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ENERGY

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CONSERVATION

DEMONSTRATION OF ENERGY CONSERVATION FOR
MULTI-DECK BOARD DRYERS

Phase I, Final Report

MASTER

February 8, 1980

Work Performed Under Contract No. AC03-78CS40167

Johns-Manville Sales Corporation
Research & Development Center
Denver, Colorado

U. S. DEPARTMENT OF ENERGY

Division of Industrial Energy Conservation

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DEPARTMENT OF ENERGY
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MASTER

DEMONSTRATION OF ENERGY
CONSERVATION FOR MULTI-DECK
BOARD DRYERS
PHASE I FINAL REPORT

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February 8, 1980

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EXECUTIVE SUMMARY

A study was made to determine the feasibility of recovery and reuse of low level heat from the exhausts of multi-deck dryers used to dry boards in the building materials industry. There are approximately 1,000 dryers of this type in the United States, most are heated by gas, and their exhaust gases are typically exhausted to the atmosphere with no heat recovery. These dryers are used in the manufacture of: (1) roof insulation board, (2) ceiling tile and panel stock, (3) wood fiber sheathing, (4) gypsum board, and (5) veneer plywood.

Design of heat recovery systems for such plants must consider the practical production problems caused by contaminants in the exhaust gases which potentially can foul heat exchange equipment. These contaminants usually include fibers which can hang up in the heat exchangers. The feasibility study was made at the roof insulation board plant operated by Johns-Manville at Rockdale, Illinois. The roof insulation board marketed under the tradename FESCO contain perlite, cellulose fibers, and asphalt. The dryer in this plant has eight

decks and processes boards twelve feet wide. There are five zones in the 461 foot long dryer. Nominal heat input to the dryer is 133 million BTU/hr. The Rockdale plant dryer exhausts contains small amounts of cellulose fiber, perlite and asphalt.

Pilot scale tests and analyses show that heat recovery designs utilizing several types of heat exchange equipment are feasible. These include the following:

1. Indirect contact air-to-air heat exchangers for preheating combustion air for the dryer furnaces.
2. Direct contact air-to-water heat exchangers using water sprays to heat process water.
3. Indirect contact air-to-liquid heat exchangers to heat recirculating liquid in a plant building heating system.

Periodic washing of the exhaust gas side of the indirect contact heat exchangers satisfactorily removes accumulated solids. The cleaning process appears simple and amenable to production processes.

The systems recommended for design and installation at the Rockdale plant include all three of the types of heat exchangers listed above. The preliminary estimate for the installed cost for these systems at the Rockdale plant is \$565,000 (1979 dollars). Annual heat recovery of 186,000 million BTU is projected with a value of \$545,000 using gas costs of \$3.00 per million BTU. Payback based on a discounted cash flow analysis using ten year depreciation is about two years.

Recommendations for preparation of detailed engineering design and cost estimates followed by installation and evaluation are made.

1.0 INTRODUCTION

A very substantial portion of the energy usage in the United States is related to industrial processes. Energy conservation in the private industrial sector is important to the Nation. Drying operations can be large consumers of energy. This is true for large multi-deck dryers which are widely used to dry board products such as (1) roof insulation board, (2) ceiling tile and panel stock, (3) wood fiber sheathing, (4) gypsum board and (5) veneer plywood. There are approximately 1,000 such dryers of this type in the United States today. Most utilize gas fired furnaces and the dryer exhaust is wasted to the atmosphere with no attempt to recover waste heat.

This report describes a study made to determine the feasibility of recovering and reusing heat from a large multi-deck dryer which is used in the manufacture of roof insulation board and is operated by Johns-Manville at Rockdale, Illinois. The dryer is the type used extensively in the building materials industry.

Under Department of Energy Contract DE-AC03-78CS40167 DoE, Johns-Manville has studied the potential for energy recovery and its uses in the manufacture of rigid mineral board thermal roof insulation at our Rockdale, Illinois plant. The scope of this phase of the contract was to make an energy audit of the operation to identify the quantity of recoverable energy and the possible areas of using this recovered energy, and to investigate various types of energy recovery equipment that could possibly be used. Johns-Manville subcontracted with the Dynatherm Corporation to study and evaluate air-to-air heat recovery systems.

The exhaust from the dryer contains a large quantity of recoverable energy in the form of low temperature energy. Approximately 3/4 of this exhausted energy is in the form of water vapor which can be extracted at temperatures well below 200°F. This is typical of dryers of this type. The exhaust for these dryers also contains particulates in the air stream which will have to be considered as a potential fouling factor when equipment is installed. These particulates, which include fibers, present a difficult problem when possible recovery of heat by use of heat exchangers is

considered. Fibers and other particulates tend to plug heat exchangers and thus reduce heat exchange efficiency.

In this study, both air-to-air type heat exchangers and air-to-liquid type heat exchangers (direct and indirect) were considered for recovery of heat from the dryer exhausts. Three potential uses for the recovered low temperature energy were identified. These were:

1. Preheating of combustion air for the dryer.
2. Heating of process water known as white water.
3. Heating building make-up air during the heating season.

The study indicates the recovery of energy from the dryer exhaust is feasible, and preliminary design and economic analysis indicates a successful installation should be cost effective. This assumes that heat exchanger performance obtained during test operations can be obtained on full scale. While the short-term testing of equipment reported herein indicates that heat recovery systems can be designed to operate satisfactorily, the question of recovery systems having

a decent service life and maintainability can only be answered by installing and running systems in this particular environment.

Recommendations are made that the Department of Energy proceed with Phases II and III of this contract to design, engineer and install, and evaluate the heat recovery systems at the Johns-Manville, Rockdale, Illinois plant.

2.0 TECHNICAL DISCUSSION

2.1 Process Description & Energy Audit

The Johns-Manville plant at Rockdale, Illinois, is a vertically integrated processing plant in which a roof insulation board is manufactured. The product is marketed under the trade name "FESCO". The major process steps include:

1. Perlite Expansion
2. Paper Slurry Preparation
3. Board Formation
4. Board Drying in a Multi-deck Dryer
5. Culling, Finishing, and Packaging

The first, second, and fourth process steps require large amounts of heat energy. Natural gas provides all of this energy. Perlite is expanded in vertical air columns operating at approximately 1600°F. Paper slurry

is prepared with newsprint and hot water. The board stock, consisting primarily of perlite, cellulose, and asphalt, is formed into a wet board on a Fourdrinier machine. The wet board, slightly more than 12 feet wide, is dried in a multi-deck hot air dryer operating in the range 400°F to 900°F. Heat from both the perlite expansion and board drying processes is dissipated directly to the atmosphere. No attempt is made to recycle energy from these exiting streams. A significant amount of heat energy is also used for building heating and scattered miscellaneous processes.

There is a large continuous electrical power demand for motive requirements and for lighting.

Extremely humid conditions prevail throughout most of the factory resulting in discomfort. Additional direct ventilation is not possible because of attendant additional, but intolerable, load on heating systems during colder months.

A summary of the average energy utilization of the Rockdale plant is shown in Figure 1. The current unit cost for process gas is \$2.79 MMBtu*, and for electricity \$30.20/Megawatt hour. Annual energy

*MMBTU = Million BTU

operating costs are shown in Table 1. In this table, a gas cost of \$3.00 MMBTU has been used.

The multi-deck, multi-zone roller dryer, which is the major user of energy (133 million Btu/hour) in the form of natural gas, is shown in Figure 2. The eight deck dryer is divided into five zones. The line of demarkation between zones is not exact as it is positioned by the air balance. Each zone is exhausted to outside air through individual stacks. In addition, the dryer has an unheated wet end seal region which is also exhausted to outside air. A detailed energy balance was made to determine input energy versus energy utilization by each dryer zone. The detailed energy balance is shown in Appendix A.

FIGURE 1

JOHNS-MANVILLE/ROCKDALE
ENERGY UTILIZATION SCHEMATIC
BASELINE SYSTEM

TABLE 1
ENERGY - ECONOMICS ANALYSIS
JOHNS-MANVILLE/ROCKDALE
BASELINE SYSTEM

PROCESS	FUEL	AVERAGE* CONSUMPTION RATE	ANNUAL* CONSUMPTION TOTAL	UNIT FUEL COST \$	ANNUAL* OPERATING COST
Perlite Expansion	Gas	30 MMBTU/HR	0.216 10^{12} BTU	3.00/MMBTU	\$ 648,000
Board Dryer	Gas	133 MMBTU/HR*	0.958 10^{12} BTU	3.00/MMBTU	\$2,874,000
Water Heating	Gas	8.4 MMBTU/HR	0.065 10^{12} BTU	3.00/MMBTU	\$ 195,000
Miscellaneous Process Heat	Gas	10 MMBTU/HR	0.072 10^{12} BTU	3.00/MMBTU	\$ 216,000
Space Heating	Gas	10 MMBTU/HR	0.055 10^{12} BTU	3.00/MMBTU	\$ 165,000
Motive Power - Light	Elec	2.5 MW	18000 MWHR	30.20/MWHR	\$ 544,000
TOTAL					\$4,693,000

*Six day week basis, 7200 hours/year (Full production requires 8000 hours/year operation).

Space heating based on 5500 hours/year.

MWHR = Megawatt Hour

MMBTU = Million BTU

Based on 1031 BTU/1000 cft. of gas used.

The average water content (average of entering and leaving) in Zone 1 is approximately 300 percent (on a dry basis), 200 percent in Zone 2, 100 percent in Zone 3, 50 percent in Zone 4, and 10 percent in Zone 5. As the board moves through the dryer, water and moisture are removed by impinging heated air against the moving board. The heated air temperatures vary from between 700°F to 800°F in Zones 1, 2, and 3; 580°F to 680°F in Zone 4; and 450°F to 500°F in Zone 5.

FIGURE 2
FESCO BOARD DRYER

2.2 Analysis of Seal and Stack Energy Losses

The analysis results of wet end seal and stack losses by zone are given in Table 2. The table was developed from the measured stack exhaust gas flow rates, dry bulb, and wet bulb temperatures. Also given in this table is the natural gas consumption for each zone.

The energy contained in the exhaust gas of each stack is partitioned into several categories for easy analysis. First, the energy content between the dry bulb and wet bulb temperatures is given for both the contained superheated moisture and air. (The dew point of the gases has not been used for the purpose of this report.) The sum of these energies represents the maximum energy that can be withdrawn from the exhaust gas without causing condensation. For example, the exhaust gas in the Zone 3 stack contains 9.73×10^6 Btu/hr. that could be extracted before condensation occurs. Table 2 shows that Zone 3 has the greatest energy which could be extracted without causing condensation.

ZONE	Exhaust	Natural Gas	DB	WB	Dry Air Rate	Water Rate	Energy to WB		** Energy of Condensation	* Energy of Air to 70°F	Total Energy
	ACFM	CFH	°F	°F	lb/hr	lb/hr	Steam	Air			
							10 ⁶ Btu/hr	10 ⁶ Btu/hr	10 ⁶ Btu/hr	10 ⁶ Btu/hr	10 ⁶ Btu/hr
Seal	10,000	---	222	152	26,520	5,226	0.16	0.45	5.26	0.52	6.39
1	23,920	38,710	354	168	46,488	14,616	1.25	2.08	14.58	1.11	19.02
2	21,490	34,740	352	180	33,648	18,282	1.45	1.39	18.10	0.89	21.83
3	58,010	21,640	412	168	107,538	31,668	3.42	6.31	31.59	2.53	43.85
4	32,440	19,820	240	126	99,822	6,570	0.34	2.74	6.72	1.34	11.14
5	35,940	14,430	272	130	105,000	7,434	0.49	3.59	7.58	1.52	13.18
TOTAL	---	129,340	---	---	419,016	83,796	7.11	16.56	83.83	7.91	115.41

*Assumes all moisture has been condensed out of air at WB temperature

**Heat content of condensed water between WB and 70°F has been neglected

TABLE 2

ANALYSIS OF SEAL AND STACK LOSSES FOR FESCO BOARD DRYER

The condensation energy (assuming all condensation occurs at the wet bulb temperature) is also given in Table 2. Note that the water content of the exhaust gas consists of combustion water, combustion air water, and water evaporated from the board.

Finally, the energy content of the remaining exhaust air between the wet bulb temperature and 70°F is given in the table. The energy content in this category is the smallest of the three basic energy categories.

The total energy contained (as referenced to 70°F, zero humidity air) in the exhaust gases in each zone is summarized in Table 2. This total energy for all zones and the wet end seal must be less than the energy put into the dryer by natural gas heating. The average heating value for natural gas in this region of the country is 1031 Btu/ft³. As shown in Table 1 the input energy is 133.4×10^6 Btu/hr. compared with the calculated exhaust energy of 115.4×10^6 Btu/hr.

Separately, the energy balance on the dryer gives 79,670 lbs./hr. as the water evaporated. The total water in all exhausts (Table 2) is 83,796 lbs./hr. When this is corrected for combustion water and combustion

air water (70°F, 65 percent relative humidity), the calculated water evaporated is 67,250 lbs./hr. This does not, however, take the moisture leaks from the dryer doors, etc., into consideration.

The moisture content of the board entering the dryer is in excess of 300 percent (Zone 1) and exits the dryer (Zone 5) as virtually dry board. In each zone, hot gases impinge on the board surface and remove moisture and organic constituents by evaporation. In addition, particulates are removed by erosion. Except for about the first 70 feet into the dryer and for the last 70 feet prior to exiting the dryer, the moisture content of the board is given by:

$$\text{Percent Moisture} = 102.8603 - 0.0046x$$

where x is the distance in feet that the mat has progressed into the dryer. This is consistent with the drying regime controlled by liquid diffusion. The absolute amount of moisture removed per zone progressively increases, maximizes in Zone 3 and, thereafter, decreases and is a minimum in Zone 5.

Aside from zone interaction caused by differences in static pressure between dryer zones, airborne contamination in Zone 1 should be a minimum because sufficient moisture is present in the wet mat to provide capillary forces needed to inhibit solids erosion and because excess moisture shields volatile organics from the heated gases. It is concluded, therefore, that a heat exchanger located in the Zone 1 exhaust stream will be the easiest one to keep clean.

As noted in Table 2, 73 percent of the energy exhausted from the dryer comes from Zone 1, 2, and 3. As reported in Section 5, heat exchanger testing of the Zone 3 exhaust showed evidence not only of solids carryover but also of asphalt volatiles. The solids can be prevented from interfering with proper heat exchanger operation by providing an inplace water wash cleaning system. Condensed asphalt volatiles on heat transfer surfaces may require periodic removal and solvent cleaning of heat exchanger modules. In order to minimize volatile carryover to heat transfer surfaces, moisture condensation in Zone 3 exhaust heat exchangers is not permitted. Within the above restrictions, Zone 3 heat exchangers appear practical.

The heat exchangers in Zones 1 and 2 can be operated with or without moisture condensation. Inspection of Table 2 shows that when the heat exchangers are operated with moisture condensation, the amount of energy that can be recovered is much greater than without condensation. The energy needed to heat white water from 95°F to 155°F is 12×10^6 Btu/hr. Both Zones 1 and 2 contain sufficient exhaust energy to heat white water if condensation is permitted.

2.3 Process Heat Recovery Options

The analysis of the energy losses and heat recovery potential plus the data obtained during the pilot tests indicates that it is feasible to recover heat from multi-deck dryer exhausts.

The uses for this recovered heat which is readily available are the following:

1. Preheat combustion air to the burners for Zones 1, 2 and 3. Preheating of combustion air for Zones 4 and 5 is not considered at this time because, unlike the first three zones, the burners on those zones

do not obtain 100 percent of their combustion air requirement from the outside.

2. Preheating white water to the hydropulper: The requirement for heated white water is 400 gpm.

3. Heating of building make-up air during the cold season. A need for about 100,000 SCFM of heated air exists in the plant.

Since Zone 3 stack contains the greatest energy available for extraction without causing condensation (Table 2), it offers the best alternative for preheating of the combustion air. The preheating of white water or building make-up air heating can be accomplished by using the exhaust air from either Zone 1 or Zone 2. It is also possible to use Zone 2 exhaust for preheating white water as well as heating building air. The latter provides the advantages of lower installation and maintenance costs.

A schematic of the options for heat exchanger locations and the purposes for which the recovered energy is used, is shown in Figure 3. Alternate locations for both building heating and white water

preheating heat exchangers are indicated in Figure 3.

The total energy recovered may be as high as 26.5×10^6 Btu/hr. in the building heating season and as low as 20.7×10^6 Btu/hr. in the summertime if only combustion air for Zones 1, 2, and 3 is preheated. In both of these cases, 400 gpm of white water is increased in temperature by 60°F.

FIGURE 3
 SCHEMATICS OF VARIOUS RECOVERY OPTIONS

(Continued on Page 24)

Expansion Tank

Option 4
Heat recirculating liquid which is used to heat building air.

Expansion Tank

Option 5
Heat recirculating liquid which is used to heat building air.

Expansion Tank

Option 6
Heat white water plus heat recirculating liquid which is used to heat building air.

Figure 3 (Cont.)
SCHEMATICS OF VARIOUS RECOVERY OPTIONS

2.4 Heat Exchanger Design Specifications for the Various Options

2.4.1 Specifications for Air-to-Air Heat Exchangers

The air-to-air heat exchangers will be used to recover energy from the Zone 3 exhaust and preheat combustion air for Zones 1, 2, and 3 (Option 1).

The exhaust side specifications for Option 1 are:

Mass flow Dry Air (lbs/hr)	108,000
Mass Flow Moisture (lbs/hr)	32,000
Exhaust Temperature (°F)	412
Maximum Pressure Drop (inches of water)	1.5
Design Temperature (°F)	500
Exhaust Stream Contaminants	See Text

The supply (combustion air) side specifications for Option 1 are:

Mass Flow Air (lbs/hr)	138,000
Supply Temperature (°F)	70
Relative Humidity (%)	65
Maximum Pressure Drop (inches of water)	1.5
Supply Stream Contaminants	None

With regard to the specification for exhaust stream contaminants, from testing in Zone 3 exhaust (see Section 5), it is estimated that 4 lbs/hr of solids and

much smaller quantities of organic volatiles (as both vapors and solids) are present in the airstream. Undoubtedly, a major constituent of the organic volatiles is asphalt. The exhaust gases do not appear to be corrosive to either aluminum or steel. The basis of this information is actual tests on Zone 3 exhausts as described in Section 5.

Testing (Section 5) has demonstrated that the solids may also be removed from heat transfer surfaces by water spray. To be effective, the heat exchanger must be oriented horizontally and the water spray must be directed downward onto the heat transfer surfaces. It is advisable to locate spray nozzles between sections or modules of the exhaust side for maximum effectiveness. Marshall Winstead of Piedmont Energy Systems, Inc., advises that, during plant shutdown, a well design spray system can be used to further wash down the heat exchanger by introducing detergents or industrial cleaning chemicals into the spray water.

2.4.2 Specifications for Air-To-Liquid Heat Exchangers

The air-to-liquid heat exchangers will be employed to recover energy from Zones 1 and/or 2 for preheating white water and heating clean liquid for use in a runaround system for building heating. The energy required for preheating white water is 12×10^6 Btu/hr. For supplying building heating, the required energy (average) is 5.84×10^6 Btu/hr. The building heating requirement is based on supplying 100,000 SCFM of heated air.

The exhaust side specifications for Options 2 and 4 (stack exhaust from Zone 1) are:

Mass Flow Dry Air (lbs/hr)	46,500
Mass Flow Moisture (lbs/hr)	14,600
Exhaust Temperature (°F)	354
Maximum Pressure Drop (inches of water)	1.0
Design Temperature (°F)	400
Exhaust Stream Contaminants	Less than Zone 2

The white water side specifications for Options 2 and 3 are:

White Water Flow (gpm)	400
Inlet Water Temperature (°F)	95
Outlet Water Temperature (°F)	155
Design Pressure (psi)	250
White Water Contaminants	0.3% solids by weight
Maximum Pressure Drop (psi)	5

The exhaust side specifications for Options 3 and 5 (stack exhaust from Zone 2) are:

Mass Flow Dry Air (lbs/hr)	33,700
Mass Flow Moisture (lbs/hr)	18,300
Exhaust Temperature (°F)	352
Maximum Pressure Drop (inches of water)	1.0
Design Temperature (°F)	400
Exhaust Stream Contaminants	Less than Zone 3

The liquid side specifications for Options 4 and 5 are as follows. The liquid is used in a recirculating system for building heating.

Liquid (water-glycol) Flow (gpm)	250
Inlet Liquid Temperature (°F)	72
Outlet Liquid Temperature (°F)	150
Design Pressure (psi)	250
Maximum Pressure Drop (psi)	5
Liquid Contaminants	None

2.5 Types of Heat Exchanger Equipment Considered

Various types of air-to-air and air-to-liquid heat exchangers are manufactured in the United States (Reference 1). These may be broadly classified as regenerative, conduction-recuperative, or transport recuperative according to the principle of heat transfer involved. Of these, the following may be considered for this dryer application:

Rotary Wheel Exchanger: Sometimes referred to as the Ljungstrom regenerative heat exchanger, the enthalpy wheel, laminar wheel, heat wheel, and ceramic wheel. Normally used in counterflow airstreams but may be used in parallel flow airstreams with reduced efficiency (effectiveness). Basically, the device operates by rotating metal or ceramic material into the hot airstream where it is raised in temperature thereby extracting energy from the gases and then rotating the heated material into the cold airstream where it is lowered in temperature thereby imparting energy to the cold gases flowing thereover (see Figure 4). This is a regenerative type of heat exchanger.

Fixed Plate Exchangers: Manufactured in various configurations and sizes for crossflow and counterflow airstreams. Fins in the channels are oftentimes employed to increase the exchanger effectiveness. Unique modular designs are manufactured which permit disassembly for inspection and cleaning. In these devices, thermal energy is transferred from one airstream to the other by ordinary thermal conduction through the heat exchanger plates separating the two airstreams. Examples of this type of heat exchanger include simple plate and shell arrangements as shown in Figures 5, 6, and 7.

Tube and Shell Exchangers: Initially, these exchangers were employed in power plant feedwater heating in 1880. The round shell suited these exchangers for high pressure applications. Since then, these exchangers have been employed in air-to-air heat transfer service in power plants and in certain petrochemical processes. For low pressure service, the shell is made into a box shape instead of a cylinder. The clean air-side is external to the tubes and the contaminated airside is internal to the tubes. This arrangement is easier to clean. Counterflow and extended surfaces internal to the tubes are employed to improve effectiveness. In applications for preheating combustion air in power plants, a steel shot dry cleaning system is employed for internal cleaning of the tubes. A tube and shell arrangement is shown in Figure 8.

Heat Pipe Exchangers: Used for a century as Perkins tubes for heating baking ovens. Recent heat transfer technology advances and the advent of extended air-side surfaces make it possible to design compact modularized air-to-air exchangers. This concept can be used where the airstreams can be brought into close proximity. This is a transport recuperative heat exchanger utilizing a working fluid as the transport medium between the hot and cold sides. Figure 9 shows such an arrangement.

Conventional Water Coil Exchanger: This is an air-to-liquid heat exchanger which, when combined with a second unit in a runaround arrangement, results in an air-to-liquid-to-air exchanger system. This system is popular in Europe for transferring thermal energy from one process location to another process location. For temperature applications up to 800°F, special high temperature heat transfer fluids are employed. The coils employ extended surfaces to improve airside heat transfer. The most popular types are tension wound spiral fins pioneered by Aerofin Corporation and plate fins pioneered by the York Division of Borg-Warner Corporation. °F or temperatures above 500°F, the fins are metallurgically integral with the tubing. Figure 10 shows such an arrangement.

Air-To-Water Direct Spray Economizer: Direct spray economizers are widely used in the paper industry for heating process water. High humidity, low temperature exhaust gases contain a large quantity of heat in the form of latent heat of vaporization. By spraying water directly into a high humidity air flow, the water vapor is condensed releasing its latent heat which can be utilized for heating the water that is sprayed into the airstream. Figure 11 shows such an arrangement.

FIGURE 4

LJUNGSTROM REGENERATIVE HEAT EXCHANGER

FIGURE 5

PLATE AND SHELL RECUPERATOR

FIGURE 6
PLATE AND SHELL RECUPERATOR

FIGURE 7
PLATE AND FIN HEAT EXCHANGER

FIGURE 8
TUBE AND SHELL AIR HEATER

FIGURE 9
STATIC HEAT PIPE HEAT EXCHANGER

FIGURE 10

RUNAROUND RECUPERATOR

FIGURE 11

DIRECT SPRAY ECONOMIZER

2.6 Conclusions From Heat Exchanger Testing

Pilot scale heat exchanger equipment was evaluated at the Rockdale plant where actual dryer exhaust streams could be used. The purpose of conducting pilot tests with the various types of heat exchangers was to evaluate heat exchanger performance when operated in a contaminated exhaust airstream and to determine design parameters to overcome fouling and provide better maintenance. This section is a summary of the results. A detailed description of the pilot scale testing is given in Section 5.

The following types of heat recovery equipment were tested:

- a) Rotary wheel exchanger obtained from the Wing Corporation.
- b) Heat pipe exchanger obtained from the Dynatherm Corporation.
- c) Direct water spray economizer obtained from the AER Corporation.

The rotary wheel and heat pipe exchangers are air-to-air heat exchangers. The economizer is a direct contact unit using a water spray. The air-to-air units were tested on the Zone 3 exhaust whereas the economizer was tested on Zone 2 and 3 exhausts.

The following conclusions were arrived at on the basis of these tests:

1. Heat recovery is feasible from the exhausts on a multi-deck board dryer.
2. An air-to-air heat exchanger can be designed to operate in a contaminated airstream if intermittent in-place washing is provided.
3. Air-to-air heat exchangers should be designed to operate at temperatures above the condensation temperature of the exhaust gases.
4. The solids can be eliminated from the exhaust airstream by using fine mesh filters but the frequency of replacement of filters makes it operationally impractical. Inertial baffles requiring little

maintenance could be used to remove the larger particles.

5. The particulate contaminants are easily removed by water sprays. The heat exchangers should be designed for easy accessibility, disassembly and reassembly, and for solvent cleaning, if required.

6. The direct spray economizer is a very effective unit for heating white water. It is not affected by the contaminants in the airstream and does a good job of washing out the particulate in the exhaust airstream. Thus, any heat exchanger installed after the spray unit will operate in a relatively clean airstream.

7. The spray nozzles used in the direct spray unit should have orifices of not less than 3/4 inch to prevent plugging from the solids in the white water. All piping should be not less than 1 inch I.D.

2.7 Evaluation of Air-To-Air Heat Exchangers

The selection of the proper fouling factor to be used for heat transfer design is always controversial. For a 0.01 inch buildup of asphalt on the heat transfer

surfaces, the calculated fouling factor is 0.0019 hr-ft²-°F/Btu. This amount of fouling will reduce the efficiency (effectiveness) of the heat exchanger by approximately 0.3 percent. Fouling factors as high as 0.006 have been used in liquid asphalt heat exchanger designs. This latter fouling factor will reduce heat exchanger efficiency by approximately 1 percent.

Although the heat exchanger performance is not very sensitive to fouling, it is not possible to estimate the rate of fouling at this time. Contacts at the Asphalt Institute inform us that vapor pressures of various asphalts are not available. In view of this, the procurement specification for this heat exchanger should call for easy disassembly and reassembly for purposes of solvent cleaning. The maximum downtime allowed for cleaning should not exceed eight hours.

As a result of the test work discussed in Section 5, the heat wheel, tube and shell, and heat pipe exchangers were selected as viable engineering alternatives for use in Zone 3 (Option 1) air-to-air heat recovery. Because of the large amount of solids contamination in Zone 3, exchanger designs employing

long and narrow passages were eliminated. The heat wheel was considered because it can be purged during every revolution. The tube and shell was considered because, although the tubes containing the contaminated gases are long, the tube ID is large and the inside of the tubes can be easily cleaned. The heat pipe heat exchanger was considered because the design can be modularized for easy cleaning either in place or external to the installation.

2.7.1 Rotary Wheel Exchangers (Option 1)

The Wing Company, manufacturers of heat transfer equipment for over 100 years, was requested to supply a quotation for energy recovery wheels for Option 1 (Specification, Section 2.4). Two of their Cormed WCM-2150 units are required. The characteristics of each unit and the system performance of both units are:

Unit Overall Dimensions	
(L x W in feet)	9.38 x 9.33
Unit Depth (feet)	2.67
Unit Weight (pounds)	3,050
Maximum Exhaust Temperature (°F)	750
% Supply Air for Purging	10
Efficiency (Effectiveness), (%)	64.5
Pressure Drop (inch of water)	0.5
Cross-Contamination (%)	0.1

The manufacturer states that these units are capable of passing particles in sizes up to 1,000 microns (39 mils) and that the purging system will reject larger particles to the exhaust stream.

The selected rotary wheel exchangers will recover 8.7×10^6 Btu/hr and preheat combustion air for Zones 1, 2, and 3 by 264°F.

2.7.2 Tube and Shell Exchangers (Option 1)

There are many manufacturers of tube and shell heat exchangers offering a wide variety of product configurations. The Heavy Duty design of industrial heat recovery exchangers manufactured by Allied Air Products Company, Inc., was selected for evaluation. These exchangers employ 18 gauge, 1.5 inch diameter, 20 foot long tubes through which the exhaust gases can be passed. These large diameter tubes are less subject to fouling by the exhaust gas solids contaminants than are various competitive designs. In order to meet the specification requirements of Section 2.4 for Option 2, two HD-96 units in parallel are required. The characteristics of each unit and the system performance of both units are:

Unit Overall Dimensions (L x D in feet)	20 x 8
Unit Weight (pounds)	30,000
Maximum Exhaust Temperature (°F)	800
Efficiency (Effectiveness), (%)	64.5
Pressure Drop (inches of water)	1.2

These units are constructed from carbon steel. The casing is constructed from 14 gauge steel and is cylindrical. The shell must be insulated to avoid overstressing due to temperature transients. These units will recover 8.7×10^6 Btu/hr and will preheat combustion air in Zones 1, 2, and 3 by 264°F.

2.7.3 Heat Pipe Exchangers (Option 1)

There are several reputable manufacturers of heat pipe exchangers in the United States and Europe. The Dynatherm Corporation CAL-REX modularized air-to-air industrial exchangers were selected for evaluation because they have been used successfully in contaminated industrial process exhausts for a number of years. This line of heat pipe heat exchangers is manufactured for use at various exhaust temperatures up to 700°F.

The characteristics of module and the system performance of the selected module assembly are:

Module Dimensions (L x W in feet)	8.33 x 3.83
Module Thickness (feet)	0.417
Module Weight (pounds)	325
Number of Fins/Inch (supply & exhaust)	6
Number of Modules in Series	5
Number of Parallel Modules	6
Overall Weight of Modules (pounds)	9,750
Maximum Exhaust Temperature (°F)	500
Efficiency (Effectiveness), (%)	64.5
Pressure Drop (inch of water)	1.0

The test heat pipe heat exchanger fouled rapidly because agglomerated particles in the airstream as large as 0.1 inch could not pass through the 0.075 inch heat exchanger airflow passages. For this reason, the heat exchanger designs selected for Option 1 now employ 0.15 inch exhaust airflow passages.

The modules are constructed from copper-nickel tubes, aluminum fins, and aluminum frames. The tubes and fins on the exhaust side are coated with a silicone copolymer for added corrosion protection and for easy cleaning. This exchanger (Figure 12) will recover 8.7×10^6 Btu/hr and will preheat combustion air in Zones 1, 2, and 3 by 264°F.

2.7.4 Capital Investment Analysis (Option 1)

Manufacturers' prices were obtained for the exchangers specified in Sections 2.7.1, 2.7.2, and 2.7.3. Because of the high level of exhaust contamination, the costs of spray systems for periodic cleaning were added for all the heat exchangers. It was assumed that the purge system of the rotary wheel exchanger was inadequate and additional cleaning systems would be required.

In order to improve maintainability, physical access to the heat transfer surfaces was considered necessary. Unfortunately, the same degree of access could not be provided for in each exchanger type. The degree of access varies from poor for the tube and shell exchanger to excellent for the modularized heat pipe exchanger (Figure 12).

To the costs of the exchanger and the provisions for cleaning and maintenance was added the system installation costs to arrive at the estimated system capital cost. Annual operating and maintenance costs were estimated separately and treated as an expense item. These various costs are summarized in Table 3.

Engineering costs peculiar to the system have not been included. Details of the capital investment cost analysis calculation procedures are given in Appendix B.

Neither the beneficial effect on the investment analysis of future increases in cost of energy nor future escalations in cost of equipment have been included in the analysis.

FIGURE 12

ELEVATION VIEW OF HEAT PIPE AIR-TO-AIR EXCHANGER
 DESIGNED FOR PLACEMENT IN STACK 3 EXHAUST
 (Top View is Shown on Page 51)

FIGURE 12 (CONT.)

EXCHANGER TOP VIEW

TABLE 3

CAPITAL INVESTMENT ANALYSIS OF RECOVERY SYSTEMS FOR OPTION 1

Type Exchanger For Option 1	Annual Fuel Savings \$	Equipment Costs Including Exchanger Blower and Packaged Spray System \$	Installation Cost Including Structural Sheet- metal Elect- rical \$	Total Capital Cost \$	Annual Operating and Mainten- ance Cost \$	Net Annual Savings \$	ROI %	Payback Years
Rotary Wheel	188,000	100,000	120,000	220,000	15,000	97,500	48	2.03
Tube & Shell	188,000	97,000	130,000	227,000	16,000	97,350	46.5	2.10
Heat Pipe	188,000	130,000	105,000	235,000	12,000	99,750	46	2.12

1. Assumes natural gas at \$3.00 per MMBTU and 7200 hours per year operation.
2. Assumes power cost at \$0.03 per KWHr and labor costs at \$10/hr.
3. Net annual savings = 50% (annual fuel savings - operating cost - depreciation) + Depreciation. Depreciation taken over 10 years on straight line basis.
4. ROI calculated by discounted cash flow method of investment appraisal. 10% investment tax credit used.
5. Payback equals Total Capital Cost/Net Annual Savings.

Details of the capital investment cost analysis calculation procedures are given in Appendix B.

2.8 Evaluation of Air-To-Liquid Heat Exchanger Equipment

There are two basic types of indirect contact exchangers manufactured in the United States for transferring thermal energy between air and liquids. A wide variety of configurations of these types are available; however, the conventional liquid coil and tube and shell, both of which have been described in Section 2.5 are useable in Zone 1 and Zone 2 exhaust airstreams. A preliminary capital investment analysis of both exchanger types showed that the return on investment of tube and shell exchanger systems was very much less than that of systems using conventional coils for building heating. In view of this, the tube and shell exchangers was dropped from further consideration. A typical installation of finned tube air-to-water heat exchangers is illustrated in Figure 13.

For the heating of white water, a direct contact spray economizer was considered because of its high efficiency and elimination of the fouling problems that would be encountered using a conventional liquid coil exchanger.

FIGURE 13

FINNED TUBE AIR-TO-WATER EXCHANGER INSTALLATION

2.8.1 Direct Contact Spray Economizer (Option 3)

This option requires the heating of 400 gpm of white water, using Zone 2 exhaust, from an initial water temperature of 95°F to a final temperature of 155°F. The total energy transferred to the white water is 12×10^6 Btu/hr. The AER Corporation's direct spray water heating unit was considered for evaluation. The unit consists of a three stage insulated stainless steel casing and sump, stainless steel moisture elimination sections and all necessary pumps and piping. The characteristics of this system are:

Unit Overall Dimensions (L x W in feet)	20 x 15
Unit Depth (feet)	10
Spray Stages (counter flow)	3
Unit Weight (approximate pounds)	20,000
Design Pressure (psi)	30
Air-side Pressure Drop (inches of water)	3.0
Rate of Water Condensation (gpm)	16

2.8.2 Conventional Liquid Coil Exchanger (Option 4)

This option requires the heating of 250 gpm of a mixture of water and ethylene glycol (using Zone 1 exhaust) from an exchanger inlet temperature of 72°F to an exchanger outlet temperature of 150°F. The system transfers 5.84×10^6 (Avg) Btu/hr to the fluid to be used in a recirculating system to provide building heating during the wintertime. The system will be designed to provide a maximum of 90°F temperature increase for the 100,000 SCFM of building heating. The heat exchanger will consist of a section of heating coils fabricated with copper tubes, copper fins and will have a stainless steel casing. An insulated stainless steel enclosure, with a sump will enclose the coil section. Intermittent washdown of the coils will be provided by a spray section ahead of the coils. The heated fluid will be circulated to two enclosed units containing recovery coils for heating 50,000 SCFM of building make up air each.

The characteristics of this system are:

Heating Unit

Unit Overall Dimensions (L x W in feet)	20 x 8
Unit Depth (feet)	10
Unit Weight (approximate pounds)	20,000
Maximum Exhaust Temperature (°F)	400°F
Operating Pressure (psi)	75
Air-side Pressure Drop (inches of water)	1.5
Rate of water Condensation (gpm)	10

Recovery Units

Unit Overall Dimensions (L x W in feet)	10 x 15
Unit Depth (feet)	15
Unit Weight (approximate pounds)	10,000
Air-side Pressure Drop (inches of water)	1.5

2.8.3 Combination Direct Spray and Water Coil Exchanger
(Option 6)

This option combines Option 3 and Option 5 using the energy in the Zone 2 exhaust airstream. This option requires heating of 400 gpm of white water and providing 100,000 SCFM of heated air for building heating using a liquid coil exchanger. The advantages of combining the direct spray and liquid coil systems are:

a) Reduced maintenance and fouling of the liquid coil exchanger which will be operating in fairly clean saturated air discharging from the direct spray unit.

b) Reduced installation and equipment costs by combining the two units into one single housing.

The characteristics of this system are similar to those outlined in Sections 2.8.1 and 2.8.2.

2.8.4 Capital Investment Analysis (Options 3, 4, and 6)

Option 3 requires the heating of 400 gpm of white water from 95°F to 155°F using the energy in the Zone 2 exhaust airstream. Option 4 requires the heating of 250 gpm of clean liquid from 72°F to 150°F using the energy in the Zone 1 exhaust airstream. The Option 4 system is assumed to operate only during the heating season (5,500 hours per year).

The capital investment analysis for these options is given in Table 4. Engineering costs peculiar to the system have not been included.

Option 2, which utilizes heat from Zone 1, would have about the same investment, ROI, and payback as Option 3. Similarly, Option 5, which utilizes heat from Zone 2 would have about the same investment, ROI and payback as Option 4.

It is obvious, when comparing the data in Tables 3 and 4, that the option with the greatest economic incentive is Option 3 (preheating white water).

TABLE 4

CAPITAL INVESTMENT ANALYSIS OF RECOVERY SYSTEMS FOR OPTIONS 3, 4 and 6

Type Exchanger	Annual Fuel Savings \$	Equipment Costs Including Exchanger Blower and Packaged Spray System \$	Installation Cost Including Structural Sheet-metal Electrical \$	Total Capital Cost \$	Annual Operating and Maintenance Cost \$	Net Annual Savings \$	ROI %	Payback Years
Direct Spray Option 3	259,600	80,000	80,000	160,000	18,000	118,000	83	1.22
Air-to-Liquid Option 4	96,000	100,000	105,000	205,000	20,000	48,750	23	3.78
Combination Direct Spray and Air-to-Liquid System Option 6	355,000	160,000	170,000	330,000	38,000	175,000	58	1.70

1. Assumes natural gas at \$3.00 per MMBTU and 7200 hours per year operation.
2. Assumes power cost at \$0.03 per KWHr and labor costs at \$10/hr.
3. Net annual savings = 50% (annual fuel savings - operating cost - depreciation) + Depreciation. Depreciation taken over 10 years on straight line basis.
4. ROI Calculated by discounted cash flow method of investment appraisal. 10% investment tax credit used.
5. Payback equals Total Capital Cost/Net Annual Savings.

Details of the capital investment cost analysis calculation procedures are given in Appendix B.

3.0 CONCLUSIONS

The analysis of the heat recovery potential and the investigation of the heat recovery equipment has shown that it is technically and economically feasible to install and operate heat recovery equipment to recover heat from multi-deck board dryer exhausts.

It is also concluded that the following options as shown in Figure 14 offer the most desirable operating features at Rockdale:

1. (Option 1) Utilize Zone 3 exhaust for preheating combustion air for the first three zones of the dryer. Use an air-to-air heat recovery system. A heat exchanger utilizing heat pipes would be desirable from a maintenance viewpoint. The modular construction available with the heat pipe exchanger which allows easy accessibility and removal for cleaning is a very desirable feature for an installation operating in a contaminated airstream. The absence of moving parts and particulate crossover contamination are also advantages

over the rotary wheel. This system will recover 8.7×10^6 BTU/hr of energy, and has an ROI of 46 percent. The estimated installed cost is \$235,000.

2. (Option 3). Utilize Zone 2 exhaust for heating white water. Use direct water spray economizer to provide contact between the exhaust gases and the white water. This system will recover 12×10^6 BTU/hour. The ROI is 83 percent based on an installed cost of \$160,000.

Options 1 and 3 are shown schematically in Figure 14.

3. Some facilities could find it desirable to utilize Option 6 rather than Option 3. This option incorporates an additional heat exchanger following the direct water spray economizer. This additional heat exchanger is part of a recirculating system used to heat building make-up air. Fouling in this heat exchanger should be minimal as the direct water spray economizer cleans the exhaust gases. This option is not recommended to the Rockdale Plant as there are alternate means to heat the building make-up air.

Option 3
Heat white water

FIGURE 14
RECOMMENDED OPTIONS FOR HEAT RECOVERY

4.0. RECOMMENDATIONS FOR IMPLEMENTATION OF HEAT RECOVERY AT ROCKDALE

It is recommended that the Department of Energy proceed with Phase II and Phase III funding of the project. These plans would be based on implementing the following energy recovery systems at the Rockdale plant (Options 1 and 3).

1. Install an air-to-air heat recovery system to preheat combustion air for Zones 1, 2, and 3 using Zone 3 exhaust energy.
2. Preheat 400 gpm of white water to the hydropulper with a direct spray economizer utilizing Zone 2 exhaust energy.

Phase II would entail:

- a. Prepare detailed engineering drawings for installation of these systems.
- b. Prepare procurement and installation specifications and obtain bids on basis of detailed drawings.
- c. Provide cost estimates with schedule for installing above systems.

- d. Update the Phase III estimate based on equipment and installation bids.

Phase III would entail:

- 1) Place orders and obtain firm delivery dates for equipment items.
- 2) Initiate in-house construction/site preparation/assembly as appropriate.
- 3) Install equipment and complete assembly of auxilliary components.
- 4) Shakedown equipment.
- 5) Demonstrate equipment and provide technology transfer as requested.

5.0 EXPERIMENTAL TESTS TO SUPPORT EQUIPMENT SECTION

5.1 Results of Tests

Reported herein are the demonstration model heat pipe and rotary regenerator air-to-air testing conducted in Stack 3 (Zone 3) exhaust, the pilot direct spray economizer testing, and the investigative work on the associated exhaust contaminants.

5.1.1 Demonstration Model Testing - Heat Pipe

A two row heat pipe exchanger was constructed by Dynatherm Corporation for testing. The exchanger was 29-1/4 inches long, 15-1/4 inches high, and 5 inches deep. The face areas of the exhaust and supply sides were each 0.87 square feet. The exchanger consisted of nine heat pipes, one row having four pipes and the other having five pipes. The pipes were constructed of one inch diameter tubing finned with 0.05 inch thick, 5/8 inch high carbon steel fins. There were eight fins per inch. The balanced efficiency when operating at a

standard face velocity of 600 feet per minute was 25 percent and the standard dry pressure drop was 0.4 inch of water.

Photographs of Stack 3 and the test installation are shown in Figure 15. The heat exchanger was instrumented with thermocouples at the locations shown in Figure 16. In addition, air velocity measurements were made at the supply inlet and exhaust outlet ducts at the locations shown in Figure 16. Pressure drop across the heat exchanger was measured using a water manometer, and air velocity was measured using a Hastings Air-Meter. The test unit was connected to the No. 3 stack using a nominal 6 inch diameter duct. The two blowers shown in Figure 15 are identical and are Model 1-472 made by North American Manufacturing Company.

Stack from Zone 3 showing steam emitted during normal operation. Pipe leading to heat exchanger is at the lower left of the picture.

Test installation for the heat pipe heat exchanger during operation. Steam is being emitted from the exhaust side of the heat exchanger duct.

FIGURE 15

PHOTOGRAPHS SHOWING THE LOCATION OF AND INSTALLATION DETAILS
OF HEAT PIPE HEAT EXCHANGER TEST

Notes: (1) Thermocouples in these locations are to be at least 12" from exchanger faces. Numbers in circles indicate thermocouple locations.

(2) Velocity measurements must be 18" beyond blower and inside duct.

FIGURE 16

SCHEMATIC OF TEST SETUP SHOWING LOCATIONS OF TEMPERATURE AND VELOCITY MEASUREMENTS

Testing began at 9:20 A.M. on the morning of March 20, 1979. Because the heavy exhaust contamination was not anticipated, the first air velocity measurements were not made until 11:00 A.M. However, reconstructing the data, the initial standard face velocities were estimated to be 847 feet/minute on the supply side and 538 ft/minute on the exhaust side. The exhaust temperature decreased by 108°F in passing through the heat exchanger and the supply air temperature was raised by 76°F. Because the exhaust airstream contains substantial moisture, its specific heat is about 5 percent higher than that of standard air. The calculated efficiencies obtained from temperature measurements are 24.2 percent on the supply side and 34.5 percent on the exhaust side. The difference in efficiency is to be expected because the heat capacity ratio of the two airstreams is 0.67. At this time, the heat transferred between the two airstreams was 60,000 Btu/hr or on the average 1.96 Kw per pipe.

At 11:30 A.M. of the same day, it was noticed that the current being drawn by the exhaust fan motor had increased from 2.87 amps to 3.0 amps. The heat capacity ratio of the two airstreams had decreased from 0.67 to 0.49. Since it can be assumed that the supply stream

air-side remained constant during the interval, a reduction in airflow must have occurred in the exhaust airstream.

Testing continued until 2:30 P.M. at which time temperature, airflow, and pressure drop measurements were made. The supply side pressure drop across the heat exchanger was measured to be 0.5 inch of water and on the exhaust side it was measured to be 9.5 inches of water. The heat capacity ratio was determined to be 0.50 from both temperature and air flow measurements. The supply side efficiency was 18.7 percent and the exhaust side efficiency was 35.3 percent as determined from temperature measurements. The total amount of heat transferred between airstreams had decreased by 800 Btu/hr or 1.3 percent since the start of testing.

At this time, the unit was removed from the test installation and examined. It was observed that the exhaust side face of the heat exchanger was fouled substantially on its entire surface. A view of the heat exchanger is shown in Figure 17. A sample of the fouling material was secured for further analysis. Various attempts to clean the heat exchanger were made using methylene chloride, tetrachloroethane, and other

cleaning fluids. Although the material was not tightly adhering, a gentle application of these fluids was not effective. Finally, a water spray was tried and this removed the material.

The unit was reassembled in the test installation and another run was made. Before any substantial data could be obtained, the unit again fouled.

It was decided to demonstrate the cleaning of the unit when installed and in operation. To this end, a spray nozzle was placed at the center of a length of 1 inch IPS pipe. One end of the pipe was capped and the other end was connected to the plant water supply. A schematic of the test setup is shown in Figure 18. The orientation of the exchanger remained vertical. The preferred orientation of heat exchangers for spray cleaning is horizontal with the spray above the exhaust side face. The recommended direction of spray and flow of exhaust gases through the exchanger are the same as was in the case in this testing.

Photograph of heat pipe heat exchanger module. Note that the exhaust side fins are fouled so fully that they are not distinguishable.

Sample of fouling material as viewed under a scanning electron microscope at 420 x. The view shows at (a) fibrous strand, (b) and (c) perlite fragments, and (d) fiber strand. The fibers are possibly from paper.

FIGURE 17
PHOTOGRAPHS SHOWING FOULED HEAT EXCHANGER AND FOULING MATERIAL

FIGURE 18

SETUP FOR SPRAY CLEANING TEST

Heat exchanger testing began at 10:15 A.M. on April 17, 1979. Exhaust side pressure drop was 1.0 inch of water at the start of testing. Although temperature and velocity measurements were obtained, the presence of water made the readings difficult to interpret. It was decided to permit the exhaust side pressure drop to increase to 2 inches of water before initiating spray cleaning. This occurred at 1:20 P.M. The spray was initiated at 1:25 P.M. (water pressure 70 psig) and remained on for 1 minute. At 1:30 P.M. the pressure drop across the heat exchanger was 1.8 inches of water. The pressure drop remained at this level until 2:10 P.M. when the spray was again turned on for 30 seconds. At 2:30 P.M., the pressure drop was measured to be 1.7 inches of water. The unit continued under test and at 4:00 P.M. the exhaust pressure drop was again at 2.0 inches of water. The spray was turned on for 1 minute and the pressure drop was measured at 1.7 inches of water. There was no indication that the process was not repeatable indefinitely.

An opening was cut into the stack side of the transition to the heat exchanger in order to view the condition of the finned surfaces after spraying. It was observed that, except for the bottom row, all surfaces

were clean. Views of both sides of the heat exchanger after removal from the duct are shown in Figure 19.

Blockage of the bottom heat pipe was caused by the masking effect of water collected at the low point of the duct and exchanger assembly during spraying. This effect was anticipated. The fouling of one pipe out of an average of 4.5 pipes (one row of four pipes and one row of five pipes) results in a decrease of face area of 28 percent. This area decrease results in an increase of face velocity of 28 percent. Since pressure drop varies as the square of the velocity (all other variables being kept constant), the observed blockage will result in an increase of pressure drop from the initial 1 inch of water to 1.65 inches of water. This agrees with the 1.7 inches minimum pressure drop observed after cleaning.

For the test module, a 1 minute spray cycle every three hours appears adequate. A wide angle full cone spray pattern is required, similar to that obtained from the Spraying Systems Company, Type GA-W spray head. A flow rate of 6 gpm is required.

Translating this to the Figure 12 design, the 84 square foot face area will require a stage flow rate of 504 gpm or 2,520 gpm for all five stages. The spraying in the Figure 12 design can be staggered so that piping to the spraying system can be based on a capacity of 420 gpm. Because of the large airstream spacing of the Figure 12 design (twice that of the test exchanger) an estimated spraying cycle of once a day is required.

Photograph of heat exchanger after removing from the cleaning demonstration test setup. View shows the side nearest the stack.

Opposite view of the heat exchanger. The black deposit on the exhaust side fins is thought to be thin layer of condensed asphalt.

FIGURE 19

PHOTOGRAPHS OF HEAT PIPE HEAT EXCHANGER AFTER CLEANING DEMONSTRATION TEST

5.1.2 Demonstration Model Testing Rotary Regeneration

A pilot energy recovery wheel demonstrator was obtained from the Wing Company and tested in the Stack 3 exhaust. The design and dimensions of the wheel supplied with the demonstrator are given in Figure 20. The manufacturer reports that the useable per side face area of the wheel is 0.62 square feet, although one-half the actual wheel area is 0.88 square feet. Some fraction of the face area is reserved for the purge section.

The test setup of the demonstration unit is shown in Figure 21. The unit was supplied with its own blowers. Blower characteristics are given in Figure 22. Instrumentation consisted of temperature measurements before and after in the exhaust and supply airstreams as indicated in Figure 21. Air velocity measurements were made at the exhaust and supply side exits from the unit. In addition, supply side exit air was monitored for contamination.

Testing began on Tuesday, May 8, 1979. Erratic results were obtained. The unit was examined and it was discovered that the blower mounts were loose or detached

from the housing frame. After these defects were corrected, consistent data was obtained. Data obtained on May 9, 1979 and May 10, 1979 are summarized in Table 5. For the wheel design supplied with this demonstrator, the manufacturer predicts an efficiency of about 45 percent at a balanced flow rate of 275 SCFM. The test, however, was conducted under unbalanced conditions. The heat capacity ratio of the two airstreams was about 0.8, and the exhaust side had the lower heat capacity. The predicted wheel exchanger efficiency as determined from the unbalanced measured efficiencies is very nearly 45 percent at a flow rate of 140 SCFM.

The wheel exchanger efficiency is not as sensitive to surface contamination by dirt as are other exchanger types. Dirt actually increases the heat capacity of the wheel itself and this has a beneficial effect on efficiency. In actual testing, the wheel exchanger efficiency changed very little as would be expected from the above observation. On the other hand, surface contamination has an adverse effect on pressure drop and this results in a decrease in flow rate of 4 to 8 percent.

Sine Wave
Element
(Typical)

FIGURE 20.

PILOT DEMONSTRATOR WHEEL DESIGN

FIGURE 21

TEST SETUP OF WING COMPANY CORMED PILOT DEMONSTRATOR

FIGURE 22
BLOWER CHARACTERISTICS

PARAMETER	5/9/79 Test	5/10/79 Test
Stack Number 3 Exhaust Temperature (°F)	444.2	452.2
Exhaust Gas Temperature at Exchanger (°F)	314.1	333.8
Exhaust Gas Temperature out of Exchanger (°F)	219.3	226.8
Inlet Supply Air Temperature (°F)	114.7	114.4
Outlet Supply Air Temperature (°F)	191.1	197.9
Exhaust Air Flow Rate (ACFM)	125.0	114.6
Supply Air Flow Rate (ACFM)	171.9	165.6
Exhaust Air Flow Rate (SCFM)	105.6	95.8
Supply Air Flow Rate (SCFM)	139.9	133.4
Exhaust Side Efficiency (%)	47.5	49.0
Supply Side Efficiency (%)	38.3	37.9
Energy Transferred (Btu/hr)	11,450	11,900

TABLE 5

SUMMARY OF TEST RESULTS OF THE WING COMPANY
CORMED PILOT DEMONSTRATION UNIT OPERATED WITH STACK 3 EXHAUST GASES

The overall energy transferred from the exhaust to supply airstreams is about 12,000 Btu/hr. This is one-fifth of the heat transferred during the previous testing with the heat pipe exchanger. The face areas of the wheel and heat pipe exchanger were approximately equal; however, the face velocity during heat pipe exchanger testing was approximately five times greater. Had the wheel exchanger been tested at the higher face velocity, the efficiency and heat transferred would have been comparable with that of the heat pipe exchanger.

The wheel exchanger was tested immediately after the dryer had been shut down for general maintenance and cleaning. Operating personnel commented that the oven exhausts were generally cleaner for the first few days after such a shutdown. The Stack 3 exhaust during oven operation is shown in the photograph in Figure 23, and may be compared with a similar photograph shown in Figure 16.

To measure the particulate crossover contamination, a porous cloth filter was placed over the exit of the supply airstream. This is shown in the photograph in Figure 23. It is estimated that during 45.5 hours of testing, 374,000 cubic feet of air passed through this

filter. The cloth was analyzed by an independent laboratory. It was determined that between 500 to 1,000 particles over 5 microns per cubic foot were present in the supply airstream.

The exhaust side of the heat exchanger installation is shown in Figure 24. Also shown in Figure 24 is the accumulated dirt in the exhaust plenum downstream of the wheel. The dirt consisted of a relatively soft mass unlike that experienced during the heat pipe exchanger testing.

The difference in the nature of the observed dirt in the heat exchangers is due to one or more of the following conditions:

The wheel exchanger was tested in the Stack 3 exhaust immediately after dryer maintenance and cleaning.

Differences existed in board thickness being processed in the dryer.

Differences existed in the exhaust air velocity in the six inch diameter duct running from Stack 3 to the demonstrators. In the wheel case, the standard duct velocity was 8 ft/second; and, in the heat pipe case, the velocity was 40 ft/second. This means that in the heat pipe test, particles ten times larger (than in the wheel test) were transported from the Stack 3 exhaust through the six inch duct to the heat exchanger.

Stack from Zone 3 showing no
Visible emissions.

Test installation of wheel exchanger
demonstrator. Note cloth over supply
air exhaust to catch particulates.

FIGURE 23

PHOTOGRAPHS SHOWING THE LOCATION AND INSTALLATION
OF WHEEL EXCHANGER DEMONSTRATOR

Exhaust side of demonstrator during operation. Six inch duct from Stack 3 comes into unit from left.

View inside of demonstrator after testing completed. Note dirt at the bottom of exhaust plenum.

FIGURE 24

PHOTOGRAPHS SHOWING THE EXHAUST SIDE OF
DEMONSTRATOR DURING AND AFTER TESTING

5.1.3 Pilot Model Testing - Direct Spray Economizer

A pilot model direct spray economizer was obtained from the AER Corporation, Ramsey, New Jersey, and tested on Zones 3 and 2 exhaust stacks. The unit was designed to handle 1,000 cfm exhaust at an average wet bulb temperature of 146°F and to heat 10 gpm of 80°F water to 135°F.

The picture of the test setup of the unit is shown in Figure 25 and a schematic sketch is shown in Figure 26. The unit consisted of a two stage insulated spray unit, arranged so that all of the water sprayed into a chamber is in intimate contact with the airstream. The water is collected in a sump at the bottom from which it is pumped to the second spray chamber using a counter flow arrangement. The airstream leaves the unit after it passes through a plastic mist eliminator section. The spray assemblies consist of brass nozzles, mounted on copper tubing. Indicating temperature gauges and water flow meters are located in the prepiped manifold for both the incoming and leaving water lines. A direct reading air flow measuring device is mounted on the outlet duct. The unit is equipped with its own adjustable drive fan, pump and motors. The overall size

of the unit is approximately 3 feet wide by 3 feet deep by 5 feet high. Since the size of the spray nozzles on this unit had only 15/64 inch orifices, the tests had to be conducted with clean water instead of white water to avoid plugging of the nozzles.

The unit was started up and tested on July 17 and July 18, 1979. The unit was connected to the third zone exhaust stack. The following observations were made after these tests:

1. The water could be heated by about 42°F to 55°F at a rate of 10-12 gpm. The airflow through the unit ranged from 900 to 400 ACFM as measured by the air flowmeter.

2. The water flow to the first and second chamber required constant adjustment because of spillover from stage 1 to 2. The flow out of the unit was erratic, and it appeared that the flexible hose used for the outlet pipe was restricting the flow meter.

3. There was a large in-draft into the unit through the open overflow pipe. There was also some in-leakage into the unit at the top and fan inlet which

resulted in a higher measured air flow at the fan outlet than what we were actually pulling from the stack.

4. The inside of the unit was clean and showed no signs of particulate buildup proving that the unit does a good job of washing out the particulates from the airstream.

We also found that the unit had apparently been left connected to the dryer stack prior to the start of the tests and the particulate in the exhaust stack had plugged up the plastic mist eliminator pads inside the unit.

The pads were removed and when the unit was turned on again the next day, there was a tremendous carryover of water from the spray chamber into the fan and discharging out into the atmosphere. All the outlet ductwork connections were leaking and the only way to reduce this problem was to cut back on the water flow to about 6 gpm.

At this time, it was decided to move the unit to test on the Zone 2 exhaust. Figure 25 shows the photographs of the test unit connected to the Zone 2

exhaust stack. In order to overcome the difficulties encountered with the initial tests, the following modifications were made to the unit:

1. The top row of sprays in both the spray chambers were turned down so that the direction of sprays was down instead of up to eliminate or reduce the free water carryover in the exhaust and also in the inlet duct connection.

2. Sight glasses were installed in both the Stage 1 and Stage 2 sumps to determine the level of water so that water inlet and flow volume could be better balanced without spillover.

3. The outlet water connection on the unit was connected to a larger pipe to eliminate any restrictions to water flow on the outlet side.

4. The overflow pipe was sealed to prevent in-draft into test unit and all duct connections were tightened to reduce leakage and spillage.

When the unit was started up again on August 17, it was noticed that the airflow meter was not working well,

so it was removed and the airflow was measured by a pitot traverse in the discharge duct. The outlet water flow was measured by timing the flow into a bucket and then calculating the flow rate. The changes outlined above made the test unit operate much better. We substantially reduced the free water discharging out of the stack and were able to balance the water flow.

The data obtained from the tests on August 21, 1979, showed that we could heat 15 gpm of water from 70°F to 130°F. This translates to 7500 BTU/min recovered with about 350 SCFM of exhaust. Table 6 lists the data collected during the tests conducted on Stacks 3 and 2. It is difficult to make accurate energy balances from the test data because of the various leaks of air and water out of and into the unit. The tests do show, however, that a 55°F - 60°F temperature increase can be achieved with a direct spray unit for heating white water and that 12×10^6 BTU/hr can be recovered in a full size installation handling 400 gpm of white water. In a full size unit, where the white water is entering at a higher inlet temperature (95°F vs. 70°F in test), it may be practical to have a third stage in the spray chamber.

All the tests with the pilot unit were conducted using fresh water whereas in an actual installation of such a system, we would use white water containing about 0.3 percent solids. In order to determine the nozzle orifices that would be best suited for white water operation, we ran a test with three nozzle sizes. Three Spraying Systems Whirljet Nozzles with 0.75 inch connection and 0.5 inch body inlets were connected to a pipe header. The outlet orifice sizes on these nozzles were 5/16 inch, 9/16 inch and 23/32 inch. At 20 psi, the flow from these nozzles is rated at 5.7 gpm, 11.3 gpm and 17 gpm respectively.

The nozzle header was connected to a white water line and allowed to run for about six hours. At the end of the run, only the 23/32 inch nozzle was spraying. The smallest size orifice (5/16 inch) tested plugged within a half hour of operation while the next larger size (9/16 inch) lasted about three hours. It appears that orifice sizes larger than 0.75 inches would not cause problems with plugging and should provide relatively troublefree operation.

Zone 2 exhaust stack with pipe leading to economizer.

Test installation for direct spray economizer showing the spray chamber, fan, pump and piping. Heated water pipe is in foreground.

FIGURE 25

PHOTOGRAPHS SHOWING THE LOCATION AND INSTALLATION DETAILS
OF DIRECT SPRAY ECONOMIZER

FIGURE 26
TEST SETUP OF AER CORPORATION DIRECT SPRAY ECONOMIZER

No. Test Parameters	Stack No. 3		Stack No. 2	
Test Date	7-17-79	7-18-79	8-17-79	8-21-79
1. Inlet Supply Air Temperature (°F)	315 170	266 DB 177 WB	330 DB 166 WB	340 DB 170 WB
2. Outlet Air Temperature °F	120 Sat	125 DB 124 WB	106 Sat	120 Sat
3. Air Flow Measured at Outlet ACFM	900	533	170	400
4. Inlet Water Temperature (Stage 1) °F	78	73	72	70
5. Stage 2 Water Inlet Temperature °F	114	117	96	100
6. Outlet Water Temperature °F	121	132	116	130
7. Heated Water Flow Leaving Unit GPM	10	10	10.4	15
8. Energy Transferred to Water BTU/Hr.	258,000	294,900	231,000	450,000

TABLE 6

SUMMARY OF TEST RESULTS OF THE AER CORP. DIRECT SPRAY ECONOMIZER

5.1.4 Investigation of Exhaust Contaminants

Samples of test exchanger fouling material, FESCO Board, TOP-LOC Board Coat asphalt, and Batch Tank Mix asphalt were obtained for evaluation.

The fouling material was examined chemically, by X-ray analysis, and by optical and scanning electron microscopes. The analysis showed that the material consisted of 47 percent perlite, 37 percent fibers, 10 percent gypsum, and 6 percent asphalt. The X-ray analysis does not measure carbon. Otherwise, the X-ray analysis indicated a predominance of silica with smaller amounts of calcium, potassium, and aluminum.

The scanning electron microscope (Figure 17) showed perlite, perlite fragments, and fibers. The fiber-like strands observed throughout the sample did not reveal much by isolated spot X-ray. These were judged to consist mainly of carbon. The perlite fragments were sluffed off the original expanded perlite spheres. One of the characteristics of perlite is the abundance of spherical or convoluted cracks that cause it to break into small pearl like masses.

Finally, the fouling sample was examined using both optical and toolmaker's microscopes. Particle agglomeration and the bridging effect of fibers were observed. The perlite itself ranged in size from 0.013 inch to 0.067 inch with a median size of 0.045 inch. The agglomerated particles were as large as 0.1 inch. These sizes agreed fairly well with sizes measured by the scaling of microphotographs of the fouling material samples.

To determine the corrosiveness of the exhaust gases to commonly used construction materials, the sample of FESCO Board was reconstituted into mat form by breaking it up and adding 300 percent water. The pH of this mixture was measured at room temperature and found to be between 5.0 and 5.1. This is not expected to create a serious corrosion problem for either aluminum or steel (Reference 4) construction materials. The use of direct-fired natural gas will not introduce the normally encountered combustion acids typical of fuel oil firing (which would normally be the other source of corrosive exhaust contaminants)..

The mechanism of asphalt transfer from the mat into the exhaust airsteam and then onto the heat transfer

surfaces was investigated. Asphalt consists of asphaltenes, resins, and oils. The resins form a coating over the particulate asphaltenes. The paraffinic, naphthenic, and naphtha-aromatic oils are the suspending oils. Additionally, asphalt can be emulsified or otherwise made less viscous by further addition of solvents. The molecular weight of the asphaltenes is approximately 30,000. At this molecular weight, the asphaltenes have a very low vapor pressure even at the highest temperature encountered in this board manufacturing process.

The vapor pressure as a function of temperature of the Batch Tank Mix was measured in the absence of air. In the temperature range between 70°F and 300°F, the vapor pressure closely corresponds to that of the hydrocarbon C_8H_{18} . A thin film sample exposed to air at room temperature dries to a tacky surface in a day. This is understandable since the measured vapor pressure at room temperature is about 10 mm Hg.

The air-dried sample was placed into an oven which was operating at 400°F. After 5 hours, the sample was still emitting a smoky vapor which normally suggests that cracking of heavier molecules is occurring.

Examination of the sample after removal from the oven and cooling to room temperature indicated a typical asphalt coating.

Long chain, high molecular weight hydrocarbons pyrolyze at elevated temperatures. Typically, in the 500°C to 600°C temperatures regime the largest hydrocarbon fragments have a molecular weight of about 1000. Theoretically, the vapor pressure of $C_{72}H_{146}$ (molecular weight of 1011.95) is 1 mm Hg at 400°C. The vapor pressures of the higher molecular weight asphaltenes themselves, are, therefore, very small at the maximum operating temperature of the process and their transfer through the vapor phase to the heat transfer surfaces is not expected. In all probability, the observed asphalt in the fouling sample was transferred mechanically by erosion of asphalt coated particulates and fibers from the mat.

The other major source of exhaust airstream contamination results from the pyrolysis of cellulose and cellulose derivatives. The products of pyrolysis of cellulose consist of water, CO_2 , and tars. Mass spectrometer analysis of the tars from cotton cellulose suggest that they consist of Levoglucosan ($C_6H_{10}O_5$) and

compounds of double bonded carbon and oxygen. The percent volatilization is determined by temperature and time at temperature. The rate of volatilization is determined by temperature and amount of material volatilized. For example, at 291°C and 10 percent loss of weight, the extrapolated rate of volatilization is 1.1 percent per minute, whereas at 261°C, the volatilization rate is 0.13 percent per minute. Since the rate of degradation in the temperature range of 200°C to 500°C tends to double for every 10°C rise in temperature, the extrapolated weight loss for a 10 minute cycle at 200°C is 0.02 percent. Clearly, this rate is too small to be observed in fouling samples obtained from short-term testing. Experience with recovery exchangers installed in cotton drying processes indicates that, after a period of months, cotton waxes and tars form on heat exchanger surfaces. These have been removed by washing with organic solvents. Materials collected on Stack 3 hat supports appear to be cellulose waxes and tars.

6.0 FIELD VISITS TO RECOVERY EQUIPMENT USERS

Field visits were made to solicit information from plant operating personnel with regard to the suitability, maintainability, reliability, and effectiveness of various types of heat recovery equipment installed in industrial processes. A summary of the installations visited and major observations thereof are given in Table 7.

The four exchanger types investigated represent technology that has been in use for more than 50 years. Only in the last decade, however, have attempts been made to use these devices under fouling conditions such as encountered at Rockdale. Progress has been slow even for well-established exchanger manufacturers. For example, a survey of installations completed by a rotary wheel manufacturer during the first ten years shows over 500 HVAC (clean air) installations, and seven in the textile industry, seven in the paper industry, ten in paint drying ovens, and ten in the chemical industry.

It will be noted that of the six installations visited (Table 7), two are in the textile industry, one

in the cotton processing field, one commercial power plant, one in the paper industry, and one in the metal fabrication industry. Because of the high degree of reliability required by commercial power plants and because preheating combustion air has been a practice for more than 50 years, a great quantity of technology has been accumulated on the nature and control of corrosion and fouling. For this reason, the power plant visit was considered to be of special importance.

6.1 Rotary Wheel and Tube and Shell Exchanger Installations

A visit was made to one Baltimore Gas Electric power plant complex which consists of several medium size coal and oil fired boilers and to United States Container Corporation, North Brunswick, New Jersey. The power plant employs both rotary wheel and tube and shell exchangers to preheat combustion air, and the United States Container plant uses a rotary wheel in a 1100°F incinerator application. The following observations summarize these visits:

TYPE FACILITY VISITED	Type Exchanger Installed	Function Performed	Exhaust Temperature	Type of Exhaust Con- tamination	Type of In Place Cleaning	Cycle of Maintenance	Observations
			°F			Months	
Utility	Rotary Wheel	Preheating Combustion Air	600	Coal & Oil Ash SO ₃ , H ₂ SO ₄	Steam Blowdown Purge	12	Heat Transfer Surfaces & Seals Replaced As Required
Utility	Tube & Shell	Preheating Combustion Air	600	Coal & Oil Ash SO ₃ , H ₂ SO ₄	Steel Balls	6	Tubes Worn Through Are Replaced As Required
Steel Products	Rotary Wheel	Preheating Process Air	1100	Various Ashes	Purge	6	Heat Transfer Surfaces & Seals Replaced As Required
Filter Products	Heat Pipe	Preheating Process Air	300	Cotton Wax & Fibers	None	3	Modules Removed, Degreased, and Air Cleaned
Textile	Heat Pipe	Plant & Process Heating	350	Oils, PCE, & Fibers	Water & Detergent Spray	12	Modules Removed and Cleaned If Required
Textile	Heat Pipe	Plant & Process Heating	350	Oils, PCE, & Fibers	Water & Detergent Spray	12	Modules Removed and Cleaned If Required
Paper	Air-To- Liquid	Plant & Process Heating	360	Tissue Dust	Water Spray	12	No signs of fouling after 6 mos. of operation. Will inspect at end of year.

TABLE 7
SUMMARY OF FIELD SURVEY

•A vertically mounted tube and shell exchanger was installed by Baltimore Gas & Electric to preheat combustion air on the premise that passive equipment is more reliable and requires less maintenance. However, to clean the inside of the tubes (flue gas side) a pneumatic steel ball system was installed. Unfortunately, accompanying the disadvantage of the additional cost of this system, the steel balls wore holes in the tubes which negated the promised reliability. Because maintenance on tube and shell exchangers is inherently more difficult than on rotary wheels, maintenance costs are high.

•The rotary wheel exchangers (manufactured by The Air Preheater Company, Inc., and Wing Industries, Inc.) were observed in operation. Glass covered viewing ports and internal lights located in the supply side ducts permitted visual inspection of cold end heat transfer surfaces during operation by maintenance personnel. Equipment in service up to 30 years was observed. Causes for maintenance are associated with bearings, seals, cold end corrosion, and fouling.

Pressure drop increase of no more than 0.5 inch of water and bypass leakages of no more than 10 percent are permitted before shutdown for maintenance is required. To increase the time between shutdowns, soot blowers, flowing cooling oil for bearing lubrication, and air temperature controls are employed. Replacement of corroded plates and worn seals are common maintenance items. Water washing has been employed to clean fouled heat transfer surfaces during the annual plant shutdown.

•The rotary wheel exchanger at U. S. Container is used to preheat oven exhaust gases to approximately 450°F prior to incineration at 1100°F. The wheel made by the Wing Corporation could not be observed while it was operating but there were indications of leakage of fumes around the unit. Maintenance problems are basically associated with seals and bearings.

6.2 Heat Pipe Exchangers

Visits were made to a manufacturer of cotton filters and to two textile manufacturers where heat pipe exchangers have been installed to extract heat from contaminated process air for purposes of preheating either process air or supplying heated air for building heating. The following observations summarize these visits:

- The final process step in the manufacture of cotton filters consists of a bleaching operation employing NaOH, peroxide, and acetic acid neutralization. Approximately 1800 pounds of water must be removed in each batch to obtain an 800 pound dry weight product. High velocity air at temperatures up to 300°F is employed in the drying operation. At the beginning of the cycle, large quantities of hot water are exhausted from the system in the airstream. A modularized six row heat exchanger was placed in the exhaust and adjacent supply air ducts. The heat exchanger was constructed of aluminum. Fins at 14 fins per inch spacing were spiral wound on the heat pipes. The performance of the exchanger was measured by recording temperatures at various locations and throughout the drying cycle. The average efficiency was 45 percent. At the beginning of each cycle, the ejected hot water washes the exhaust fin surfaces clean of cotton waxes deposited thereon at the high temperature end of the previous cycle. The most troublesome aspect of the operation is the accumulation of cotton fibers which have become airborne and are injected into the supply intake of the system. The system is designed so that the heat exchanger modules can be slipped out, cleaned, and reinstalled in 4 hours. Cleaning is required after every 1.5×10^6 pounds of finished product has been processed, and consists of degreasing the modules in a commercial

degreaser and blowing off the remaining loose cotton fibers with compressed air.

•Modularized heat pipe exchangers have been used in the textile industry as part of exhaust air cleaning systems. In these systems, the exchangers serve to reduce exhaust airstream temperatures to permit coalescing condensed organics on fiber glass filters, and to preheat clean process air. Water sprays with the capability of occasionally injecting detergents into the spray water are employed for exhaust side cleaning of the exchangers. The modules are constructed of aluminum and are of such weight and size that they can be easily removed, cleaned, and reassembled into the system. Where particularly corrosive exhaust airstreams are encountered a double coating of heat curing silicone copolymer is applied to the exhaust side surfaces. The exchangers have performed well for a number of years in these engineered packaged systems. Removal, cleaning, and replacement of a 5 module exchanger was estimated by plant maintenance personnel to take less than an hour. Disposable air filters are utilized in front of the heat exchangers to collect lint.

6.3 Air-To-Liquid Heat Recovery Installation

A visit was made to the Mosinee Paper Company in Wausau, Wisconsin to observe an air-to-liquid heat exchanger installation. This installation is used to recover heat from the exhaust of a Yankee paper dryer. The recovered heat is used for process air heating and plant make up heating during the winter. The heat exchangers and all related equipment were supplied by the AER Corporation. The heat exchangers utilize finned coils through which a diluted ethylene glycol solution

(50 percent) is circulated. At the time of the visit, the installation had been in operation for about six months and observation of the heat exchanger showed no signs of deterioration or fouling of the tubes. The customer indicated no maintenance problems associated with this system and is quite pleased with its operation.

1. Data Used for Heat BalanceA. Board Properties

Board Width (FT-IN)	12-4
Mat Weight/sq. ft. (LBS/FT ²)	3.79
Moisture Content into Dryer*	331
Moisture Content--Zone 1 Exit*	246
Moisture Content--Zone 2 Exit*	158.4
Moisture Content--Zone 3 Exit*	72.7
Moisture Content--Zone 4 Exit*	22.2
Moisture Content out of Dryer*	0.2
Specific Heat of Board (BTU/LBM-°F)	0.3
Board Temperature into Dryer (°F)	100
Board Temperature out of Dryer (°F)	145

B. Dryer ConditionsWet End Seal

Exhaust Volume (CFM)	10,000
Dry Bulb Temperature (°F)	222
Wet Bulb Temperature (°F)	152

Zone 1

Exhaust Volume (CFM)	23,920
Dry Bulb Temperature (°F)	354
Wet Bulb Temperature (°F)	168
Fuel Flow (CFH)	38,710
Operating Temperature (°F)	738
Combustion Air Volume (SCFM)	12,710

Zone 2

Exhaust Volume (CFM)	21,490
Dry Bulb Temperature (°F)	352
Wet Bulb Temperature (°F)	180
Fuel Flow (CFH)	34,740
Operating Temperature (°F)	736
Combustion Air Volume (SCFM)	8,770

Zone 3

Exhaust Volume (CFM)	58,010
Dry Bulb Temperature (°F)	412
Wet Bulb Temperature (°F)	168
Fuel Flow (CFH)	21,640
Operating Temperature (°F)	718
Combustion Air Volume (SCFM)	9,130

*% Moisture - Dry Basis

Zone 4

Exhaust Volume (CFM)	32,440
Dry Bulb Temperature (°F)	240
Wet Bulb Temperature (°F)	126
Fuel Flow (CFH)	19,820
Operating Temperature (°F)	581
Combustion Air Volume (SCFM)	1,430

Zone 5

Exhaust Volume (CFM)	35,940
Dry Bulb Temperature (°F)	272
Wet Bulb Temperature (°F)	130
Fuel Flow (CFH)	14,430
Operating Temperature (°F)	486
Combustion Air Volume (SCFM)	500

C. Water Removed per Zone in Pounds per Hour

Zone

1	20,460
2	21,090
3	20,630
4	12,160
5	5,300
Total	79,630

2. Heat Loads for Dryers

A. Transmission Loads

Example--Zone 1

Heat Load=Area x Heat Loss

Factor Based Upon Difference
Between Surface Temperature
and Ambient

$$\begin{aligned} &= (82 \times 18.5) \times (114 - 70) + (82 \times 14.5) \times \\ &\quad (130 - 70) + (82 \times 18.5) \times (130 - 70) \\ &= 0.229 \text{ MMBTUH} \end{aligned}$$

Summary of Transmission Loads in MMBTUH

Zone

1	0.229
2	0.334
3	0.324
4	0.337
5	0.357
Total	1.581

B. Heating Board

Example--Zone 1
Heat Load lbs/hr (solids) x
Specific Heat x ΔT
= $(24,071.53) \times (0.3) \times (115-100)$
=0.108 MMBTUH

Summary of Loads in MMBTUH

Zone

1	0.108
2	0.072
3	0.058
4	0.051
5	0.036
Total	<u>0.325</u>

C. Heating Water

Example--Zone 1
Heat Load lbs/hr (Water) x
Specific Heat x DT
= $(20,460) \times (1) \times (200-100)$
=2.046 MMBTUH

Summary of Loads in MMBTUH

Zone

1	2.046
2	2.109
3	2.063
4	1.216
5	0.530
Total	<u>7.963</u>

D. Evaporating Water

Example--Zone 1
Heat Load lbs/hr (Water) x
Latent Heat
= $(20,460) \times (980)$
=20.052 MMBTUH

Summary of Loads in MMBTUH

Zone

1	20.052
2	20.665
3	20.216
4	11.913
5	5.190
Total	<u>78.036</u>

E. Superheating Water
(inc. water of combustion)

Example--Zone 1
Heat Load=lbs/hr (Water) x
Specific Heat x DT
=(20,461+3,635)x(0.44)x354-200)
=1.633 MMBTUH

Summary of Loads in MMBTUH

Zone

1	1.633
2	1.628
3	2.114
4	0.247
5	0.211
Total	5.833

F. Exhaust (Dry Air)

Example--Zone 1
Heat Load=lbs/min (Dry Air) x
60 x Specific Heat x DT
=(23,920/31.3364)x(60)x(0.24)
x(354-80)
=3.012 MMBTUH

Summary of Loads in MMBTUH

Zone

1 (inc. seal)	3.910
2	2.144
3	8.433
4	3.824
5	4.826
Total	23.137

G. Summary of Calculated Heat Loads
in MMBTUH

Zone

1	27.978
2	26.952
3	33.208
4	17.588
5	11.150
Total	116.875

H. Heat Input to Dryers Based on
Lower Heating Valves

Rockdale Gas Lower Heating
Valve=930 BTU/FT³

Example--Zone 1
Heat Input=Fuel Volume x LHV
=(38,710)x(930)
=36.000 MMBTUH

Summary of Heat Input
Quantities in MMBTUH

<u>Zone</u>	
1	36.000
2	32.308
3	20.125
4	18.433
5	13.412
Total	<u>120.278</u>

I. Heat Input to Dryers Based on
Higher Heating Value

Rockdale Gas Higher Heating Value
=1031 BTU/FT³

Example--Zone 1
Heat Input=Fuel Volume x HHV
=(38,710)x(1031)
=39.910 MMBTUH

Summary of Heat Input
Quantities in MMBTUH

<u>Zone</u>	
1	39.910
2	35.817
3	22.311
4	20.434
5	14.875
Total	<u>133.347</u>

J. Unaccountable Losses

Example--Zone 1
Loss=Heat Input (LHV)-Calculated
Input
=(36.000-27.978)
=(+)8.022 MMBTUH

Summary of Unaccountable Losses in MMBTUH

Zone

1	(+)	8.022
2	(+)	5.356
3	(-)	13.083
4	(+)	0.845
5	(+)	<u>2.262</u>
Total	(+)	3.403

NOTE: (-) indicates that the actual heat input is less than the calculated input, i.e., there is an interaction between zones.

K. Energy Required to Remove Water from Fesco Board

$$\begin{aligned}\text{Energy Required} &= \text{Total Energy Input} \div \text{Total Water Removed} \\ &= 133,347,000 / 79,630 \\ &= 1675 \text{ BTU/LBH}_2\text{O}\end{aligned}$$

APPENDIX B

The calculation of ROI for the various options has been calculated as shown below for Option 6. This is a standard discounted cash flow method of investment appraisal.

Option 6 - Combination direct spray and air-to-liquid system

1. Total Capital Cost	\$330,000
2. Less Capital Cost Investment Tax Credit (10%)	33,000
3. Net Capital Cost	297,000
4. Estimated Annual Fuel Savings	355,000
5. Estimated Annual Operating and Maintenance Cost	38,000
6. Annual Savings less operating cost	317,000
7. Depreciation (10 years straight line basis)	33,000
8. Annual Savings less Depreciation	284,000
9. Taxes @ 50% of Annual Savings	142,000
10. Annual Savings after Taxes and Depreciation	142,000
11. Add back Depreciation	33,000
12. Cash Net Income (Savings)	175,000
13. Present Worth Factor (PWF) = Net Capital Cost/Cash Net Income	1.70
14. ROI (Rate of Yield) from Present Worth tables (10 year life) (%)	58
15. Payback based on Present Worth Calculations = Net Capital Cost/Cash Net Income (Yrs.)	1.7

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