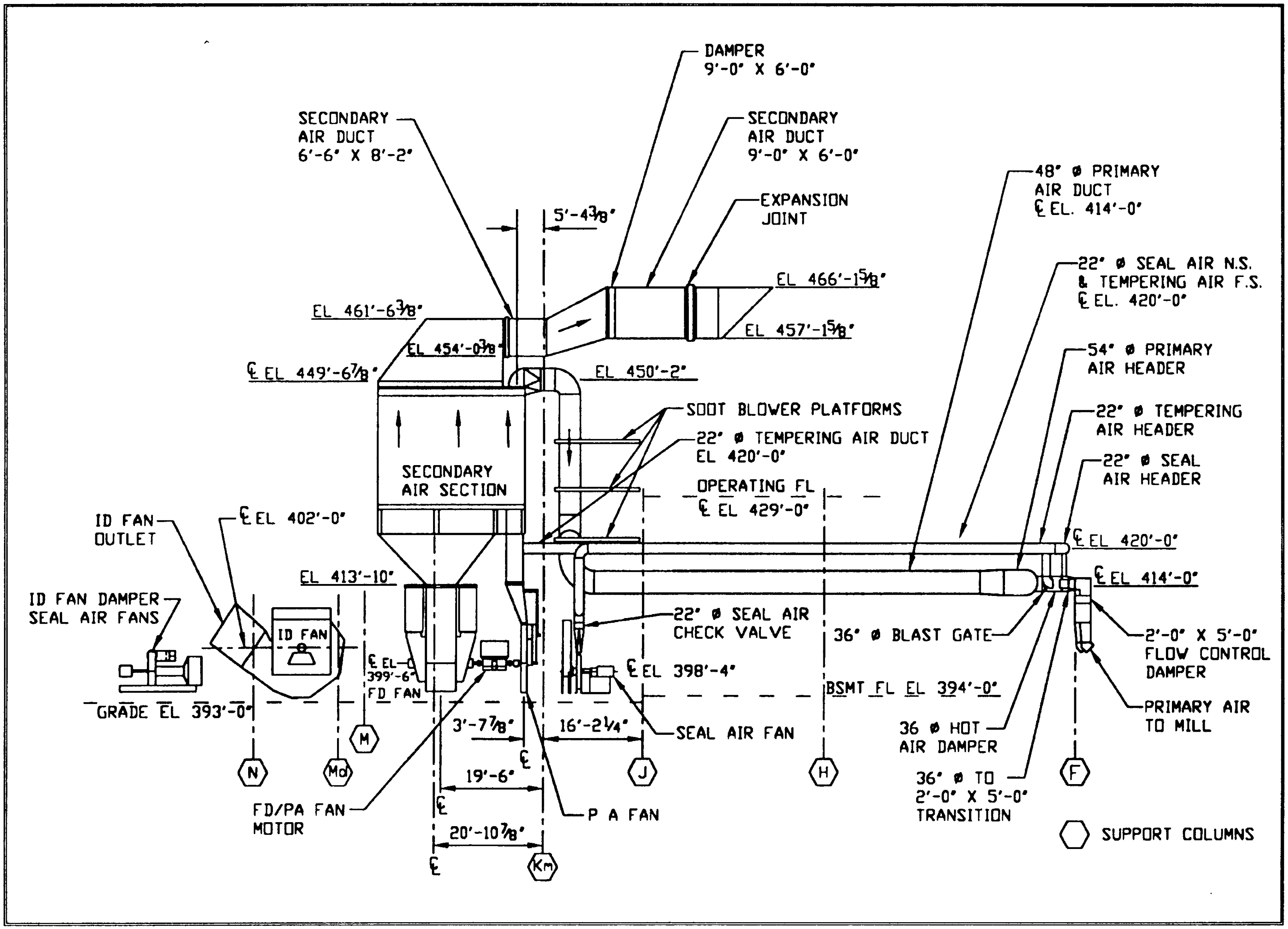

Figure 15. View (looking north) of heat pipe air sections.

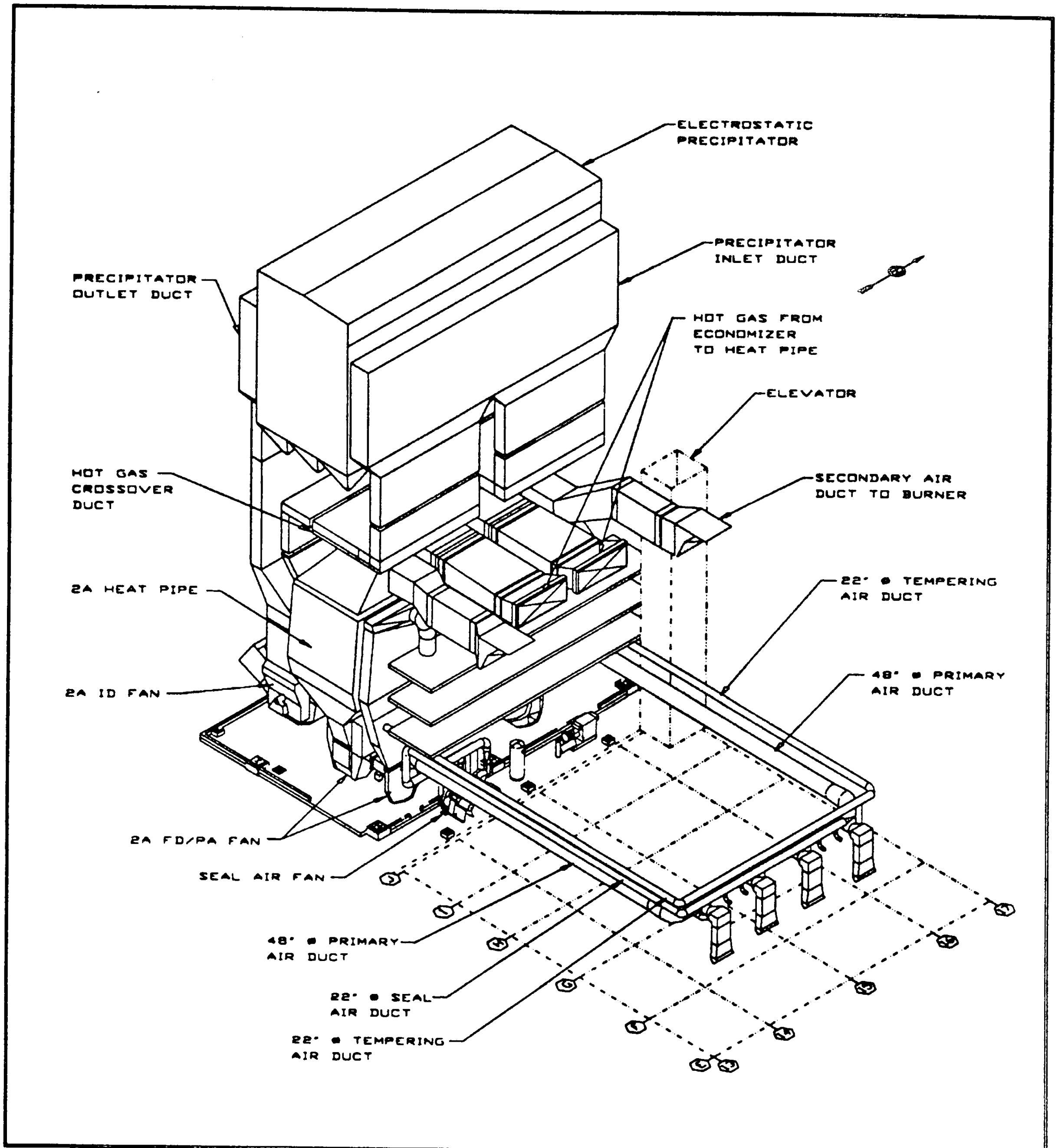

Figure 16. Isometric view of heat pipe air heater and ductwork.

3.6 Flow Modeling

A major goal of the test program was to operate the heat air heaters at the lowest possible outlet temperature consistent with low tube corrosion. Achieving a uniform flue gas distribution in the air heater is critical in limiting the development of cold gas spots. To help insure optimum equipment performance, flow model testing of the heat pipe and ductwork was performed. This allowed optimizing the gas/air flow profiles in the heat pipe and ductwork and minimization of ductwork pressure drops. The model test results were incorporated into the final full-scale ductwork design.

Fluid Systems Engineering Incorporated, of Parsippany, New Jersey conducted the flow testing in a dimensionally correct, 1/12 scale, cold-flow model of the heat pipe and ductwork. The model was fabricated from 1/4" thick clear Plexiglas with turning vanes made from thin 24 gage galvanized sheet steel. The heat pipe bundles were simulated using perforated plates within the heat pipe cases. On the flue gas side, the ductwork included all the ducting between the economizer outlet and the heat pipe inlet and all the outlet ductwork from the heat pipe outlet to the vertical riser duct at the ESP inlet (see Figure 13). On the air side, the ductwork included all ducting from the FD fan discharge to the heat pipe and all the secondary air ductwork from the heat pipe to the boiler (see Figure 15).

Flue gas and air flows through the full-scale prototype heat pipe system were simulated by drawing ambient air through the scale model using a laboratory fan. Air rates were $[1/12]^2$, i.e., [1/144] the full-scale design rate. This insured turbulent conditions in the scale model and provided a 1:1 velocity ratio between the full-scale prototype unit and the scale model. The use of a 1:1 velocity ratio coupled with geometric similarity with the full-scale unit, allowed the model to be used effectively in flow evaluations and correction of distribution problems. The use of cotton streamers and smoke observations allowed visualization of flow through the unit. Pitot and hot-wire anemometer measurements were used to quantify velocity profiles within the ductwork and in the heat pipe. Fly ash fallout in the ductwork was simulated with fine silica test particulate.

The model testing achieved the following:

1. Developed flue gas side inlet duct vane and inlet hood ladder vane designs which provided uniform flow distribution at the heat pipe entrance and within the tube banks. The design resulted in a very good velocity distribution in the center of the tube banks with a root-mean-square (RMS) deviation of only 7.63%.

2. Developed a FD fan discharge ductwork splitter vane design which improved the secondary air flow distribution to the heat pipes. The design achieved an acceptable 25.2% RMS velocity distribution in the center of the heat pipe tube banks. Additional testing indicated (no data presented) that inclusion of perforated plates in the inlet ductwork would further improve the air flow profiles. These plates were later installed following the initial operation of the full-scale air heaters.

3. Developed a design for the high baffles of the heat pipe bottom ash hoppers which minimized flow scouring in the hoppers so that the hoppers acted as an effective dropout zone for fly ash.

4. Optimized the design of the flue gas outlet duct turning vanes to establish a desired gas flow profile to the ESP particulate collector.

5. Optimized the flue gas outlet crossover duct roof baffle design to eliminate dropped out solids accumulations.

6. Developed air side outlet hood and outlet ductwork vane designs which minimized pressure drop and achieved uniform flow distribution.

The complete flow modeling test report can be found in Appendix C.

4.0 PERFORMANCE TESTING

4.1 Test Procedure Development

After a new piece of equipment is installed, the purchaser often wants to know, first, if the equipment meets design performance and then, how well does the equipment perform after being in service for an extended period? To answer these questions for the Milliken Station heat pipe air heaters, the thermal performance was measured under: (1) clean unfouled conditions, (2) fouled conditions after six months of operation, and (3) cleaned condition following a water washing to establish any performance decline. A detailed equipment test procedure specifically for the Milliken heat pipe air heater arrangement was developed (Appendix D). The detailed procedure is based on the American Society of Mechanical Engineers (ASME) Performance Test Code for Air Heaters.[4] It specifies how the air heaters will be operated, what data (temperature, pressure, composition, flow rate, etc.) will be obtained, how the data will be obtained, and how the data will be used in certain calculations. The equipment test procedure was followed each time the heat pipes were tested.

Because of the importance of air heaters to the operation of fossil fuel fired utility boilers, the ASME developed a general procedure, PTC 4.3, for establishing equipment performance. Rarely is it possible to determine equipment performance by establishing design inlet conditions to compare directly the measured flue gas outlet temperature with the design value. Fuel feedstocks may change, so flue gas composition will be different from design; ambient air temperatures change with the time of year and even the time of day; and flue gas temperatures to the air heaters will depend on boiler conditions such as cleanliness, excess air, load, steam attemporation rates, etc. The ASME code procedure avoids this problem by not requiring that design inlet conditions be established. Rather, performance data are collected under some stable operating condition (usually at high boiler load) and then corrections are applied to adjust the flue gas outlet temperature back to design conditions.

The code requires that corrections be applied for differences from design inlet air temperature, design inlet flue gas temperature, design inlet flue gas rate, and design X-ratio. The corrections are based on a simplification of the heat transfer process physics. For example, corrections for differences from design inlet air and inlet flue gas temperatures are derived based on the assumption of constant gas side efficiency (effectiveness). Applying the correction factors results in a “totally” corrected flue gas outlet temperature. Performance is determined by comparing this temperature with the design flue gas outlet temperature. If the totally corrected temperature equals the design flue gas outlet temperature, the performance exactly matches design; a higher temperature indicates a poorer than design performance; and a lower temperature indicates a better than design performance.

The ASME test code specifies how the first two corrections (for differences from design inlet air and flue gas temperature) are to be calculated but does not specify exactly how to calculate corrections for flue gas flow and X-ratio. A method for calculating these corrections is presented in the uncertainty analysis report for the totally corrected flue gas outlet temperature in Appendix E.

In addition to providing a method for comparing the measured thermal performance with design, the ASME test code also specifies procedures for comparing air leakage and air and flue gas side pressure drops with design values. Since the ASME Test Code specifies what data are to be collected and how most of the calculations are to be done, use of the code helps to reduce disputes between the supplier and the end user concerning the actual performance.

4.2 Test Port Requirements

Determining the average temperatures and compositions of all streams around the air heaters is critically important in assessing the thermal performance of the units. Multi port probe traverses are generally used to obtain temperature/flow data in the large ductwork around full-scale air heaters. Because of potential flow stratification, simple averaging of the temperature and composition data may lead to inaccurate performance calculations. To avoid this, the ASME code procedures recommend that flow weighted average temperatures and/or gas compositions generally be obtained. This was done for all performance tests.

For the Milliken test program, the ASME recommendations covering test port layouts were followed, i.e., for rectangular ducts, ports were no more than three feet apart and at least four ports were installed on each duct; for round ducts, two ports were installed at a 90-degree separation. NYSEG, ABB API, and CONSOL R&D worked together to identify sampling port needs and locations. Each heat pipe required 40 sampling ports and 20 special taps for code performance measurements. In addition to these ports and taps, 26 taps were required on each heat pipe for diagnostic purposes. The ports and taps are listed in Table 3.

Installing the ports and taps was costly since most were added to the ductwork or heat pipe in the field. Costs can be reduced if the required number and locations of ports and taps can be identified during the design phase to take advantage of shop fabrication.

Provisions also had to be made for personnel access to the sampling ports. The most difficult port location was the flue gas outlet duct on the west side of the heat pipes. Access to this area required approximately 135 feet of supported catwalk, railings, and three metal ladders. The additional structure made sampling convenient and safe and was justified based on safety concerns alone, since the ports were approximately 40 feet above the ground floor.

4.3 Performance Guarantees

The main reason for replacing the Unit 2 Ljungstrom air heaters with heat pipes was to decrease the plant heat rate sufficiently to off set most of the incremental power needed for operation of the FGD system. The heat pipe design offered the potential of achieving this by operating with lower flue gas outlet temperatures to recover more heat and by reducing fan (PA, FD, ID) and FGD pump power requirements through elimination of air leakage. The specific guarantees for the combined, two heat pipe system were to reduce the temperature of 1,500,000 lb/hr of flue gas from an entering temperature of 680°F to 253°F using 125,000 lb/hr of primary air entering at 80°F and 1,125,000 lb/hr of secondary air entering at 80°F based on a flue gas side specific heat of 0.2597 Btu/lb-°F and

Table 3 Sample Port/Tap Requirements for Each Heat Pipe				
Location	**Duct Size, Width-Depth**	**Traverse Points**	**Port Size**	**Number**
Primary Air Inlet (on HP)	17.5'-3.28'	12	2"	6
Primary Air Outlet Duct	48" dia.	20	2"	2 @ 90°
Secondary Air Outlet Duct	6'-9'	24	2"	4
Flue Gas Inlet Duct	14.5'-5.5'	20	4"	5
Flue Gas Outlet Duct	34'-2.5'	24	4"	12
Pri Flue Gas Out (on HP)	17.92'-3.28'	14	2"	7
Secondary Air Bypass (1)	17.5'-2.09'	8	2"	4
Total				**40**
Special TC Taps				
Secondary Air FD Fan Discharge		4	½"	4
Pressure Taps on Heat Pipe (2)				
Primary Air Inlet			3/8"	2
Primary Air Outlet			3/8"	2
Secondary Air Inlet			3/8"	2
Secondary Air Outlet			3/8"	2
Primary Flue Gas Inlet			3/8"	2
Primary Flue Gas Outlet			3/8"	2
Secondary Flue Gas Inlet			3/8"	2
Secondary Flue Gas outlet			3/8"	2
Total				**20**
Diagnostic Pressure Taps				
Primary Flue Gas Damper DP			3/8"	2
Flue Gas Tube Bank DPs (Front Wall)			3/8"	8
Flue Gas Tube Bank DPs (Side Wall)			3/8"	16
Total				**26**

(1) Code requirement of three foot maximum distance between ports was not adhered to since the ports were only used to check for zero flow in the bypass duct.

(2) Pressure taps are two taps spaced one foot apart and Y'd together.

an air side specific heat of 0.2469 Btu/lb-°F. The air side pressure loss was not to be more than 5.35 in. WC and the average flue gas side loss was not to exceed 3.65 in. WC. The unit is guaranteed for zero air to gas leakage. Additionally, the unit is guaranteed to operate for six months without a water wash while a 3.2% sulfur coal is fired. System cleanliness is expected to be maintained using a maximum of four sootblowing cycles per day. The thermal performance, gas-side pressure drop, and zero leakage guarantees extend to the end of the six-month period of acceptable operation.

4.4 Uncertainty Analyses

Measurement errors are a concern for all parties involved in equipment performance evaluations, particularly when determining if guarantees are being met. To determine what allowance should be given for such errors, ABB API and NYSEG requested that CONSOL R&D calculate the overall uncertainties for:

(1) The weighted average inlet and outlet temperatures for the primary air, secondary air, and flue gas streams.

(2) The air and flue gas flow rates.

(3) The air-to-gas leakage.

(4) The totally corrected flue gas temperature leaving the air heater.

Two uncertainty analyses were performed. The first analysis dealt with the items 1-3 while the second analysis covered item 4. Both analyses are presented in Appendix E.

Measurement errors fall into two categories, bias errors and random errors. The bias errors are fixed errors which remain constant during a test and cannot be reduced by repeated measurement of a parameter. An instrument off set would be an example. Random errors are errors which can be reduced by repeated measurement. Errors caused by signal noise or reading errors due to changes in personnel are examples. Both types of errors are propagated separately through the performance code calculation procedures to obtain an estimate of the individual uncertainties (bias or random) in the calculated result. These uncertainties are then combined using an appropriate statistic for the uncertainty interval of interest.

For a 95% confidence level, the uncertainty in the weighted temperatures was shown to be about ±1% of the measurement in Fahrenheit degrees. The uncertainty in the air and gas calculated flow rates ranged from 4.9% to 6.7% of the value. The air leak uncertainty was shown to be about 1.7% absolute. These uncertainties are all low and provide confidence in the calculated results. The bottom line in evaluating the thermal performance is, however, the uncertainty in the totally corrected flue gas outlet temperature for the combined primary and secondary flue gas sections. The uncertainty in the ASTM code procedure for calculating this temperature was shown to be about ±4.4°F. Therefore, for a 253°F performance target, a totally corrected flue gas outlet temperature of 257°F would still be in the expected uncertainty range and would indicate acceptable performance.

4.5 Thermal Performance

For the Milliken CCT-IV test program, the ASME PTC 4.3 procedure was used to evaluate the thermal performance of the new heat pipe air heaters. This procedure is costly and time consuming to conduct properly, but it provides an ideal means of evaluating the air heaters to determine if guarantee performance is achieved and can be used to track performance loss due to mechanical failure or fouling. The procedure establishes the actual performance regardless of shifts in inlet conditions from the design basis. As explained in Section 4.1, this is done by calculating temperature corrections to the measured flue gas outlet temperature which refer back to the design conditions. If the recalculated, i.e., totally corrected, flue gas outlet temperature is equal to or less than the design outlet temperature, the performance matches or exceeds the design. Use of the code procedure is a more exact means of assessing performance than other techniques such as, comparing or following changes in total heat transferred, thermal effectiveness, log mean temperature difference, or UA (overall heat transfer coefficient x area).

Three detailed, high load performance tests were conducted. Detailed reports covering each test may be found in Appendix F. The first test was a clean condition test conducted 41 days after a boiler start-up. The second test was a fouled condition test conducted just over six months (187 days) after a clean condition start-up. The third test was a clean condition test conducted 20 days after heat pipe washing. This last test was used to assess the thermal recovery following water washing of fouled units and to address guarantee performance since the cleaning prior to the test was considered acceptable by the manufacturer ABB API. Test results are summarized in Table 4. The table provides temperatures and flows for the main streams which pass through the heat pipes, the temperature correction terms and the corrected flue gas outlet temperatures from the primary and secondary sections, and the combined totally corrected flue gas outlet temperatures.

The clean condition tests were conducted to assess the guaranteed performance, so these tests were done in duplicate. However, for the first 2B heat pipe clean condition test (May 14, 1996), a problem with the inlet flue gas analysis was discovered after the data were collected. Although adjustments were made using an alternate calculation procedure, the result is not presented here. The fouled condition tests were conducted to establish the degree of performance decline after six months of operation. These tests were mainly of academic interest, so to save costs duplicate testing was not done.

The test results indicate the following:

1. Under clean operating conditions, the thermal performance of both heat pipes approached but never met or exceeded the guaranteed flue gas outlet temperature; i.e., the totally corrected flue gas outlet temperature was never equal to or below 253°F. The average combined temperature approach to design for the two heat pipes was 15.7°F ((18+17+12)/3) and 18.5°F ((20+23+16+15)/4) for the first and second clean condition tests, respectively. Based on the CONSOL analysis, the uncertainty in these numbers is about ±4.4°F for a 95% confidence level.

2. When clean, the 2B heat pipe performed slightly better than the 2A heat pipe. For the first clean condition test, the approach to the design flue gas outlet temperature was 12°F for 2B versus 17.5°F (avg.) for 2A. For the second clean condition test, the approaches to the design flue gas outlet temperature were 15.5°F and 21.5°F for the 2B and 2A heat pipes, respectively.

3. During the period when the fouled condition performance was measured, the 2B heat pipe fouled more rapidly than the 2A unit. This is indicated by the higher approach to design flue gas outlet temperature for the 2B heat pipe at the end of the six-month test period, i.e., 83°F for the 2B versus 30°F for 2A and by higher flue gas side pressure drops, i.e., 9.0 in. WC for the 2B heat pipe and 5.9 in. WC for 2A (pressure drops corrected to design flow and temperature). Slight differences in the flue gas flow balancing and temperature control between the two heat pipes is likely responsible for the more rapid fouling of the 2B heat pipe during this particular test period. A review of other operating periods indicates a random behavior with respect to which heat pipe fouled quickest.

4. Washing the heat pipes was very effective in removing cold-end fouling deposits and recovering thermal performance. This is shown by flue gas side pressure drop recoveries after washing and by the close approach of the totally corrected flue gas outlet temperatures for the clean condition tests.

5. The results indicate a slight performance decline for both heat pipes between the two clean condition tests. The approach to the design flue gas outlet temperature increased 4°F (17.5°F avg. increasing to 21.5°F avg.) for the 2A heat pipe and 3.5°F (12°F increasing to 15.5°F avg.) for the 2B unit. This loss in performance, may or may not, be real since it falls within the ±4.4°F uncertainty of the analysis procedure. If the decline is real, it maybe due to a slight difference in the heat transfer surface fouling between the two tests, or to loss or deterioration of some heat transfer fluids. Longer term performance monitoring is needed to establish if there is a trend.

Table 4
Heat Pipe Thermal Performance
Totally Corrected Flue Gas Outlet Temperatures (1)

	2A					2B				
Date	5/14/96	5/15/96	10/7/96	11/7/96	11/8/96	5/14/96	5/15/96	10/8/96	11/7/96	11/8/96
Test Condition	**Clean**		**Fouled**	**After Wash**		**Clean**		**Fouled**	**After Wash**	
Boiler Load, MW net	**149.0**	**147.2**	**142.4**	**146.9**	**147.8**	**(2)**	**147.2**	**142.2**	**146.8**	**147.8**
Primary Section										
Primary Air Rate, lb/hr	75,900	73,600	56,400	58,800	73,500		73,500	59,300	53,600	68,700
T In, °F	111	116	120	115	106		116	112	115	107
T Out, °F	593	605	600	604	576		590	614	614	592
Primary Flue Gas Rate, lb/hr	98,070	105,700	85,300	92,980	97,000		85,900	111,800	80,510	91,570
T In, °F	675	677	667	659	658		671	655	663	661
T Out, °F	294	304	330	324	288		269	368	325	283
T Out No Leak, °F (3)	323	356	368	368	322		287	404	349	317
Temperature Corrections For Difference From:										
Design Entering Air Temp, °F	273	280	305	303	271		243	351	303	265
Design Entering Flue Gas Temp, °F	296	305	335	332	295		272	380	332	290
Design X-Ratio, °F	327	307	318	299	315		330	293	316	308
Air Leak Correction, °F	29	52	38	44	34		18	36	24	34
Design Flue Gas Flow Rate, °F	292	300	332	324	287		270	362	328	283
Corrected Outlet Temp, °F	335	332	338	329	337		325	319	328	329
Secondary Section										
Secondary Air Rate, lb/hr	497,200	481,200	498,200	531,100	539,700		482,100	500,900	516,300	535,400
T In, °F	89	95	102	94	85		94	93	94	83
T Out, °F	612	619	588	584	578		612	490	596	595
Secondary Flue Gas Rate, lb/hr	649,200	636,300	659,600	655,100	658,600		606,200	562,300	658,200	686,500
T In, °F	675	677	667	659	658		671	655	663	661
T Out, °F (Ht. Bal)	296	303	321	283	276		281	323	290	283
Temperature Corrections for Differences From:										
Design Entering Air Temp, °F	291	293	307	274	273		271	315	281	281
Design Entering Flue Gas Temp, °F	298	304	326	290	284		284	333	296	290
Design X-Ratio, °F	264	267	284	268	264		260	331	265	257
Design Flue Gas Flow Rate, °F	297	304	321	283	276		284	328	290	281
Corrected Flue Gas Outlet T, °F	261	260	276	265	267		257	339	262	260
Totally Corrected Temp, °F	**271**	**270**	**283**	**273**	**276**		**265**	**336**	**269**	**268**
Approach To Design, °F	**18**	**17**	**30**	**20**	**23**		**12**	**83**	**16**	**15**

(1) Based On CONSOL Method Of Calculating Flue Gas Outlet Temperature Corrections For Deviations From Design X-Ratio And Design Gas Flow.
(2) Test Result Questionable Due To Problem In Measuring The Inlet Flue Gas Composition. See Performance Report For Discussion.
(3) Air Leak From Sootblower Penetrations Assumed To Flow Into Primary Flue Gas Section.

4.6 Air Leakage

The all welded construction of the heat pipe air heaters prevents air leakage from the higher pressure primary and secondary air sections into the flue gas sections. The modules are seal welded together and all tubes are welded to the divider plate which separates the flue gas and air sections. Baring cracked or missing welds, no leakage should occur. However, because the flue gas sections operate at nominally 10 in. WC to 15 in. WC negative pressure, significant amounts of air can be drawn into the heat pipes through check valves on the sootblowers and at the sootblower wall penetrations. The check valves allow a continuous ambient air purge to sweep fly ash and flue gas from the lances when the sootblowers are inactive. The leak at the wall penetrations is due to the designed-in, loose-fit (3/32" annular gap) between the sootblower lance and the wall seal ring.

The ASME air heater performance code was followed to determine the total leakage associated with the ambient air infiltration. This was done by determining the inlet and outlet flue gas flow rates and then subtracting the inlet rate from the outlet rate. The results for the May and November 1996 clean condition tests are presented in Table 5 as percentages of the inlet flue gas flows. By the code procedure, the gas rates are calculated based on the measured inlet and outlet flue gas compositions, the measured coal feed rate, and the coal composition.

Table 5

Measured Air Leakages, wt. % of Inlet Flue Gas Flow

Heat Pipe		**2A**		**2B**	
Date	Boiler Load MW	Total Leakage	Unaccounted Leakage	Total Leakage	Unaccounted Leakage
5/14/96	149	2.7	1.9	ND	ND
5/15/96	147	4.4	3.6	1.4	1.2
11/7/96	147	2.5	2.4	1.2	1.0
11/8/96	148	2.4	2.2	2.2	2.1
Average		3.0	2.5	1.6	1.4

During both clean condition tests, the lance purges for all 32 sootblowers were measured by sealing one end of a 2-inch diameter plastic tube around the lance check valves and using a mini pitot tube to measure the air velocity through the tube. The total purge rates averaged 2,680 lb/hr or about 84 lb/hr per sootblower. This is typically less than 0.18% of the inlet flue gas flow at full boiler load (nominally 1,500,000 lb/hr). Similar measurements at the sootblower wall penetrations were not possible due to equipment clearances. The leak at the wall penetrations was taken to be the unaccounted air leak. This is the difference between the total leak rate determined by the ASME code and all air in flows which can be accounted for, such as, the lance purge flows and the air consumption of the infrasonic cleaner (4,800 lb/hr for the 2A heat pipe only, for some tests).

As shown in Table 5, the leak rates are quite low for the heat pipes. Total leakages averaged 3.0 wt. % and 1.6 wt. % of the inlet flue gas flow for the 2A and 2B heat pipes, respectively. The uncertainty in the leak rate is about ±1.6 wt. % absolute. Similarly, the unaccounted leakages, which are taken to be mainly the leak at the sootblower wall penetrations, averaged 2.5 wt. % and 1.4 wt. % for the 2A and 2B heat pipes, respectively. The somewhat higher leak rates for the 2A heat pipe maybe due to differences in wear/fit of sootblower lance wall seal rings or due to the presence of other leaks, such as, leaks at manway door seals.

4.7 Unit Pressure Drops

Checks were made of the guaranteed pressure drops for both air heaters during the clean condition performance tests. To insure accuracy of the differential pressure measurements, special pressure taps were installed on the heat pipe casings per ABB API instructions. These taps consisted of two 1/8" diameter holes drilled through the casings and spaced horizontally one foot apart. The holes were deburred to prevent turbulence at the inside wall opening. Each tap pair was "Y'ed" together and then connected to the appropriate side of a liquid manometer for differential pressure measurement.

The operating pressure drop checks were done in accordance with the ASME PTC 4.3 code procedures. These procedures correct the measured pressure drops for deviation from design gas or air flow and temperature. The design pressure drops are:

Flue Gas 3.65 in. WC
Primary Air 3.60 in. WC
Secondary Air 5.35 in. WC

Measured performance results are presented in Table 6 as a percent of design. The results show that the actual performance essentially met or exceeded the design.

Table 6
Performance Summary -- Heat Pipe Pressure Drops
Clean Condition Tests

	Boiler Load	Fully Corrected Pressure Drops, % of Design		
Date	MW net	Flue Gas	Primary Air	Secondary Air
Unit 2A				
5/14/96	149	98%	74%	106%
5/15/96	147	99%	73%	110%
11/7/96	147	95%	86%	99%
11/8/96	148	94%	73%	98%
	Average	**97%**	**76%**	**103%**
Unit 2B				
5/14/96	151	106%	75%	104%
5/15/96	147	107%	78%	102%
11/7/96	147	101%	96%	99%
11/8/96	148	95%	87%	94%
	Average	**102%**	**84%**	**100%**

4.8 Corrosion Monitoring Program

Previous work done by EPRI and NYSEG had demonstrated that the CAPCIS corrosion monitoring system could be used in condensing environments, such as the flue gas streams in and around utility air heaters. NYSEG purchased two CAPCIS electrochemical corrosion probes and DOS based VISICOR™ software to log and graphically present the data as part of the preliminary testing done to select materials of construction for the Milliken heat pipe air heaters. The testing included monitoring corrosion rates in two pilot heat pipes, one at EPRI's Environmental Control Technology Center, the other at NYSEG's Milliken Station, and corrosion monitoring in the ductwork ahead and downstream of the Milliken Unit 2 precipitator. To reduce the preliminary test costs, probe temperature control hardware and field data acquisition/signal processing electronics were borrowed from EPRI. The experience gained with this equipment indicated that the corrosion probe system was sensitive to changes in corrosion rates and that after being calibrated, could provide reasonably accurate estimates of the actual corrosion rates.

As part of the CCT-IV test program, NYSEG committed to installing and testing an on-line, real-time corrosion monitoring system supplied by CAPCIS March Ltd. Two air-cooled corrosion probes, (Cor-Ten A™ , SA-178A carbon steel) were installed at the outlet of the Milliken Station 2B heat pipe and two, passive (not air-cooled) wall corrosion probes (SA-178A CS) were installed in the ductwork just ahead of the Unit 2 FGD scrubber. This probe combination allowed for corrosion monitoring of the actual materials of construction in the expected severest condition locations. The air-cooled probe temperatures were controlled to match or be offset from the temperatures of cold-end heat pipes which had been fitted with thermocouples. This provided a means of maximizing heat recovery by operating at the lowest flue gas outlet temperature consistent with a low corrosion rate. After gaining confidence in the monitoring system, the intent was to use the corrosion probe signals to control the secondary air bypass damper in the heat pipe.

NYSEG had CAPCIS refurbish the air-cooled corrosion probes and purchased all new temperature control hardware, field electronics, and software. At the time of purchase, CAPCIS was significantly revising their corrosion monitoring systems. The field electronics was redesigned and the DOS software was replaced with a graphical interface UNIX based database system. These changes created many hardware and software problems which are being addressed at the time of this writing. Currently, there is insufficient historical data on either the air-cooled or the passive probes to draw any conclusions concerning corrosion monitoring. At this time, CAPCIS (now Integriti Solutions) is standing behind their equipment and is working with NYSEG to provide an operable, debugged monitoring system.

5.0 OPERATIONS

5.1 History

Table 7 summarizes the operations history for the heat pipe air heaters. A more detailed history is presented in Appendix G. Between June and December 1994, the Unit 2 boiler was off line to allow rebuilding and upgrading of the electrostatic precipitator particulate collectors, construction and tie-

Table 7
Milliken Heat Pipe Air Heater Operations Summary

	1994
6/18 - 12/11	Unit 2 outage to rebuild and modify the ESP, S-H-U scrubber tie-in, and heat pipe installation.
12/11	Start up of boiler, heat pipe put into service.
	1995
1/25 - 1/27	ABB API obtains field data on heat pipe performance. Thermal performance is less than expected. Flue gas outlet temperatures are higher than design.
2/27 - 3/3	Initial repairs made to the heat pipes (installed perforated distribution plates on primary and secondary air fan outlets, installed special condenser end baffles in the primary section of the 2B heat pipe, checked vacuum on approximately 110 tubes).
	When heat pipes were back in service, the operation of unit 2B primary air heating section improved.
	ABB API's analysis of the gas from naphthalene tubes reveals that high levels of H_2, CO_2, and ethylene are being generated.
5/16	CONSOL analysis of naphthalene samples indicates that two low level contaminants in the naphthalene are breaking down and are likely responsible for the non-condensable gases found by ABB API in the naphthalene tubes.
9/15 - 10/2	ABB API repaired both heat pipes. Vented and resealed 2,400 naphthalene containing tubes to remove non-condensable gases.
10/2 - 3/15/96	Operation of heat pipes monitored. Repairs resulted in improved performance. Performance observed to gradually decline due to cold-end fouling. Sootblowers not effective in keeping units clean.
	1996
3/15 - 4/3	Heat pipes washed. Low frequency (infrasonic) cleaner installed on Unit 2A heat pipe.
5/13 - 5/17	First detailed air heater performance tests conducted.
7/19	Ceased operation of the infrasonic cleaner due to development of cracks in the inlet ductwork to the 2A ESP.
8/30	Placed infrasonic cleaner back in service at 75% of full power.
9/6	Began operating infrasonic cleaner at full power again.
10/7 - 10/8	Detailed fouled condition heat pipe performance tests conducted.
10/11 - 10/18	Shutdown Unit 2 boiler for heat pipe cleaning and repair of fatigue cracks in the 2A ESP inlet ductwork. Ductwork stiffened.
11/7 - 11/8	Second detailed clean condition heat pipe performance tests conducted.
	1997
3/31 - 4/20	Unit 2 boiler off-line for annual outage. Heat pipes cleaned. Operation of infrasonic cleaner was discontinued because cracks were found in division wall separating the primary and secondary flue gas sections in the 2A heat pipe.
10/24 - 10/31	Unit 2 taken off line -- convenient time for heat pipe cleaning.
	1998
4/24	Unit 2 boiler off-line for annual outage.

in of the S-H-U flue gas desulfurization scrubber system, and installation of the heat pipe air heaters. The Unit 2 boiler was placed back in service on December 11, 1994.

The initial operations indicated that the heat pipe air heaters were not performing up to design expectations, i.e., the full load flue gas outlet temperatures were in range of 270-290°F compared to expected temperatures of 250-260°F. In late January 1995, ABB API obtained field data which confirmed these results. The field data also indicated that there were problems with the inlet air side flow distributions. To correct the problems and allow further analysis, ABB API recommended (1) installation of additional tube diagnostic thermocouples (TCs), (2) installation of special condenser-end baffles to help redirect flue gases traveling down the heat pipe walls back into the tube bundles, and (3) the sampling of selected heat pipe tubes to determine if the heat transfer fluid had deteriorated.

Initial repairs were made to the heat pipes between February 27 and March 3, 1995, during a boiler shutdown to clean turbine screens. The recommended ABB API TCs were installed; perforated plates were added to the primary air and FD fan discharges to improve the air flow distribution to the heat pipes; condenser end baffles were installed in the primary section of the 2B heat pipe; and the contents of several tubes were sampled. One hundred and ten tubes were evacuated and resealed. After the boiler was back in service, the operation improved for the primary air heating section of the 2B heat pipe.

An ABB API analysis of heat pipe contents indicated noncondensable gases containing hydrogen, carbon dioxide, and ethylene were being generated in the naphthalene tubes. An analysis done by CONSOL of fresh naphthalene and “used” naphthalene from the Milliken air heaters indicated that certain contaminants in the naphthalene were decomposing and generating the gases. To eliminate the noncondensing gases, ABB API recommended that the heat pipes with naphthalene be reevacuated under cold conditions and momentarily vented under hot conditions. This procedure would correct the problem assuming that the contaminant decomposition was a one time event.

Heat pipe repairs and modifications were affected by ABB API during the two week outage beginning September 15 to October 2, 1995. By October 5, 2,400 tubes were evacuated, hot vented and resealed. When the units were again in operation, ABB API evaluated the performance and notified NYSEG that the heat pipes were performing as designed and recommended that a performance test be conducted within 60 days. To insure that continued decomposition of naphthalene contaminants would not be a problem, the performance tests were deferred. Additionally, there was concern that the heat pipes were not clean enough for a guarantee performance test. The boiler outage schedule had allowed time for only a partial washing of the heat pipes. A complete washing was not done until April 1996.

Between October 2, 1995 and March 15, 1996, the performance of both heat pipes gradually deteriorated due to cold-end fouling. The flue gas side outlet temperatures and the pressure drops across the tube banks both increased. Unit 2 was shut down between March 15 and April 3, 1995 for water washing of the heat pipes and to allow installation of an infrasonic cleaner (Infrafone™) on the 2A heat pipe. For comparison, the 2B heat pipe air sootblowers were fitted with special expanding nozzle jets.

NYSEG and CONSOL conducted the first detailed ASTM Code procedure performance tests between May 13, and May 17, 1996. These tests were not observed by ABB API personnel since they felt that the units had not been cleaned sufficiently. The performance results were, however, the best obtained during the test program.

When first placed in service, the infrasonic cleaner was operated continuously at full power level. This appeared to stave off fouling in heat pipes, particularly within the 2A unit. However, because of the intense sound level (140 dB) within the equipment, resonation caused cracks in the ductwork between the 2A heat pipe outlet and the 2A ESP inlet. The infrasonic cleaner was then taken out of service for approximately 1.5 months (7/19/96 to 8/30/96) until temporary repairs could be made to the ductwork. The unit was placed back in continuous service at three-fourths power on August 30, 1996. Fouling of both heat pipes appeared to intensify after the infrasonic cleaner was placed back in service. However, it is not known if the apparent increase in fouling was caused by the sudden sloughing of deposits which had been laid down while the infrasonic cleaner was out of service, or to increased acid/fly ash deposition due to the change over to a higher sulfur coal. Commencing on September 6, 1996, the infrasonic cleaner was again operated at full power. This did not appear to reduce the fouling or fouling rate.

On October 7, and October 8, 1996 heat pipe tests were again conducted to establish the fouled condition performance after approximately six months of continuous operation. The tests showed that the 2A heat pipe with the infrasonic cleaner, was somewhat less fouled and operating better than the 2B heat pipe. However, the thermal performance of both heat pipes had degraded significantly and flue gas side pressure drops were high. The Unit 2 boiler was taken out of service between October 11, and October 18, 1996 for heat pipe washing and S-H-U scrubber cleaning. During this period, the ductwork between the 2A heat pipe and the ESP inlet was stiffened to eliminate resonant vibrations. The stiffening was accomplished by internally bracing the vertical ductwork using a series of four-inch diameter pipes welded to opposite duct walls.

On October 17, 1996, representatives of NYSEG, ABB API, and CONSOL R&D inspected the washed heat pipes. One small area with some deposition was located in the 2B heat pipe. After this area was cleaned, all parties agreed that the units were clean and ready for testing. Performance tests were then scheduled and conducted on November 7 and November 8, 1996. These tests served a dual purpose of being both guarantee performance tests and tests which demonstrated thermal performance recovery following cleaning. The tests were the last detailed performance tests conducted during the demonstration program. Operations of the heat pipes were followed using computer-logged process data for the remainder of the test program.

An annual outage for the Unit 2 boiler occurred between March 31 and April 20, 1997. During the outage, the heat pipes were inspected and cleaned. Cracks were found in the division walls between the primary and secondary flue gas sections in the 2A heat pipe. It was speculated that the operation of the sonic cleaner may have contributed to the cracking. The cracks were repaired and it was decided to discontinue the operation of the sonic cleaner.

After the annual outage, the Unit 2 boiler and heat pipes were placed back in service on April 20, 1997. Throughout 1997, the heat pipe air heaters were operated in a normal fashion. Except for one short boiler outage in May to fix a superheater tube leak, Unit 2 remained in service until October 24, 1997 when it was convenient to wash the heat pipes. Between October 31, 1997 and April 24, 1998, the Unit 2 boiler and heat pipe air heaters were again operated in the normal fashion with no major operating problems experienced.

5.2 Primary/Secondary Air Distribution Problems

After initial operations indicated below design performance for the heat pipe air heaters, ABB API obtained diagnostic performance data on the units during the last week of January 1995. Analysis of the secondary air outlet temperature data indicated a north-south bias in the temperature distributions. The temperature biasing for the A and B heat pipes appeared to be mirror images indicating that the non-uniform discharge velocity profiles of the FD fans might be responsible. Additionally, primary air temperatures within the upper sootblower lanes showed steep drop offs with distance from the partition plate (distance from the evaporator end). This indicated the possible presence of noncondensing gases in the hottest heat pipes.

To help flatten the fan velocity profiles, ABB API recommended installation of perforated plates at the primary air and secondary air (FD) fan outlets. The original heat pipe flow modeling study recommended similar distribution plates (see Appendix C). Figures 17 and 18 show how the perforated plates were installed. The plates were made of 10 gage carbon steel with 63 percent open area.

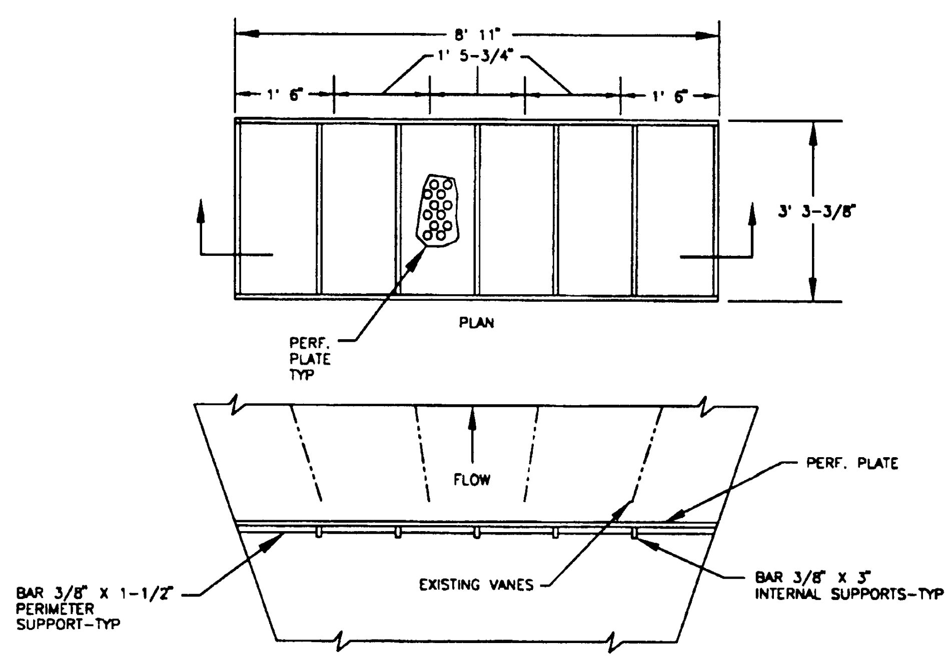

Figure 17. Perforated plate installation in the primary air fan discharge duct.

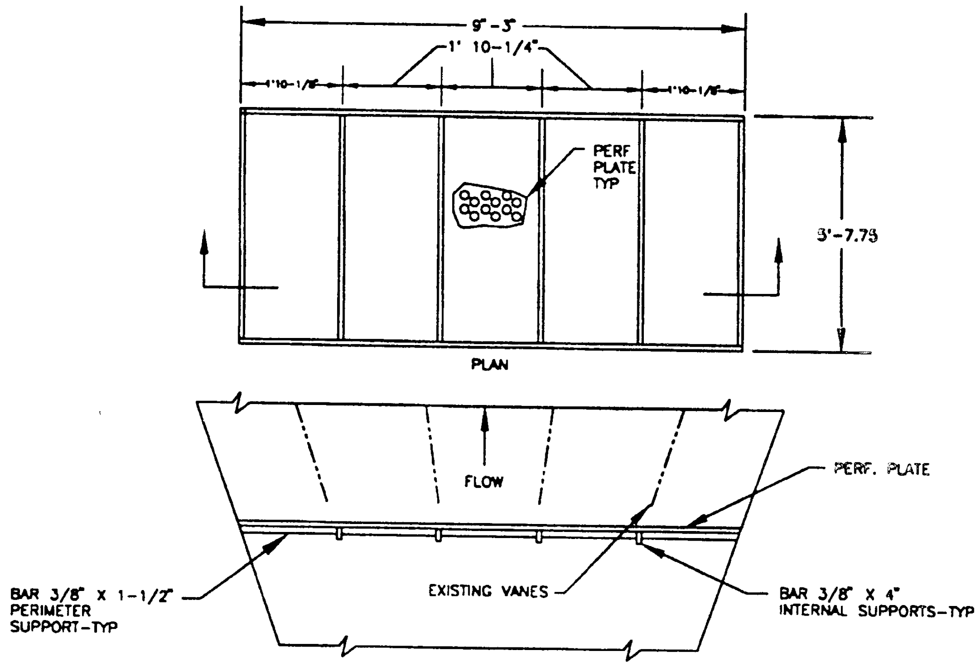

Figure 18. Perforated plate installation in the FD fan discharge duct.

A small amount of noncondensable gas in the heat pipes is often unavoidable due to the presence of trace contaminants in the working fluids and/or the breakdown of the contaminants. In an operating heat pipe the noncondensing gases are forced to the top end of the condenser tubes. This creates a dead zone where little heat transfer takes place. To improve the overall heat transfer performance, ABB API recommended installation of short (12" to 20" wide) condenser end baffles. The baffles force air flowing across the dead zones back into the center of the heat pipes where the air can be heated. The baffle installation is shown in Figure 19. Baffles were installed in both the primary and secondary air heating sections of both heat pipes.

The effects of the perforated air distribution plate and condenser-end baffle modifications cannot be isolated. The changes were made simultaneously along with the re-evacuation of the naphthalene filled tubes to remove noncondensable gases. However, the combination of changes improved the heat pipe thermal performance and reduced the north-south outlet temperature bias of the secondary air from the air heaters. The original ABB API measurements taken in January 1995 showed an average north-south temperature difference of about 40 °F. Data taken during the first performance test in May 1996 with the air heaters in a clean condition, indicated average temperature differences of only 11°F-16°F.

5.3 Heat Transfer Fluid Degradation

Based on the results of the January 1995 heat pipe air heater diagnostic tests, ABB API elected to measure the pressures in selected heat pipe tubes and to analyze the vapor components. During the February 27 to March 3, 1995 Unit 2 shutdown, ABB API checked the vacuum in approximately 110 tubes. Many of the tubes with naphthalene working fluids had internal pressures exceeding the naphthalene vapor pressure indicating the presence of noncondensable gases. Analyses of the gas revealed high levels of hydrogen (60-65 vol. %), carbon dioxide (18 vol. %), and ethylene (18 vol. %).

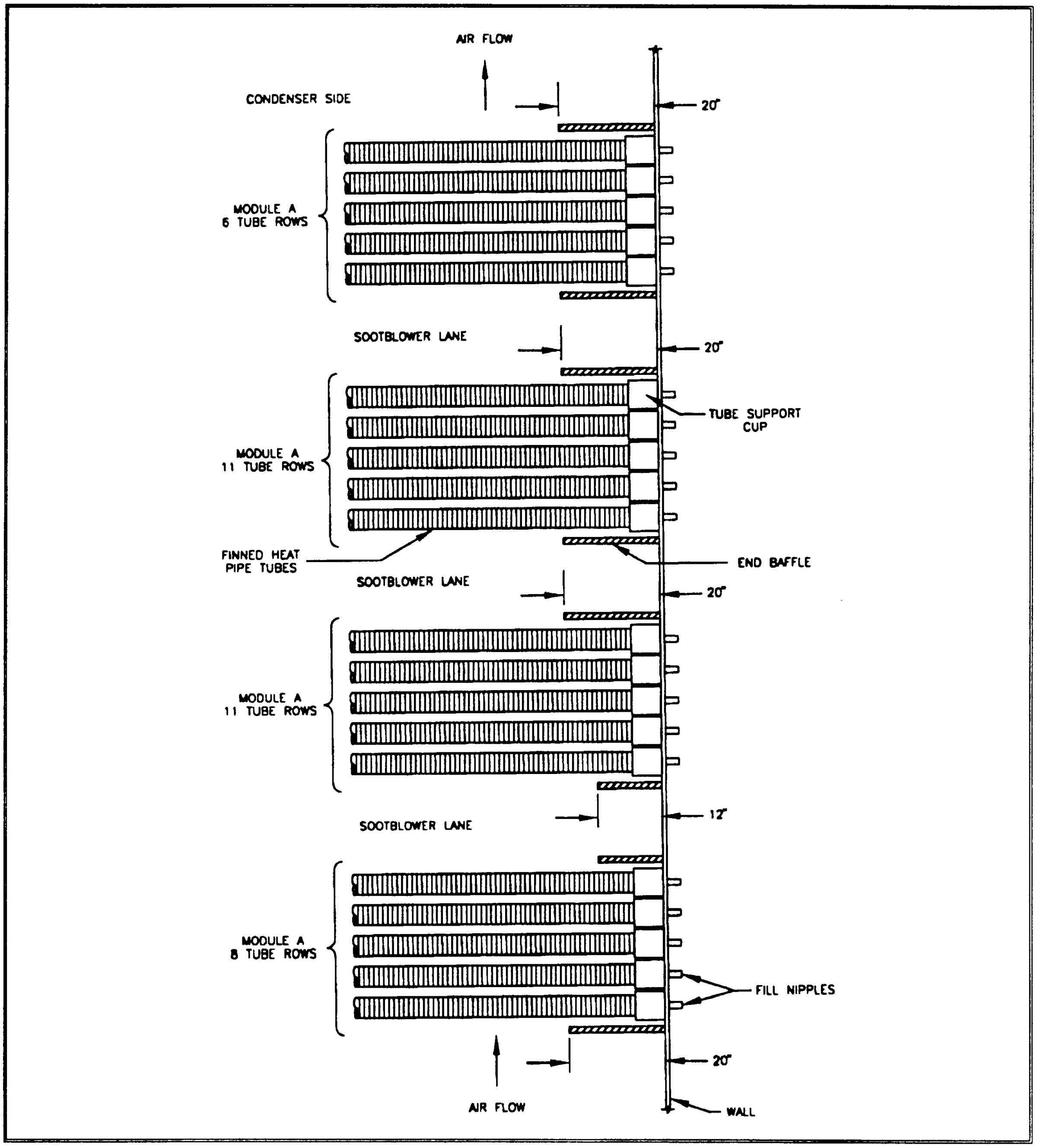

Figure 19. Condenser end baffle installation.

ABB API specified a 99.95 wt. % purity for the naphthalene use in the heat pipes. However, some naphthalene at 99.5 wt. % purity was later determined to have been received. When it was know that noncondensable gases were being generated, NYSEG requested samples of fresh unused naphthalene, used naphthalene from the heat pipes, and thionaphthene, the contaminant suspected of causing the gas generation. CONSOL R&D analyzed the samples by gas chromatograph mass spectroscopy (GC/MS). The results, which are presented in Appendix H, showed that the thionaphthene concentrations in the fresh and used samples were quite similar indicating that thionaphthene was not decomposing and was not likely responsible for the gas generation. The more likely cause was decomposition of two unidentified compounds which were found in the fresh but not in the used naphthalene. Subsequent analyses indicated that there were no strong inorganic acids in either the fresh or used naphthalene which could have reacted with the heat transfer fluid or heat pipe tube metal to generate noncondensable gases.

To eliminate the gas generation problem, ABB API proposed to install valves on the heat pipe fill stems so that the naphthalene tubes could be re-evacuated under cold conditions, and then, to vent the heat pipes under hot conditions to remove gases forced to the condenser end. Since the unidentified compounds boiled at temperatures below naphthalene, this procedure had a high potential of success. During the September to October 1995 shutdown of Unit 2, ABB API vented and resealed 2,400 naphthalene-containing tubes.

When Unit 2 was back in service, ABB API determined that the heat pipes were working as designed and recommended performance testing within 60 days. The first clean condition performance tests were, however, not conducted until May 1996. This insured against an overly optimistic performance result by providing some additional time for any remaining naphthalene contaminants to decompose. A second clean condition test was conducted in November 1996. The ABB API re-evacuation/hot vent procedure appears to have been successful in removing the contaminants since where was only a small deterioration in thermal performance between the two heat pipe tests. The thermal performance decline amounted to a 2°F to 5°F increase in the totally corrected flue gas outlet temperature. Alternate explanations, for the small performance decline, include: test result variation (the result difference is about equivalent to uncertainty level), the possibility of a difference in fouling level, or, as will be discussed in the next section, loss of some of the naphthalene fill fluid.

5.4 Naphthalene Leaks

Working fluid leakage was not a concern or a problem for the heat pipes as originally constructed. The individual heat pipes had an all-welded construction with seal welded end caps and crimped and seal welded fill tube connections (see Figure 3). Leakage became a concern to NYSEG after the naphthalene filled tubes were modified to remove the buildup of noncondensable gases as discussed in the previous section. ABB API recommended and installed Swagelok “P” series purge valves with short capped extension nipples on each heat pipe that contained naphthalene. These modifications allowed removal of the noncondensable gases from the heat pipes and future re-venting should additional gas generation take place.

After the modifications were made, there was a strong naphthalene odor in the plant and no odor prior to the modifications. Because of this, NYSEG instituted a naphthalene monitoring program. There was concern that some or all the modified tubes were leaking and with time would become exhausted of heat transfer fluid.

Figure 20 shows the end of a modified heat pipe tube sheet with the capped fill tube nipple extensions protruding through soft insulation. The insulation covers the fill tube purge valves. Under normal conditions, the tube sheets ends are covered by casing panels. To check for naphthalene leaks, test ports were added to the casing panels. Each port consisted of a capped pipe nipple extending from the casing panel through the external insulation and the corrugated lagging. One test port was installed for each naphthalene module, three ports in the primary sections and four ports in the secondary sections of each heat pipe (refer to Figure 7 for the layout of the naphthalene modules).

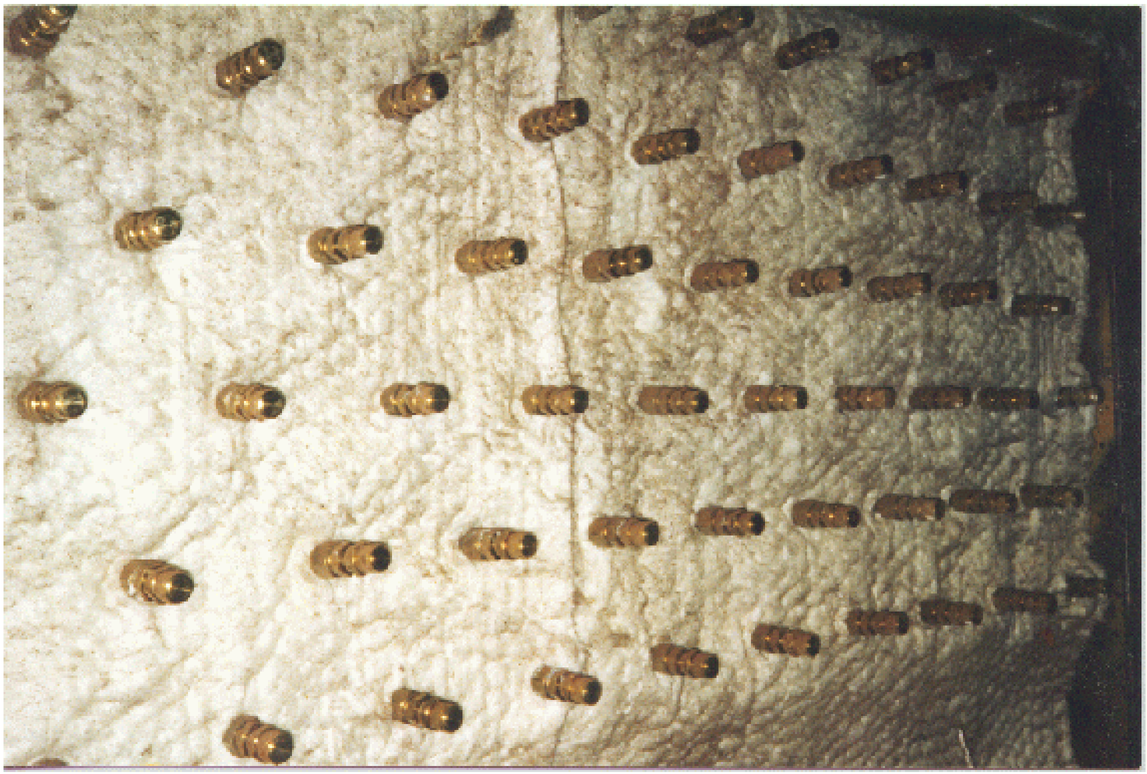

Figure 20. Modified fill nipples on heat pipes with naphthalene working fluid.

Naphthalene leak measurements were made using a Photovac Microtip HL-200 analyzer calibrated with 98.5ppm isobutylene. The analyzer measures the presence of hydrocarbons using a photo ionization detector. The amount of naphthalene was determined from the instrument output and a relative response factor for naphthalene. Typically, the sampling procedure was to open a port for 10 seconds, sample for 10 seconds and take a reading.

The leak check results are presented in Table 8. For the first four sampling periods, the data indicate an overall higher leak rate for the 2A heat pipe than for the 2B heat pipe. Most importantly, the last sampling data (December 9, 1997) suggests that the leaking tubes in both heat pipes have become exhausted. This may mean that all of the originally leaking tubes are now empty and that there will be no additional effect on the heat pipe thermal performance. However, this still remains to be proven. Continued periodic leak check monitoring will be needed to determine if the system has stabilized.

Table 8						
Naphthalene Leak Check Measurements						
		Naphthalene Concentrations, ppm				
Heat Pipe	Module (1)	5/21/96	9/18/96	10/28/96	3/4/97	12/9/97
2A	A1	1,400	8,100	$2,000	3,000	66
2A	A2	87	24	$30	50	8
2A	A3	265	230	$700	400	22
2A	B1	76	360	#5	200	22
2A	B2	28	230	#100	400	18
2A	B3	95	0	#10	30	1.5
2A	C3	0	0	0	0	0
2B	A1	NS	0	$400	70	20
2B	A2	NS	0	0	0	0
2B	A3	N/I	N/I	0	0	0
2B	B1	95	0	$100	300	8
2B	B2	N/I	N/I	$100	0	0
2B	B3	N/I	N/I	0	20	0
2B	C3	86	360	$200	550	0
(1) See Figure 7. NS - Not sampled, N/I - Port not installed						

5.5 Cold-End Fouling

As shown in Figure 5, the heat pipes are constructed with four levels of tube banks (modules). Since the flue gas flow through the air heaters is downward, the bottom tube banks are the cold-end modules. As with Ljungstrom and tubular air heaters in coal-fired service, these cold-end sections tend to gradually foul. The Milliken heat pipe cold-end deposits contained high levels of sulfur (14 wt. %) indicating that the fouling is caused by sulfur trioxide (SO_3) condensation from the flue gases (see the fouled condition performance report in Appendix F). Condensing SO_3 reacts with water vapor forming a sticky sulfuric acid liquid which traps fly ash. Gradually, fly ash/acid deposits build up in the cold-end module restricting the flue gas flow through the unit. The fouling is dependent upon the amount of SO_3 in the flue gases, the cold end metal temperatures, and the effectiveness of the sootblowing.

At Milliken, the heat pipes are washed approximately every six months to remove the cold-end deposits. In all but the cold-end modules, the heat pipes on the flue gas side appear as shown in Figure 21 with tubes and fins free of deposits. Normally, most of the top side of the cold-end tube bank will also be free of deposits. However, in some localized areas, deposition appears as shown in Figure 22 indicating the beginning of the fouling zone. Figure 23 shows the typical condition of the cold-end module as seen from the bottom. Throughout the module, deposition occurs mostly on the top side of the tubes due to the direct impact from the downward flowing flue gases and fly ash. The deposition appears to increase with depth as the flue gases flow through the tube bank and progressively contact colder heat pipes. Figure 24 shows the tube bank after cleaning. When clean, a light, placed below the cold-end module, can be seen (bottom center) through the eight-row deep tube bank.

Figure 21. Typical condition of heat pipe tubes in the top three modules.

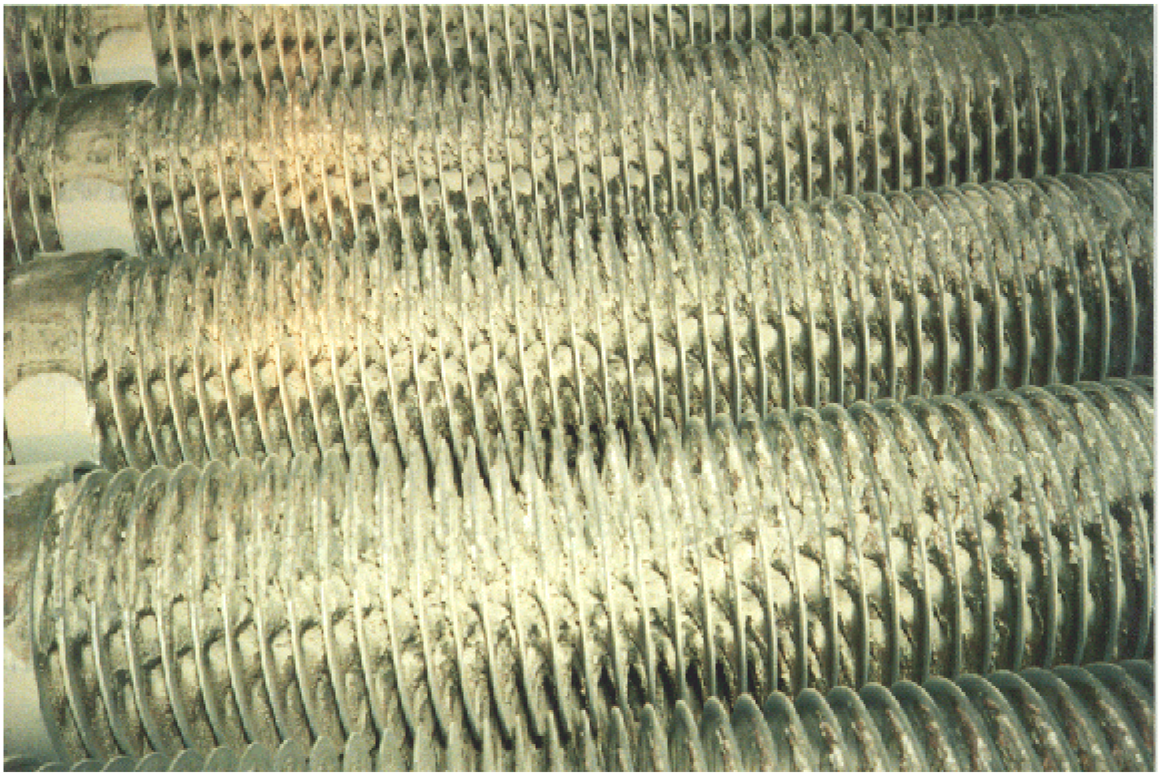

Figure 22. Fouled area on inlet flue gas side of the bottom cold-end heat pipe module.

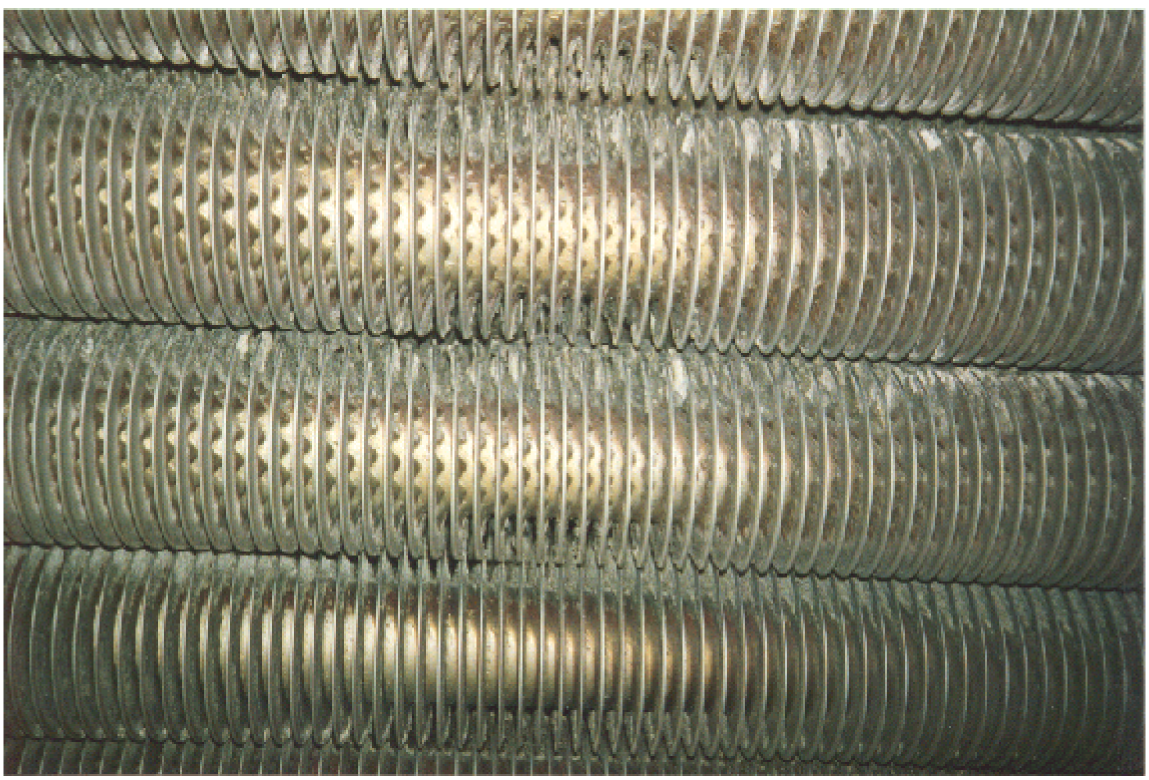

Figure 23. Bottom view of cold-end tube bank showing deposits on top side of tubes.

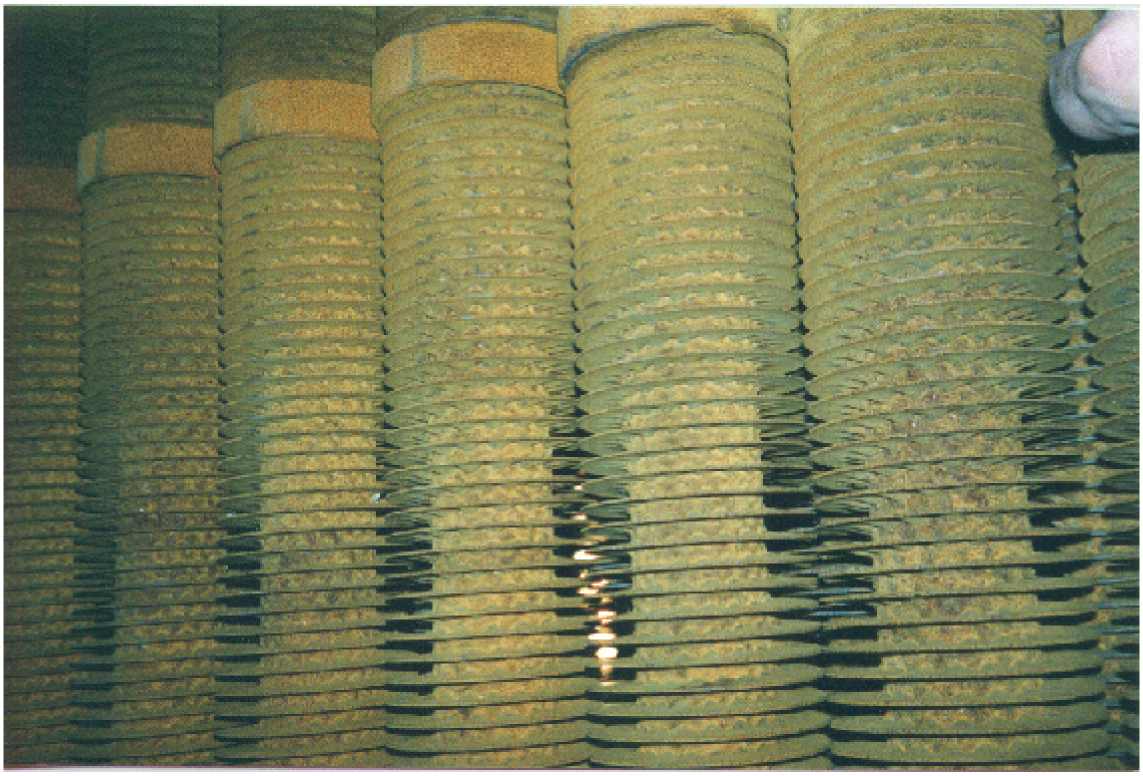

Figure 24. Top view of a cold-end tube module after cleaning -- note light showing through tubes.

Increasing flue gas side pressure drop and flue gas outlet temperature are signs of cold-end fouling. Figure 25 shows typical flue gas side pressure drops for the heat pipe air heaters under high load conditions. All pressure drops are corrected to a common basis. Breaks in the plotted data indicate the times when the Unit 2 boiler was off-line for maintenance. During these periods, the heat pipes were washed to remove the cold-end deposits. Figure 25 shows that for one or both heat pipes the flue gas side pressure drops generally increase to high levels in five to six months after cleaning. The figure also shows that the baseline pressure drop of about 4 in. WC is recovered following each full cleaning. Clearly, for the last four wash operations, equivalent cleanliness was achieved for the two heat pipes. The reader may note that the baseline pressure drops are slightly higher (0.1 in. WC - 0.5 in. WC) than the pressure drops reported for the performance tests. This is mostly due to differences in measuring equipment and pressure tap location for the plant process control system versus the special high accuracy taps and instrumentation used for the performance tests. The differences are not significant for the day-to-day system performance monitoring.

For the last operating period shown in Figure 25, the heat pipe pressure drops were better behaved and did not yet show the typical high pressure drop increase. This may be attributable to instituting a practice of not operating the boiler at less than 80 MW load and more attention to balancing heat pipe flows and temperatures. These operating practices help to avoid excessively low cold-end heat pipe temperatures which promote fouling.

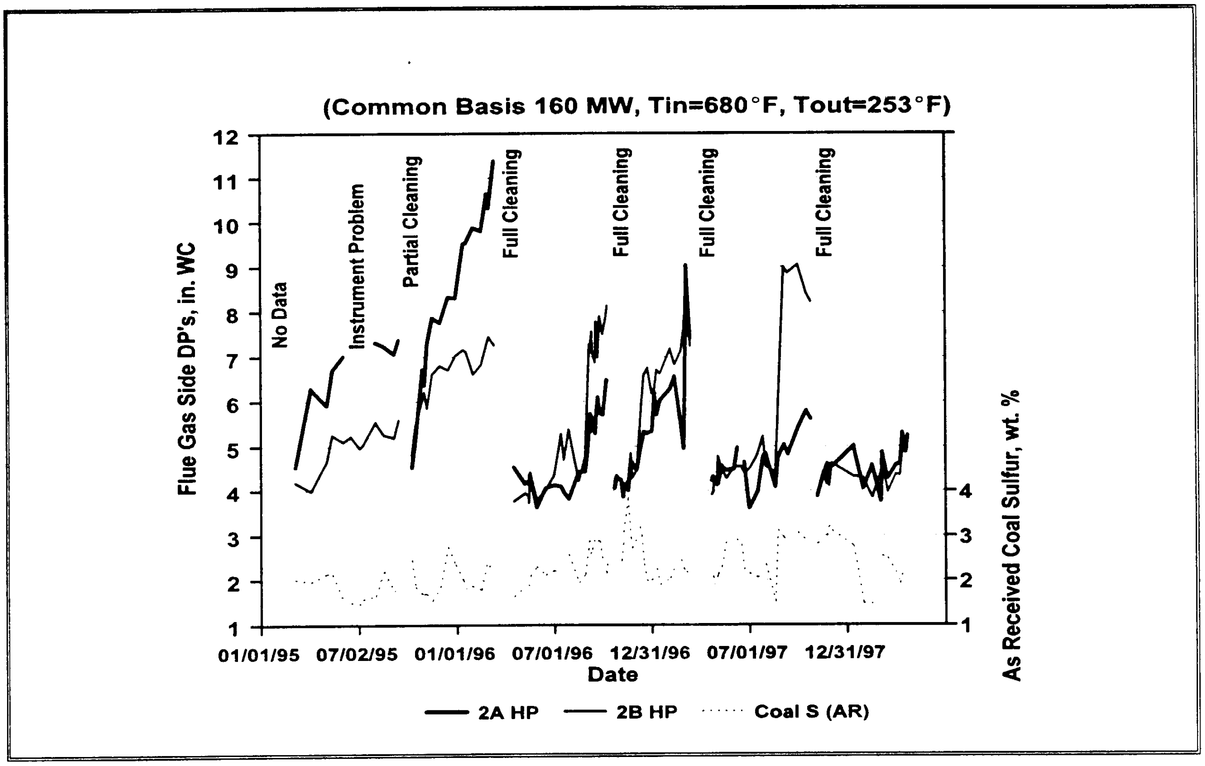

Figure 25. Flue gas side pressure drops and coal sulfur level 1995 to 1998.

5.6 Wash Methods

Heat pipe cleaning is simplified through the use of internal wash pipes incorporated in the original design. Each air heater is equipped with 18 stationary wash pipes for off-line cleaning. The wash pipes each consume 180 gpm of water at 75 psig. Figure 26 shows the layout of the wash pipes above the topmost heat pipe module. There are similar headers above each flue gas side module and each wash pipe is equipped with several nozzles.

When plant personnel first attempted to use the wash system, many nozzles were found to be plugged with fly ash deposits. Fly ash migration into the open nozzles during normal plant operation coupled with moisture and acid condensation caused the deposits. The problem is now avoided by operating the heat pipes with the nozzles removed and the nozzle connections capped. The nozzles are installed just before the heat pipes are to be washed. After washing, the nozzles are all removed and stored until needed again.

While the boiler is being taken out of service, the heat pipes are sootblown to remove as much fly ash and deposit material as possible. Washing of the heat pipes begins with the bottom module. The cold-end deposits are hard but are easily removed through a combination of deluge washing using the stationary wash headers and hand lancing. To facilitate the cleaning, the heat pipe high pressure air sootblowers are operated with the water sprays in service. This helps to loosen and break up deposits attached to the tubes and fins. After all modules have been cleaned by deluge washing, plant personnel inspect the modules. Cleanliness is determined by viewing, from the top side of a module, a light placed under the module. In this fashion the modules are inspected row by row.

Plugged areas are noted and manually cleaned by water jet using 1/4" tubing lances attached to fire hoses.

Figure 26. Wash pipes above top heat pipe module.

With experience, heat pipe washing has become more routine. Plant personnel have reported that heat pipe washing originally took seven days to complete and used more than 400,000 gallons of water. Currently, the heat pipes can be cleaned in 2.5-3 days with less than 200,000 gallons of water. The water usage is now similar to what is required for the Unit 1 Ljungstrom units.

5.7 Infrasonic Cleaner Testing

The original air sootblowers were not very effective in keeping the heat pipe cold-end tube modules free of deposits. Both the sequencing and frequency of sootblowing were changed without much success. To improve cleaning NYSEG decided to evaluate the Infrafone™ sonic cleaner on the 2A heat pipe. This device produces low-frequency (infrasonic) sound energy, which is introduced into a boiler or duct cavity. The sound energy acts within the flue gas volume and activates any fly ash or particulate in the gas stream, keeping it in motion and thereby inhibiting its accumulation on surfaces. The technology is used in Europe, Japan, and the United States in both oil-fired and coal-fired boilers to clean economizers and air preheaters.

Figure 27 shows the general equipment configuration of the AP-5000 Infrafone installation at Milliken. System design specifications are presented in Appendix I. The installation consists of a low frequency sound generator (pulsator), a resonance tube, and a rotary lobe type blower package which supplies motive air to the pulsator. The pulsator and resonance tube combination is connected to a ductwork port just blow the cold-end tube banks of the 2A heat pipe. To achieve the highest cleaning benefit, the decision was made to operate the Infrafone continuously rather than intermittently. Once the performance was established, the intent was to begin intermittent operation to establish the minimum required cleaning level.

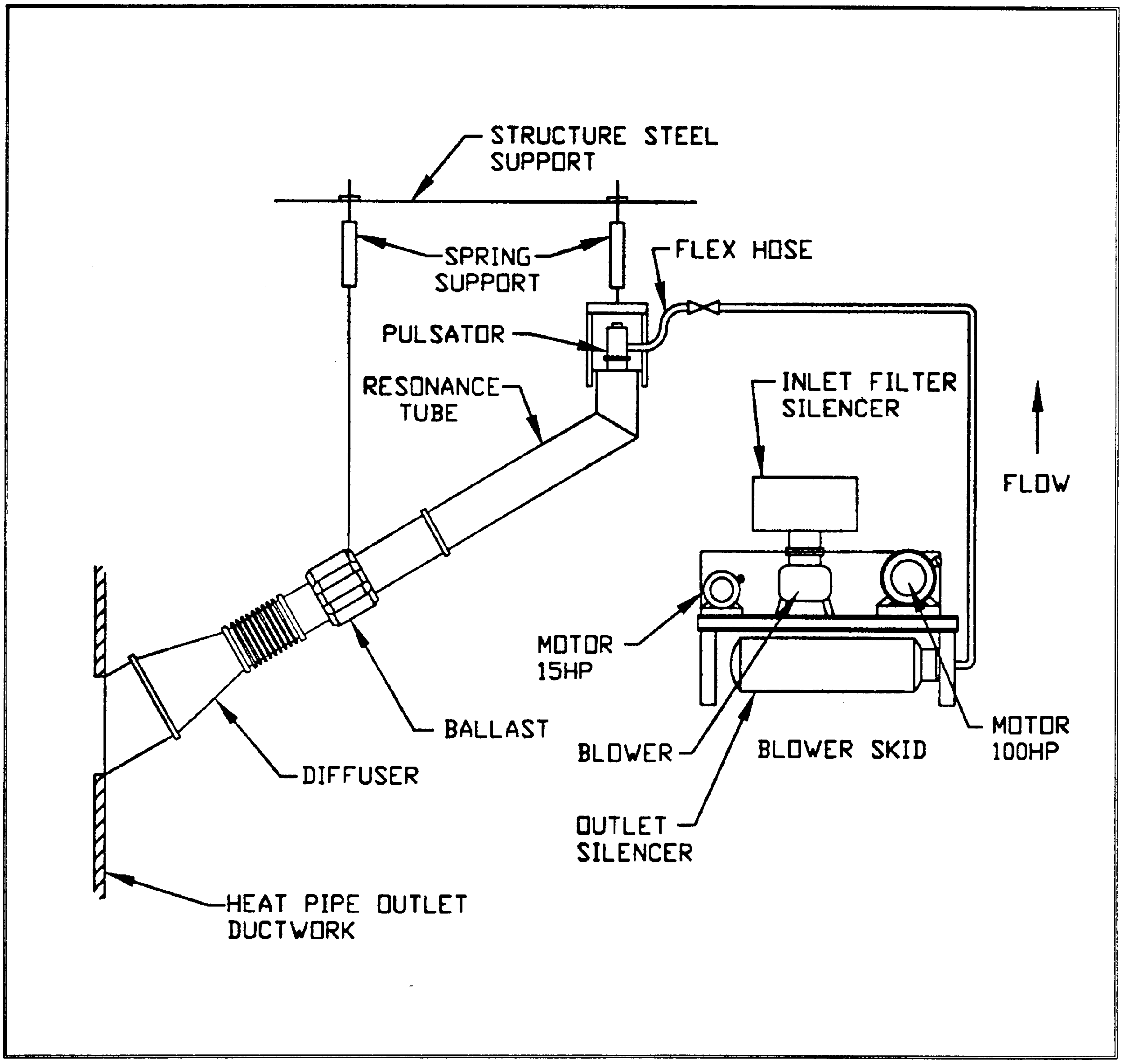

Figure 27. General layout of infrasonic cleaner system.

Although initial operations of the infrasonic cleaner indicated reduced fouling in the 2A heat pipe, subsequent operations without the cleaner in service indicate that the effect was marginal for the Milliken installation. These conclusions are supported by Figures 25 and 28 which show high boiler load, flue gas-side pressure drops for both heat pipes. Since tube fouling restricts the gas flow in the heat pipes, the flue gas side pressure drops adjusted to a common basis can be used to follow fouling. Figure 28 shows the pressure drop behavior after the Infrafone was first placed in service in early April 1996. Between April 17, 1996 to July 19, 1996, there is a slight decline in the 2A heat pipe pressure drop (4.5 in.WC dropping to 4.0 in.WC), indicating that the cleanliness of the 2A heat pipe was perhaps improving. At the same time, the common basis pressure drop for the 2B heat pipe increased from about 3.8 in.WC to 4.7 in.WC, indicating that the unit was experiencing increased fouling.

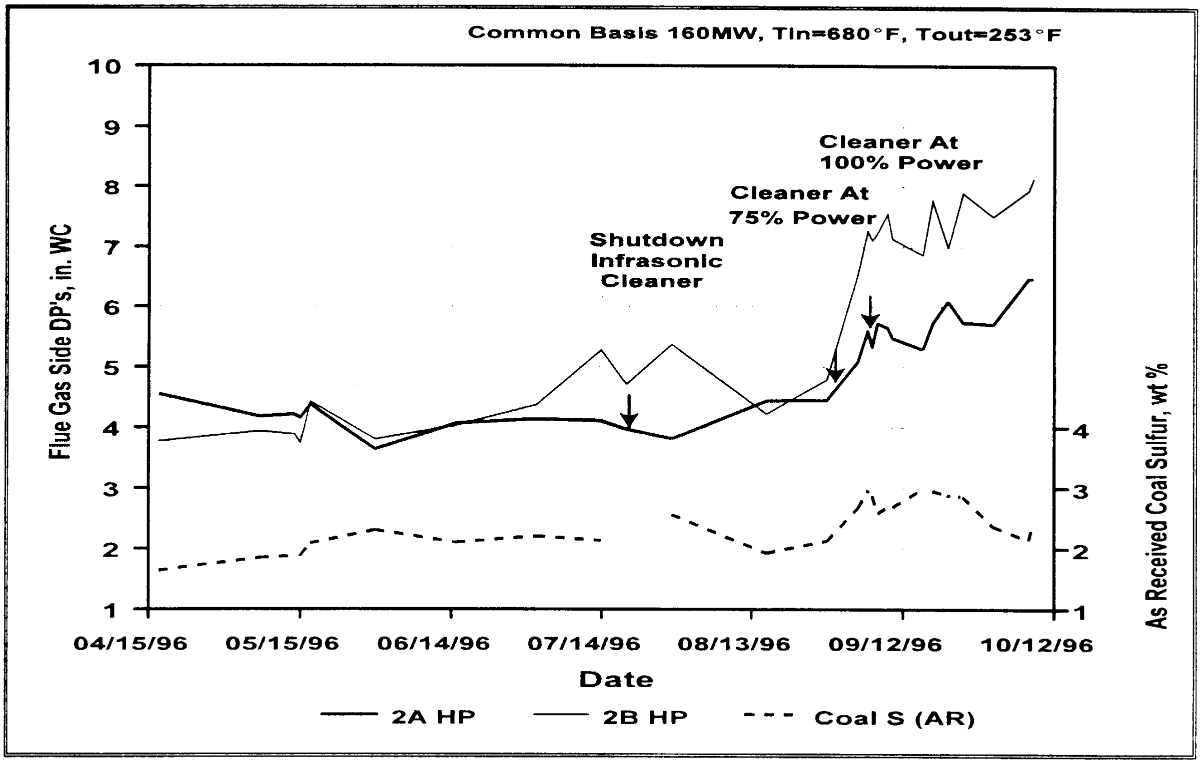

Figure 28. Flue gas side pressure drops and coal sulfur during first operation of Infrafone™.

On July 19, 1996 the Infrafone was shut down to allow repair of the ductwork between the 2A heat pipe and the particulate collector. Infrasonic resonance vibrations caused metal fatigue which produced cracks and holes in the ductwork. As shown in Figure 28, while the Infrafone was out of service, the 2A heat pipe pressure drop increased from 4.0 in. WC to 4.5 in.WC indicating increased fouling. However, for the same period, the 2B heat pipe pressure drop varied somewhat but overall remained essentially constant changing from 4.7 in.WC to 4.8 in.WC; indicating little or no significant fouling. When the Infrafone was placed back in service at 75% power, the pressure drops across both heat pipes began to rapidly rise. The renewed operation of the Infrafone may have suddenly loosened accumulated deposit and fly ash materials which then blocked flow channels as the materials traveled downward through the cold-end modules. Alternately, the change over of the plant fuel from a 1.8-2.2 wt. % S to a 2.8-3.0 wt.% S coal may have increased acid deposition in the cold-end modules causing the pressure drop rise. Unlike early in the run, operating the Infrafone at a 100% power level did not again reduce the pressure drop across the 2A heat pipe. When Unit 2 was shut down in October 1996 after six months of operation, the cold-end fouling in both heat pipes was by visual inspection, essentially the same.

Figure 25 provides additional evidence that the use of the infrasonic cleaner did not significantly reduce heat pipe fouling. The figure shows the heat pipe pressure drops on a common basis and the as received coal sulfur levels from March 1995 to April 1998. The first two periods show the behavior before the infrasonic cleaner was installed. The third and fourth periods are plant operating periods when the infrasonic cleaner was operated. The last two periods are with the infrasonic cleaner out of service. Fouling rates are relatively low for the first period (3/95 - 9/95) when coal sulfur levels were low at 1.8-2.1 wt. % S and the heat pipe thermal performance was degraded, due to gas generation in the naphthalene filled tubes which kept cold-end temperatures high. Between periods 1 and 2, repairs were made to the heat pipes to remove the noncondensable gases from the naphthalene tubes and the heat pipes were partially washed. The second period data shows that the partial washing was not adequate since fouling of both heat pipes was very rapid.

Between periods 2 and 3, the Infrafone was installed and special care was taken in cleaning the heat pipes. During periods 1 and 2, the 2A heat pipe fouled more quickly than the 2B unit. For periods 3 and 4 with the Infrafone operating, the 2A heat pipe fouled somewhat more slowly than the 2B heat pipe indicating that the infrasonic cleaner provided some benefit. This, however, now appears to be an operational artifact since the 2A heat pipe also fouled more slowly during period 5 when the Infrafone was not operated. For period 6, fouling rates appear to be about the same for both heat pipes. These results show that the Infrafone was not able to significantly improve on-line cleaning above what was achieved with the air sootblowers.

It is well known that dry ash particles and deposits are more readily activated by sound than are sticky particles or deposits. As previously mentioned in Section 5.5, the heat pipe fouling occurred in the cold-end modules and was associated with sulfuric acid condensation from the flue gases. In such a condensing environment, the Infrafone sonic cleaner was not expected to eliminate fouling, but rather to impede the rate of fouling and buildup. The Infrafone may work well in other utility boiler applications but its use does not appear to be of benefit in reducing heat pipe cold-end fouling.

Because of the penetrating nature of low frequency sound and the high acoustic energy levels used, any application of infrasonic cleaning technology must address excitation and possible resonance vibrations in equipment and structures. At Milliken when the Infrafone was operated, the concrete flooring below the Unit 2 precipitators vibrated enough that there was concern for failure due to a possible resonance situation. Structural dynamics and vibration studies determined that the vibration caused by the Infrafone was not detrimental to the structural integrity of the flooring. However, ductwork leading to the precipatators and the 2A heat pipe suffered damage caused by the intense low frequency sound. Figure 29 shows some of the metal cracking which occurred in the ductwork from the 2A heat pipe to the 2A precipitator. This vibration problem was cured by stiffening the ductwork. The ductwork (34' x 2.5' cross section) was internally stiffened using 4-inch diameter pipes welded, at several elevations, across the 2.5' ductwork width. Sixty stiffening pipes were installed. To provide some streamlining and reduce the gas flow turbulence around the stiffeners, 2"x2" angle iron strips were welded to both the leading and trailing edges of each pipe.

Figures 30 and 31 show some of the cracking which developed in the primary flue gas/secondary flue gas division walls inside the 2A heat pipe. Figure 32 shows how vibration of the side wall or possibly the finned tube caused 5/8" deep slots to be cut into a gas diversion plate in the 2A heat pipe

Figure 29. Vibration-damaged duct between the 2A heat pipe outlet and the precipitator inlet.

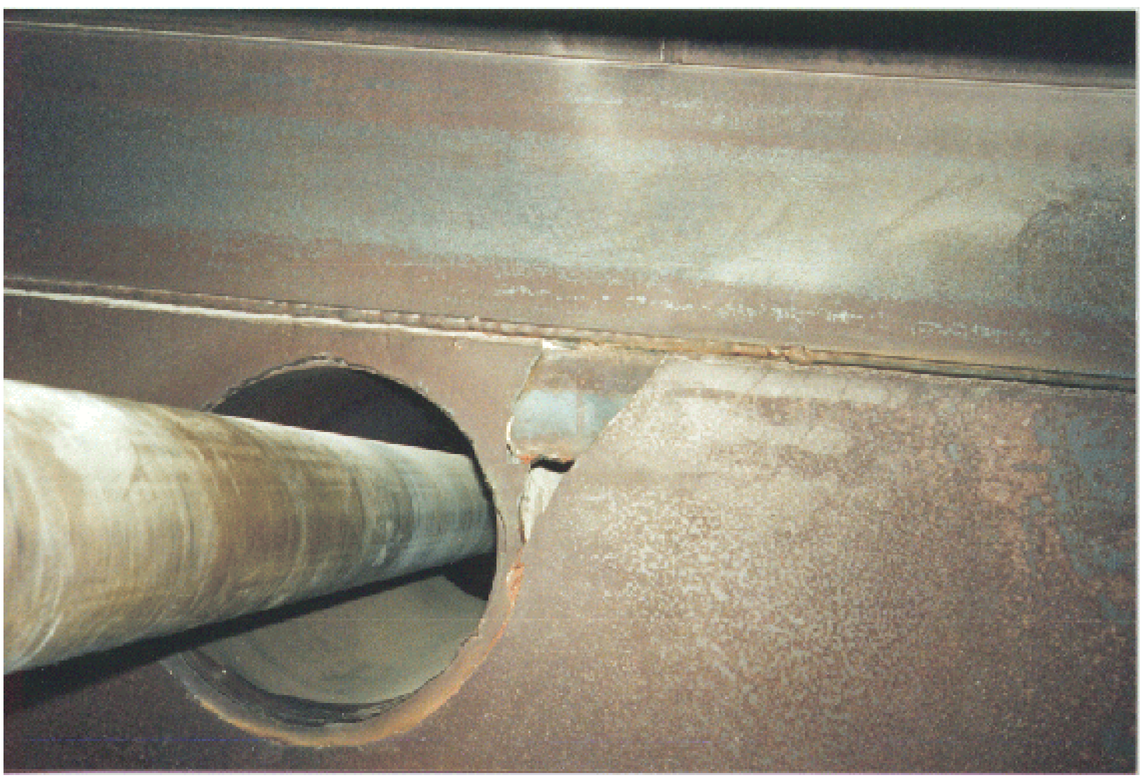

Figure 30. Vibration-caused wall cracking in the 2A heat pipe at an internal sootblower lance port.

Figure 31. Repaired cracks in the primary/secondary flue gas division wall of the 2A heat pipe .

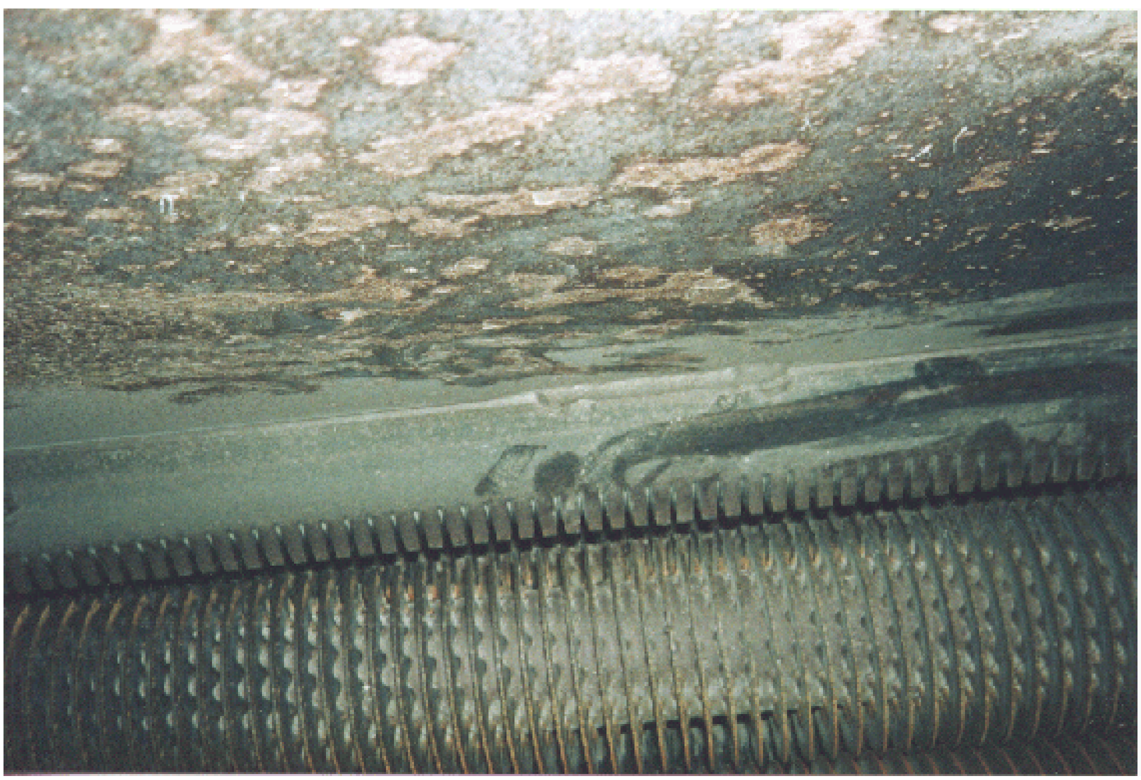

Figure 32. Vibration cut slots in gas diversion plate of 2A heat pipe.

cold-end module. After the internal damage was discovered, the decision was made to cease further operation of the infrasonic cleaner.

5.8 Sootblower Modifications

To compare the Infrafone operation with improved sootblowing, modifications were made to the row of sootblower lances located above the cold-end module (Module D see Figure 7) of the 2B heat pipe. The standard 1/2" diameter Bergemann cone nozzles were replaced with special 5/8" diameter diverging venturi nozzles (CFE nozzles) on four lances. This allowed the peak impact pressure at the heat pipes to be increased without consuming additional air.

The performance of the Infrafone against the modified sootblowers was discussed in the previous section. Since the Infrafone was not operated during the last two plant operating periods (see Figure 25), a performance comparison can be made between the originally installed sootblowing lances in the 2A heat pipe and the modified lances in the 2B heat pipe. The heat pipe pressure drop data shown in Figure 25 do not indicate any significant benefit of using the CFE nozzles. Toward the end of the fifth operating period, the 2B heat pipe pressure drop actually rose more rapidly than for the 2A heat pipe, indicating possible poorer performance. However, for the sixth operating period, the level of cleaning appears to be about the same for both heat pipes. These results do not show improved cleaning performance for the Milliken application of the CFE nozzles.

6.0 HEAT PIPE AIR HEATER PERFORMANCE BENEFITS

6.1 Thermal Performance Comparison with Rotary Air Heater

Originally the Milliken Units 1 and 2 were both equipped with rotary (Ljungstrom) air heaters. For the Milliken CCT-IV program, only the Unit 2 Ljungstroms were replaced with heat pipe air heaters. Since Milliken Units 1 and 2 are essentially identical units with identical capacities, conducting simultaneous detailed performance tests would be an ideal means of comparing the performance of the two air heater system designs. However, this was not part of the Milliken test program. It would have doubled the detailed testing costs and would have required installation of many new test ports around the Unit 1 air heaters.

To compare the heat pipe air heater thermal performance against the Ljungstrom system, data for ESP performance tests conducted in 1994[5] were used. The ESP data included coal analyses, and ESP inlet ductwork (air heater outlet) pitot traverses which provided average flue gas compositions and temperatures. The data did not include air heater inlet gas compositions and temperatures, so the total heat recovery in the air heaters could not be determined for the comparison. However, the data were sufficient to calculate the flue gas heat losses to the stack based on the ESP inlet conditions using the method shown in "Steam/its generation and use."[6] Assuming similar air heater inlet conditions, lower heat losses to the stack indicate improved thermal performance. Table 9 summarizes the stack heat losses. The detailed calculations can be found in Appendix J. The results indicate, that in a clean condition, the new heat pipe air heaters are just as thermally effective as the original Ljungstrom units. Both air heater systems leave as sensible heat about 9.9% of the energy contained in the fuel with the flue gases flowing to the FGD or stack.

6.2 Air Leak Reduction Benefits

The results presented in Table 9 also show that the excess air levels at the outlet of the Ljungstrom units were typically 12% to 24% higher than for the heat pipes. The higher excess air levels are due to air side-to-flue gas side leakage within the Ljungstrom air heaters. The leakage increases the power requirements for the primary air, secondary air, and induced draft (ID) fans and can result in more pumps being placed in service in the FGD. For Units 1 and 2, typical fan power data obtained at approximately one-month intervals between November 1996 and February 1998 are shown in Figure 33. The data are for stable operating periods when units 1 and 2 were operated together at approximately the same boiler loads with the economizer exit oxygen concentrations at similar levels (see Table 10).

Figure 33 shows that the Unit 1 fan amperages are significantly higher than the Unit 2 amperages under both low and high boiler load conditions. The differences are approximately 103 amps and 120 amps at boiler gross loads of 100 MW and 160 MW, respectively. Assuming a 0.90 power factor (PF) for the fan motors, the amperage differences are equivalent to 0.67 MW for the 100MW gross load operation and 0.78 MW for the 160 MW gross load operation.[1] These differences represent 0.67% and 0.49% of the low and high load gross power generations, respectively. The results indicate that the use of zero air leak designs, such as provided by the heat pipe air heaters, can provide small but significant performance improvements which can reduce power generation costs.

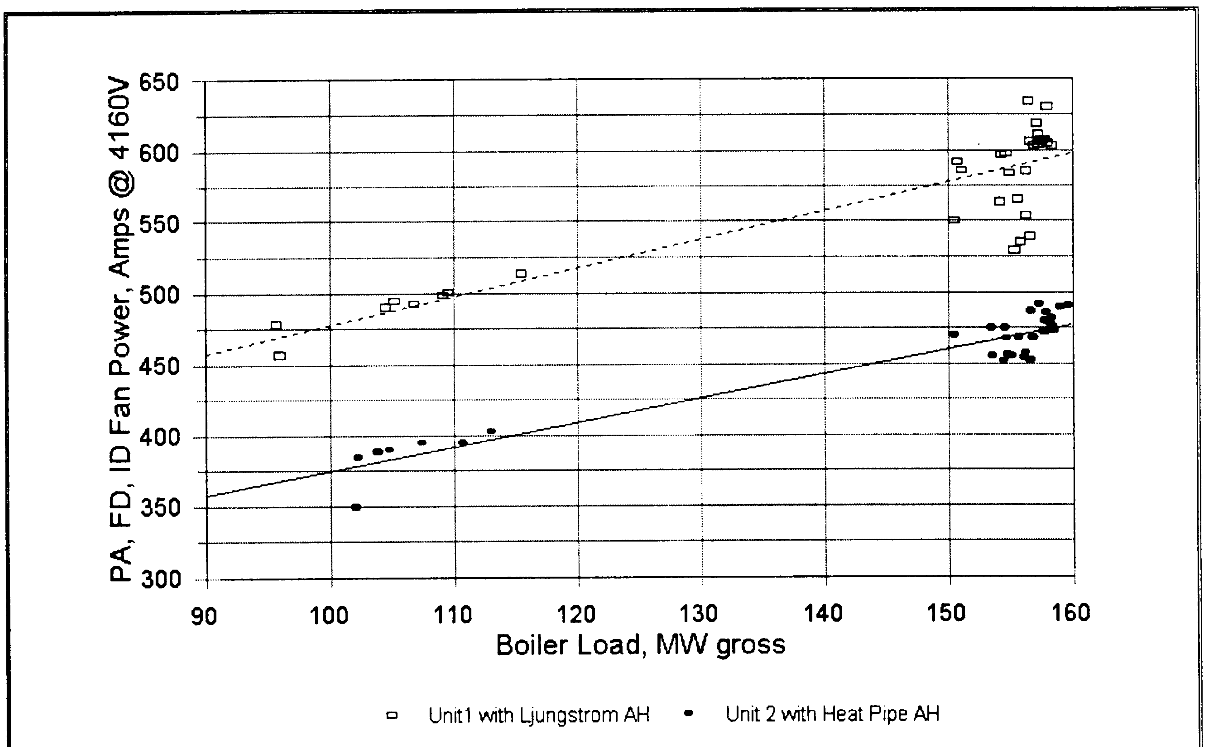

Figure 33. Fan power requirements for Milliken Units 1 & 2.

[1] $MW = \sqrt{3} \times PF \times Amps \times 4160V / 10^6$

Table 9

Stack Heat Loss Comparison for Ljungstrom and Heat Pipe Air Heaters

Milliken Unit 2 -- Full Boiler Load Operations

Air Heater Type	**Ljungstrom**		
Date	4/18/94	4/19/94	4/20/94
Flue Gas Temp, °F	264	266	258
Composition, mol. %			
O2	7.0	7.1	7.1
CO_2	12.5	12.4	12.4
N_2	80.5	80.5	80.5
% Excess Air	49	50	50
Stack Heat Loss (1)	9.90	10.09	9.77
Avg. Heat Loss to Stack		**9.92**	

Air Heater Type	**Heat Pipe**					
Date	10/17/95	10/18/95	5/14/96	5/15/96	11/7/96	11/8/96
Flue Gas Temp, °F	289	294	288	290	292	281
Composition, mol. %						
O2	5.8	5.8	4.4	4.6	4.6	4.9
CO_2	13.3	13.2	14.5	14.3	14.3	14.0
N_2	81.0	81.0	81.2	81.2	81.1	81.1
% Excess Air	37	37	26	27	27	30
Stack Heat Loss (1)	10.07	10.16	9.64	9.81	9.86	10.05
Avg. Heat Loss to Stack			**9.93**			

(1) Percent of Fuel Energy

Table 10
Performance Summary for Units 1 & 2

Date	Time	MW Gross		Economizer O2,		Fan Currents, Amps (1)		
M/D/Y	Hr:Min	Unit 1	Unit 2	Unit 1	Unit 2	Unit 1	Unit 2	Diff. (2)
11/8/96	06:00 - 08:00	151.1	158.4	(3)	3.2	585.9	475.6	-110.3
11/8/96	08:00 - 12:00	157.5	158.5	(3)	3.2	606.6	472.5	-134.1
11/8/96	12:00 - 16:00	154.2	157.7	(3)	3.2	596.6	471.9	-124.7
1/9/97	10:00 - 12:30	154.7	158.3	3.3	3.3	597.8	477.0	-120.8
1/9/97	16:00 - 20:30	156.6	157.8	3.3	3.4	606.3	479.4	-127.0
2/11/97	08:00 - 12:00	157.5	159.7	3.3	3.3	605.7	490.5	-115.2
2/11/97	12:00 - 15:00	157.8	159.1	3.3	3.3	607.3	489.6	-117.6
2/11/97	20:00 - 23:00	104.5	103.9	4.5	4.5	490.2	388.3	-101.9
3/1/97	03:00 - 05:00	157.3	157.4	3.3	3.3	610.9	491.7	-119.2
3/1/97	07:00 - 13:00	150.8	150.6	3.4	3.4	591.7	470.1	-121.7
3/1/97	14:00 - 24:00	109.0	107.5	4.4	4.5	498.8	394.8	-104.0
4/22/97	00:00 - 05:00	105.1	102.2	4.9	4.8	494.9	384.4	-110.5
4/22/97	07:00 - 14:00	157.2	156.8	3.6	3.3	618.6	467.7	-151.0
4/22/97	20:00 - 21:00	150.5	153.6	3.4	3.3	550.6	455.2	-95.4
5/3/97	04.00 - 07:00	115.5	113.0	4.2	4.6	514.1	402.8	-111.3
5/3/97	08:00 - 12:00	156.5	154.6	3.3	3.3	634.5	474.5	-159.9
5/3/97	19:00 - 21:00	109.5	110.8	4.4	4.4	500.5	394.8	-105.8
6/2/97	14:00 - 18:00	154.1	154.8	2.9	3.3	563.5	467.5	-96.0
6/2/97	18:00 - 21:00	156.2	155.7	3.4	3.3	585.8	467.9	-117.9
7/1/97	01:30 - 04:30	95.9	102.1	5.3	4.8	457.2	349.7	-107.5
8/1/97	00:00 - 05:30	106.7	104.8	4.2	4.6	492.7	390.0	-102.7
8/1/97	10:00 - 16:00	157.3	157.9	3.1	3.3	604.2	485.3	-119.0
8/1/97	19:00 - 22:00	156.9	157.9	3.0	3.3	602.8	471.6	-131.3
9/4/97	11:00 - 16:00	158.4	156.7	3.4	3.3	603.2	486.3	-116.9
9/30/97	12:00 - 13:30	Off Line	153.5	Off	2.9	Off Line	474.6	
11/5/97	08:00 - 18:00	156.5	158.0	3.3	3.3	598.2	483.1	-115.2
12/2/97	08:00 - 16:00	157.9	158.4	3.3	3.3	630.6	476.2	-154.4
1/30/98	06:00 - 08:00	154.9	156.3	3.4	3.0	583.9	457.5	-126.4
1/30/98	08:00 - 12:00	155.6	156.1	3.3	3.0	565.5	454.4	-111.2
1/30/98	12:00 - 16:00	156.2	156.6	3.3	3.0	554.2	452.5	-101.7
2/16/98	01:00 - 03:00	156.5	155.1	3.1	3.2	539.1	455.0	-84.1
2/16/98	04:00 - 06:00	155.8	154.8	3.1	3.3	535.0	456.2	-78.8
2/16/98	18:00 - 20:00	155.3	154.5	3.1	3.2	529.2	451.7	-77.5

(1) For Primary Air, Forced Draft and Induced Draft Fan Amps at 4160V.
(2) Unit 2 - Unit 1 Value.
(3) Problem With Analyzer Signal.

7.0 PERFORMANCE MONITORING

7.1 On-Line Monitoring

The Milliken Station boiler and FGD systems are controlled using the Westinghouse Distributed Processing Family (WDPF™) distributed control system. Control room operators communicate with field data processing units (DPUs) through CRT terminals to monitor and adjust process conditions.

The WDPF system automatically stores the process operating data on VAX-4300 computers. This provides historical data files and allows plant operators to view trend plots of most process variables. The basic configuration of the monitoring and control system is shown in Figure 34. The figure also shows the set up for remote monitoring which was used by CONSOL R&D to obtain daily operating data for process performance tracking and evaluation. Most of the data used by CONSOL to follow heat pipe performance changes were obtained using the remote monitoring system shown.

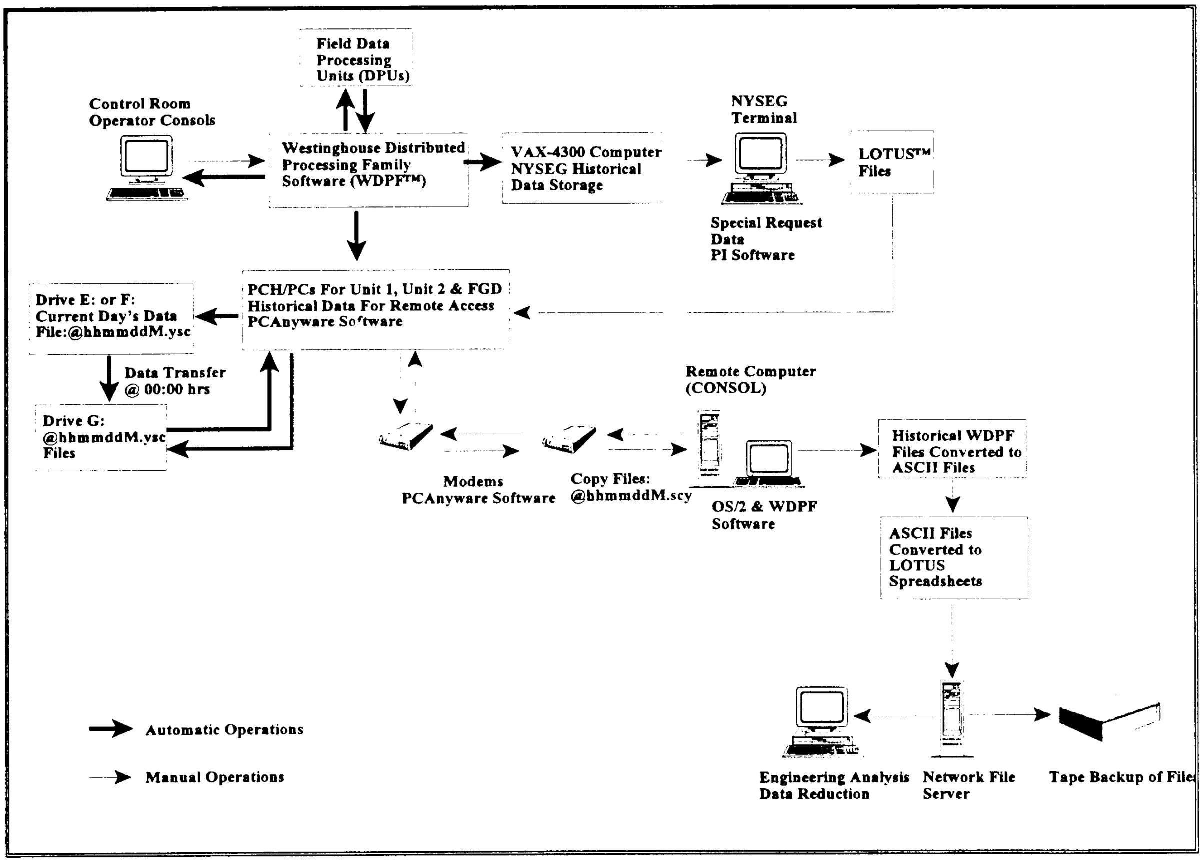

Figure 34. Milliken process control and data acquisition systems.

7.2 Remote Monitoring

Plant data are collected by the Westinghouse WDPF Data Acquisition System. This system includes three PCH/PC data collection personal computers (PCs). Separate PCH/PC units are interconnected to Unit 1, Unit 2, and FGD Unit. Each day, data from the WDPF system is accumulated in a separate file in the E: or F: drives of the PCH/PC computers. The file name, @hhmmddM.syc, identifies the data collection start times where:

hh	=	hour
mm	=	minute
dd	=	day
M	=	month (0 - 9, N, D)
y	=	year (0 - 9)

At 12:00 midnight, these files are transferred to the G: drives and new day files are started on the E: and F: drives. Files on the G: drive are then available for remote access and retrieval.

CONSOL retrieves data from the PCH/PC computers via telephone modem using pcAnywhere™ software. The historical WDPF day files are transferred to a remote computer in Library, Pa. Before the data can be evaluated, the WDPF day files which are in a proprietary format must be converted to ASCII format which is then used to generate Lotus™ files. The Lotus files are placed on a Network file server for access by engineers performing data analysis. Periodically the data files are backed up and archived on tape.

The PCH/PC computers store only a portion of the data collected by the WDPF system since not all the data are of interest in the performance evaluations of the boilers, air heaters and FGD system. Occasionally, data not stored on the PCH/PC computers must be retrieved. This is done by special request to the plant. The needed data are obtained from the plants VAX-4300 computer or from optical disk storage files. The data are converted to Lotus files at the plant which can then be forwarded to CONSOL or placed on the PCH/PC computer fixed drives for remote retrieval.

For a more detailed description of the remote data retrieval system refer to Appendix K.

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APPENDICES