PLANT WIDE ASSESSMENT

for

JOHNS MANVILLE WATERVILLE OHIO PLANT 1

6050 River Road
Waterville, Ohio 43566

Co-funded By:

Johns Manville

and

U.S. DEPARTMENT OF ENERGY PLANT-WIDE ENERGY EFFICIENCY OPPORTUNITY ASSESSMENTS

Principle Participants at Johns Manville:

Neil Farrow, Electrical Engineer
Richard LeBoeuf, Senior Director of Manufacturing Technology
James McCormick, Plant Manager
Tom Allen, Plant Engineer

Report Submitted By:

Kelly Kissock, Energy Information Systems
Arvind Thekdi, E3M

July 2, 2006

Disclaimer

The purpose of this assessment is to identify and approximately quantify savings opportunities. The report is not intended to provide detailed engineering plans or designs for implementing the recommendations. Estimates of savings and costs are based on the best information available to the authors within the scope of the assessment. However, the authors make no warranty with respect to the accuracy of the savings estimates or contents of the report. The client is encouraged to evaluate each opportunity and attain further engineering analysis, if desired, to verify or refine savings estimates.

TABLE OF CONTENTS

I. Executive Summary	4
II. Plant Overview	5
Johns Manville Waterville Plant 1	5
Process Description	5
Plant Layout	9
Melter Efficiency Analysis	10
Process Heating Analysis	12
Compressed Air System Analysis	22
III. Whole Plant Utility Analysis	29
Electricity, Natural Gas and Water Use	29
Avoided Costs	29
Estimated Electricity Use Breakdown	30
Estimated Natural Gas Use Breakdown	31
2003 Measured Natural Gas Breakdown	32
IV. Assessment Recommendations	34
M-1: Use Melting Furnace Flue Gases For Absorption Cooling	35
M-2: Increase Electric Boost in Melting	41
FH-1: Convert 9211 Forehearth to Oxy-fuel Burners	43
FH-2: Reposition Mullite Zone Dividers in Forehearths	46
FH-3: Install Electric Resistance Heating in 9211 Forehearth During Rebuild	48
FH-4: Insulate Forehearths	52
MO-1: Use Mat Oven Incinerator Exhaust Air	55
MO-2: Use Outdoor Air As Combustion And Make-Up Air In Mat Oven	59
MO-3: Reduce Air Entering Mat Oven	64
PC-1: Install Variable Speed Drives on Transfer Pumps	69
PC-2: Install Variable Speed Drives on Cold Well Building Pumps	73
PC-3: Install Variable Speed Drive on Chilled Water Supply Pumps	75
PC-4: Install 100-ton Dedicated Chiller for Binder Room	78
CA-1: Purchase Compressor Sequencer	82
CA-2: Increase Compressed Air Storage and Decrease Pressure	85
CA-3: Institute Program to Fix Compressed Air Leaks	89
L-1: Replace HID Lights with High-Bay Fluorescent Lights	91
V. Implementation and Replication Plan.....	95
Replication within Corporation	96
Replication throughout The Glass Industry	96
VI. Project Participants.....	98

I. Executive Summary

In May 2004, Johns Manville submitted a proposal to the U.S. Department of Energy for a plant wide assessment of the Waterville Ohio Glass Plant. The project was selected for an award and a project kick-off meeting took place at the plant on March 22, 2005.

During the next several months, assessment team personnel visited the plant. With the help of Johns Manville personnel, the team collected energy use, construction, process, equipment and operational information about the plant.

Based on this information, the team identified and quantified 18 energy savings opportunities with a total potential savings of about \$3.5 million per year and a combined simple payback of about 15 months. Implementation of these recommendations would reduce CO₂ emissions by about 53,000,000 pounds per year. These savings opportunities are summarized in the table below. The body of the report contains more detailed descriptions of the production processes, how energy is used in the plant, and the energy conservation opportunities identified.

Table ES1. Summary of Assessment Recommendations (AR)

Assesment Recommendation	Projected Resource Savings			Projected Cost Savings		
	Fuel 10 ⁶ Btu/yr	Electricity kWh/yr	Emissions* lb CO ₂ /yr	Annual Savings \$/yr	Capital Cost \$	Simple Payback
Melters						
M-1: Use Melting Furnace Flue Gases For Absorption Cooling		3,016,000	6,937,000	123,492	574,819	56 months
M-2: Increase Electric Boost in Melting Furnaces	17,844	-3,013,026	-4,914,000	7,400	None	Immediate
Forehearths						
FH-1: Convert 9211 Forehearth to Oxy-fuel Burner	114,730	-2,266,397	7,751,000	714,721	2,000,000	34 months
FH-2: Reposition Mullite Zone Dividers in Forehearths	15,900		1,797,000	111,000	Negligible	Immediate
FH-3: Convert Forehearths to Electric Resistance Heating	197,100	-4,108,440	12,823,000	1,219,471	1,000,000	10 months
FH-4: Insulate Forehearths	135,000		15,255,000	945,000	112,000	1 month
Mat Oven						
MO-1: Use Mat Oven Incinerator Exhaust Air For Absorption Cooling	16,716		1,889,000	117,000	182,000	18 months
MO-2: Use Outdoor Air As Combustion And Make-Up Air In Mat Oven	6,662		753,000	46,600	32,300	8 months
MO-3: Reduce Air Entering Mat Oven	6,158		696,000	43,100	20,000	6 months
Pumping and Cooling						
PC1: Install Variable Speed Drives on Transfer Pumps		281,196	342,500	10,967	25,950	28 months
PC2: Install Variable Speed Drives on Cold Well Building Pumps		306,600	705,200	11,957	14,175	14 months
PC3: Install Variable Speed Drives on Chilled Water Supply Pumps		841,157	1,935,000	32,805	18,600	7 months
PC4: Install 100-ton Dedicated Chiller for Binder Room		910,212	2,093,000	35,498	99,900	34 months
PC5: Modify Chiller Control						
Compressed Air						
CA1: Purchase Compressor Sequencer		499,320	1,148,000	19,473	20,000	12 months
CA2: Install Adequate Compressed Air Storage		920,165	2,116,000	35,886	113,500	38 months
CA3: Institute Program to Fix Compressed Air Leaks Every Month		85,848	197,500	3,348	500	2 months
Lighting						
L1: Replace HID Lights with High-Bay Fluorescent Lights		920,165	2,116,000	35,886	113,500	38 months
Total**	510,110	-1,607,200	53,640,200	3,513,604	4,327,244	15

* Assumes 2.3 lb CO₂/kWh and 113 lb CO₂/mmBtu natural gas

II. Plant Overview

Johns Manville Waterville Plant 1

Johns Manville, a Berkshire Hathaway company, is a leading manufacturer and marketer of building insulations, commercial roofing, and specialty products for commercial, industrial, and residential applications. In business since 1858, the Denver-based company has sales in excess of \$2 billion and holds leadership positions in all of the key markets that it serves. Johns Manville operates four manufacturing divisions including the Building Insulations Group, the Performance Materials Group, the Roofing Systems Group, and the Engineered Products Group. The company employs about 8,500 people and operates 43 manufacturing facilities in North America and Europe.

There are two different plants in Waterville, Ohio, which are designated Plant 1 and Plant 7. This assessment is focused only on Plant 1. Plant 1 is part of the Engineered Products Group (EPG). The EPG operates facilities manufacturing filtration media, glass textile wall coverings, non-woven mats, and fiberglass reinforcements. Products are both glass and synthetic based and are manufactured in both North America and Europe.

Waterville Plant 1 employs about 360 people. It operates 24 hours per day, 365 days per year. The plant includes about 65,000 ft² of office area and 520,000 ft² of manufacturing space. The plant produces glass fibers that are used for both roofing and specialty mat applications, as well as reinforcements applications. The plant also manufactures finished specialty mat products. Two oxy-fuel fired glass furnaces produce fibers. The mat product is produced on a single high-speed fiberglass mat line. The facility includes an on-site oxygen generation facility, raw material batch plant and water treatment facility.

Process Description

The Waterville Plant 1 produces fiberglass products. Figure 1 shows an overview of the processes.

Figure 1. Overview of Fiberglass Production at JM Waterville Plant-1

Batch House

The principle raw materials, silica and borax, are brought into the batch house on rail cars. The materials are transported pneumatically to day bins, where each batch is mixed. Cullet from the plant is added to each batch. Each batch produces about 4 hours of product. Augers transport the material from the day bins to the melt furnaces. The cost of a batch of raw materials is typically about \$60 per ton (“Advanced Heating Techniques for Glass Melting”, Plum, Graf, Beckers, Boersma, Dungen, Frijns and Vervakel, Eindhoven University of Technology, Netherlands, March 5, 2002.)

Melters

The plant has two melt furnaces: Furnace 11 and Furnace 12. Each melter is about 30 ft long, 15 feet wide. The molten glass fills the furnace to a depth of about 2 feet. The arched roof of each furnace is about 4 feet above the glass surface at the center line. Below the glass line, the refractory, which is mainly alumina silica (AZS), is about 12-inches thick. Above the glass line, the arched roof is about 22-inches thick. The rate at which the refractory degrades increases as the temperature of the refractory increases. Furnace 11 is near the end of its 7 to 9 year life. Furnace 12 has been recently rebuilt. Each rebuild costs about \$40,000,000.

Each melter has five zones, numbered sequentially in the direction of glass flow. Each zone has oxy-fired gas burners and submerged electric resistance booster heaters. Glass temperature increases from about 2,200 F in Zone 1 to about 2,500 F in Zone 5. Air temperature in the oven is about 2,700 F. The temperature of the inside surface of the refractory is about 1,500 F. Neither melter uses a recuperator or regenerator since the oxy-fired burners need no combustion air.

Forehearths

From each melt furnace, the molten glass flows through a forehearth that is heated with atmospheric gas-fired burners. Each forehearth has eight legs. Two legs from each forehearth supply molten glass to each forming room; thus there are four forming rooms per furnace.

Measured gas consumption by the forehearth is about 550 mcf/day and 240 mcf/day, which corresponds to annual consumption of about:

$$550 \text{ mcf/day} \times 365 \text{ dy/yr} = 200,750 \text{ mcf/yr}$$

$$240 \text{ mcf/day} \times 365 \text{ dy/yr} = 87,600 \text{ mcf/yr}$$

These numbers are used to estimate annual gas consumption in the Estimated Natural Gas Breakdown.

Forming Rooms

Each furnace supplies molten glass to four forming rooms. In each forming room, two forehearth legs feed a series of electrically-heated bushings that produce glass fibers. Some fiber is wound onto spools, cured in dielectric ovens, and shipped as a final

product. Some fiber is chopped into 1.5 inch pieces, loaded into boxes, and sold as final product. Some fiber is chopped into 1.5 inch pieces, loaded into boxes, and used to produce a mat in the mat machine. Scrap fiber is chopped and transported back to the batch house for cullet. Forming Room 11 is air conditioned by two chillers and Forming Room 12 is air conditioned by three chillers.

Mat Machine

Chopped fiber is combined with “white water” in the pulper. A 600-hp slurry pump moves the slurry from the pulper to the hydro-former, where it is applied on a conveyor as a mat. Vacuum systems pull water downward through the conveyor from the mat. The water is treated and returned to the pulper as “white water”. The total pumping power for the vacuum and white water treatment processes is about 1,500 hp. Water is used for pump seals and disposed to sewer. Binder is applied in the next section of the mat machine. The mat then enters the curing oven, which has 4 gas-firing zones rated at 6 mmBtu/hr each. Water is evaporated in the first part of the curing oven and the binder is cured in the second part. After leaving the curing oven, the mat is slit and wound. Rolls are transported to the warehouse. Hot exhaust air from the curing oven is treated in two three-stage regenerative thermal oxidizers before being discharged to atmosphere.

Space Heating

Space heating is provided by five boilers, gas-fired radiant tube heaters and a make-up air unit near the end of the mat machine.

Melter Air Abatement

Air leaves the melters at about 2,700 F. Atmospheric air is entrained after each melter to reduce the temperature to about 800 F. The air is then drawn down through a tunnel where water is sprayed in through the top. The moist air mixture leaves the tunnels at about 350 F. The air streams are combined before being pulled through a bag house by 350 hp induced air fan.

Binder Room

Binder and chemicals are mixed in vats and pumped to the rest of the plant. Water runs continuously down the drain to dilute chemicals from broken pumps.

Chillers

The plant uses five York chillers. The chillers are rated at 460 V and 707 amps each (190 kW) each. Condenser water leaves the chillers at about 72 F and is returned from the cooling towers at about 82 F. Chilled water is supplied to the plant at about 90 F and returned at about 98 F. The chilled water is used to cool the bushings in the forming rooms.

Air Compressors

Most of the compressed air for the plant is supplied by a 700-hp water-cooled reciprocating compressor with a synchronous motor. The discharge air pressure is 110 psig. In addition, some “process” air is supplied by a 75-hp water-cooled, oil-free compressor rated at 336 cfm. The discharge air pressure is 99 psig.

B 175290-2

Melter Efficiency Analysis

Theoretical Melting Energy

Amorphous solids, such as glass, do not have a sharp melting point. Instead, they pass from solid to liquid over a wide range of temperatures. The complete reaction includes:

- an increase in the specific heat as the polymer goes through the glass transition temperature, T_g
- an exothermic reaction when the polymer crystallizes at the crystallization temperature, T_c , and
- additional heat to break the crystals during melting
- evaporation of gas volatiles

Until the mid-19th century, the heating, fining (removal of gasses and bubbles) and homogenization processes occurred in discrete batch processes. Today, all three steps are performed continuously in the melter.

The best measure of theoretical melting energy is the sum of the reaction heat, heat for melting and heat for volatile reaction products. This sum can be calculated for each specific type of glass. For soda lime glass (container glass) with no cullet, the theoretical melting energy is about:

Reaction:	209 Btu/lb	487 kJ/kg
Melt:	757 Btu/lb	1,760 kJ/kg
Volatiles:	124 Btu/lb	289 kJ/kg
Total:	1,102 Btu/lb	2,563 kJ/kg

When 100% cullet (recycled glass) is used as feed, it is not necessary to include the heat of reaction or the heat of driving off the volatiles. Thus, the theoretical melting energy for 100% soda lime cullet is:

Melt:	757 Btu/lb	1,760 kJ/kg
-------	------------	-------------

Benchmarks of Actual Melting Energy

Actual glass manufacturing processes use additional energy, since energy is lost:

- Through the walls of the furnace
- In the exhaust gasses

According to (Glass Manufacturing”, Pollution Prevention and Abatement Handbook, World Bank Group, July 1998) actual source energy requirements for glass production range from 1,591 to 2,580 Btu/lb (3.7 to 6.0 GJ per metric ton). More specifically, Beerkins (“Future Industrial Glass Melting Concepts”, Proceedings of the International Congress on Glass, Edinburgh, Scotland, 6/2001, pp 180 – 192.) shows actual source energy requirements ranging from 1,290 to 3,010 Btu/lb (3.0 to 7.0 GJ per metric ton) for a variety of modern glass melting technologies.

Johns Manville Melter Efficiency

According to measured data, each melter produces about 170 tons of glass per day. Average gas consumption by each melter is about 31 mcf/hr. Thus, gas consumption by each melter is about:

$$\begin{aligned} 31 \text{ mcf/hr} \times 24 \text{ hr/day} &= 744 \text{ mcf/day} \\ 744 \text{ mcf/day} \times 365 \text{ day/yr} &= 271,560 \text{ mcf/yr} \\ 744 \text{ mcf/day} / 170 \text{ tons/day} &= 4.38 \text{ mcf/ton-glass} \\ 4.38 \text{ mcf/ton} \times 1 \text{ mmBtu/mcf} / 2,000 \text{ lb/ton} &= 2,190 \text{ Btu/lb-glass} \end{aligned}$$

These numbers are consistent with measured gas consumption of about 700 mcf/day and the annual gas consumption estimate in the Estimated Natural Gas Breakdown.

According to plant personnel, each zone adds 200 kW of electric booster heat. If so, the electrical heat added in each furnace is about:

$$\begin{aligned} 200 \text{ kW/zone} \times 5 \text{ zones} \times 24 \text{ hours/day} &= 24,000 \text{ kWh/day} \\ 24,000 \text{ kWh/day} \times 3,412 \text{ Btu/kWh} &= 81.9 \text{ mmBtu/day} \\ 24,000 \text{ kWh/day} \times 365 \text{ days/year} &= 8,760,000 \text{ kWh/yr} \\ 24,000 \text{ kWh/day} / 170 \text{ tons/day} &= 141 \text{ kWh/ton-glass} \\ 24,000 \text{ kWh/day} / (170 \text{ tons/day} \times 2,000 \text{ lb/ton}) &= 0.0706 \text{ kWh/lb-glass} \end{aligned}$$

Annual plant electricity use is about 108,000,000 kWh/yr. Thus, booster heating represents about 16% of plant energy use. This number is inconsistent with the estimate of 230,000 kWh/yr in the Estimated Electricity Use Breakdown.

Using the Lower Heating Value for natural gas, the total natural gas and electrical heat energy added to each melter is about:

$$\begin{aligned} (744 \text{ mcf/day} \times 1 \text{ mmBtu/mcf}) + 81.9 \text{ mmBtu/day} &= 826 \text{ mmBtu/day} \\ 826 \text{ mmBtu/day} / (170 \text{ tons/day} \times 2,000 \text{ lb/ton}) &= 2,429 \text{ Btu/lb} \end{aligned}$$

Thus, using 100% batch for soda lime glass as the reference, melter efficiency on a site basis is about:

$$1,102 \text{ Btu/lb} / 2,429 \text{ Btu/lb} = 45\%$$

According to Plum et al. ("Advanced Heating Techniques for Glass Melting, Plum, Graf, Beckers, Doersma, Dungen, Frijns and Verbakel, Eindhoven University of Technology, Netherlands, March 5, 2002), the electricity required to manufacture oxygen in typical modern processes is about 11.33 kWh/mcf (0.4 kWh/m³).

Average oxygen flow by each melter is about 80 mcf/hr. Thus, daily and annual electricity consumption for producing oxygen for each melter is about:

$$\begin{aligned} 80 \text{ mcf/hr} \times 11.33 \text{ kWh/mcf} \times 24 \text{ hr/day} &= 21,753 \text{ kWh/day} \\ 21,753 \text{ kWh/day} \times 365 \text{ day/yr} &= 7,940,064 \text{ kWh/yr} \\ 21,753 \text{ kWh/day} / (170 \text{ tons/day} \times 2,000 \text{ lb/ton}) &= 0.0640 \text{ kWh/lb} \end{aligned}$$

This estimate of about 16,000,000 kWh per year for oxygen plant electricity use is not consistent the estimate of annual oxygen plant electricity use of about 54,350,000 kWh/year in the Estimated Electricity Use Breakdown.

Assuming the electricity generation is 33% efficient, the total unit source energy use, including oxygen production, is about:

$$\begin{aligned} 0.0640 \text{ kWh/lb} + 0.0706 \text{ kWh/lb} \times 3,412 \text{ Btu/kWh} / 33\% &+ 2,190 \text{ Btu/lb} = \\ 1,392 \text{ Btu/lb} + 2,190 \text{ Btu/lb} &= 3,582 \text{ Btu/lb} \end{aligned}$$

Oxygen:	0.0640 kWh/lb x 3,412 Btu/kWh / 33% =	662 Btu/lb (18%)
Elec Boost:	0.0706 kWh/lb x 3,412 Btu/kWh / 33% =	730 Btu/lb (20%)
Gas:	2,190 Btu/lb	= 2,190 Btu/lb (62%)
Total:		3,582 Btu/lb

This is at the higher end of Beerkins (“Future Industrial Glass Melting Concepts”, Proceedings of the International Congress on Glass, Edinburgh, Scotland, 6/2001, pp 180 – 192.) range of 1,290 to 3,010 Btu/lb.

The unit energy cost of melting, including electricity to generate oxygen, is about:

Oxygen:	0.0640 kWh/lb x \$0.041 /kWh =	\$0.002624 /lb (14%)
Elec Boost:	0.0706 kWh/lb x \$0.041 /kWh =	\$0.002895 /lb (15%)
Gas:	2,190 Btu/lb / 10 ⁶ Btu/mcf x \$6.10 /mcf =	\$0.013359 /lb (71%)
Total:		\$0.0189 /lb

$$\$0.0189 \text{ /lb} \times 2,000 \text{ lb/ton} = \$37.76 \text{ /ton}$$

Process Heating Analysis

During this study we analyzed total energy use for the plant and the energy used for process heating equipment such as glass melting furnaces, fore hearths, various ovens, mat line and thermal oxidizer. The data obtained and used for this analysis represents values for a short duration but it reflects general trends in energy use for the equipment considered.

The plant purchases natural gas and electricity to meet the energy needs of the manufacturing operations. Detail analysis of natural gas usage in the plant equipment is discussed in an earlier section (Section III Whole Plant Utility Analysis). This section

concludes that the plant natural gas use is slightly more than 1 million Thousand Standard Cubic Feet (MCF) for the year 2003, the latest year for which full data is available for this report. At an average gas cost of \$7 per MCF or Million Btu (MM Btu), total cost of energy used for fired system in the plant is in excess of \$7 million per year. Figure PH 1 shows an overview of the natural gas energy use distribution for the heating equipment in the plant. .

Figure PH 1. Natural gas energy use distribution for the plant equipment

The data show that the two furnaces and their fore hearths (FH) (11 and 12) consume 80% of the total natural gas used in the plant. These furnaces and fore hearths are operated at very high temperature with exhaust gases being discharged at a temperature in excess of 2500 deg. F. On the other hand the third large energy user equipment, Mat oven, is operated at relatively low temperature (less than 500 deg. F.). Exhaust gas (air) from the oven is raised to approximately 1400 deg. F. in a thermal oxidizer (TO). However the TO system includes a regenerative heat recovery to preheat the oven exhaust gases to raise its temperature to very close to the oxidation process resulting in very small amount of fuel in the TO.

The glass melting furnaces use oxy-fuel firing in which pure oxygen is for combustion of natural gas. As shown in Figure PH 2 below, use of oxy-fuel firing results in approximately 52% fuel savings since the available heat with use of oxygen to replace air increases from 34.4 % to 71.2%. This results in fuel savings of 52%. It should be noted that this is based on use of cold air. With use of preheated air for combustion in conventional furnaces the available heat is 54%. Hence use of oxy-fuel to replace preheated air system in glass melting results in 31 to 32 % fuel savings.

Calculations for Savings - O2 Enrichment		
	Current	New
% O2 in comb. air	21%	100%
Flue gas temp. (F)	2425	2425
Av ht. With std air	34.4	34.4
Av ht. With enriched air	34.4	71.2
Fuel savings (%)	Base	52%

Parameter	Current
Furnace flue gas temp. (F)	2,425
Percent O2 in flue gases	2
% Excess air	9.61
Combustion air temperature (F)	1,200
Av. Heat (%)	54.02

Figure PH 2 Fuel savings with use of oxy-fuel burners in glass melting

At an annual fuel consumption rate of 255,600 MCF for melter #11 and 236,900 MCF for melter #12 or total of 492,500 MCF combined for the two melting furnace, heat loss in flue gases from these oxy-fuel fired furnaces is $(100 - 71.2 =)$ 29.8 % of the fuel input or 146,765 MCF gas or \$1.027 million per year.

The fore hearths use conventional air-fuel combustion with cold or ambient temperature air. The two fore hearths discharge flue gases at temperature varying from 2300 deg. F. to 2550 deg. F. If we take an average temperature of 2425 deg. F., the available heat for the combustion is 34.5% and the heat loss is $(100-34.5=)$ 65.5% of the heat input into the fore hearths. Based on annual gas consumption of 140,800 MCF for #12 fore hearth and 98,000 MCF for #12 fore hearth, the annual heat loss is 238,800 MCF gas or \$1.67 million per year. Note that these losses represent more than 38% of the TOTAL gas used by the plant and an energy loss costing the company more than \$2.7 million per year in the gas cost.

The mat line gas use is divided into two major areas: the mat line ovens and TO or incinerator. Annual gas use for mat line oven is 106,600 MCF and for the TO is 22,000 MCF. The exhaust gases from the oven are fed to the TO and they together with combustion products from the TO burners are discharged at relatively low temperature of 350 deg. F. It is difficult to define efficiency of this system because it uses a large amount of dilution air in the oven and the oven gases are preheated in a regenerative heat recovery device. However it is possible to estimate total heat loss from the system by measuring temperature and oxygen content of the exhaust gases.

Actual measurement of O2 in the TO exhaust gases was made during the plant visit and following readings were noted.

Location	Oxygen (%)	CO (ppm)
Incinerator inlet	17.4 %	49
Incinerator outlet	16.9 %	108

The mass balance for incinerator gases is used to estimate additional flue gases from the incinerator burner. The calculations show that 7% additional flue gases with 2% O₂ must be added to reduce O₂ from 17.4% to 16.4%.

Total fuel consumption of the oven-incinerator system is 128,600 MCF while the O₂ content of 16.4% from the incinerator indicates 400% excess air as shown in the Figure below.

Combustion air for stoichiometric combustion of N. gas = 128,600 MCF gas/yr x 10,000 scf air = 1,286 x 10⁶ scf/yr. With 400% excess air total volume of flue gases will be = 1,286 x 10⁶ x (1 + 400/100) = 6,430 x 10⁶ scf/year.

Total heat lost in incinerator exhaust gases depend on the exhaust gas temperature. The measurement showed 350 deg. F. temperature. At this temperature total heat content above 60 deg. F. = 0.02 x (350 – 60) = 5.8 Btu/scf.

Hence estimated heat content of exhaust gases = 6,430 x 10⁶ x 5.8 = 37,294 MM Btu/year. This is about 30% of the total heat in gas input for the system.

A check on these figures can be obtained by calculating available heat for exhaust gases from n. gas combustion being discharged at 350 deg. F. with 400% excess air. As shown in Figure ML-2 the available heat is approximately 68% at these conditions. This indicates that theoretically 32% of the heat is contained in exhaust gases. In actual operation the system heat is lost by radiation and convection from the ducts, incinerator and regenerator walls etc. and also a little variation in exhaust gas oxygen may introduce errors.

It can be safely concluded that 30% of the heat used in the mat oven and incinerator is lost in flue gases. This heat is equal to 37,000 MM Btu/year with value of 37,000 MM Btu/year x \$7 per MM/Btu or \$259,000. This is considerably less than the heat content in flue gases from the fore hearth and furnaces but not negligible.

Figure PH 3 Available heat for natural gas fired heating systems

Recovery Methods

Several possible methods of reducing, recovering or recycling the heat in exhaust gases from heating equipment (furnaces, fore hearth and mat oven) were considered during this assessment. There are two major sources of waste heat in this plant. Each of them are discussed below.

Heat Recovery From Incinerator Exhaust Gases

As mentioned above the incinerator exhaust gases contain approximately 37,000 MM Btu/year at approximately 350 deg. F. This is considered as low grade heat and cannot be easily recovered. Following methods were considered for heat recovery:

- (i) Combustion air preheating for the incinerator burners
- (ii) Recycling of heat by using exhaust gases into the mat oven zones to replace cold make up air and reduce heat for raising temperature of make-up air into the oven
- (iii) Water heating by using incinerator flue gases and using hot water for process uses in the plant or for absorption chilling system.

After review of technical and economic issues it is concluded that the first two methods are economically or technically feasible at this time. However the plant should consider the third method as a possible method of heat recovery from incinerator exhaust gases. Reasons for this conclusion are given below.

Combustion Air Preheating

With flue gas temperature of 350 deg. F. it may be possible to heat combustion air to maximum 250 deg. F. Since the exhaust gases are discharged at 350 deg. F. from the system, and the burners are operated with about 10% excess air potential, energy savings with use of preheated air is 4.2%. The calculations are shown in Figure ML-3. The incinerator fuel consumption is burners for the is only 350 deg. F. a

Calculation for Savings - Efficient Combustion		
	Current	New
Furnace flue gas temp. (F)	350	350
Percent O2 in flue gases	2	2
% Excess air	9.61	9.61
Combustion air temperature (F)	80	250
Av. Heat (%)	82.84	87.01
Fuel savings (%)	Base	4.18%

Figure ML-3 Calculations for savings through combustion air preheating

Total energy use for the incinerator is 22,000 MM Btu/year. Savings due to use of preheated air would amount to 924 MM Btu/year with economic value of \$924 MM Btu/year x \$7/MM Btu = \$6,468 per year.

This heat recovery method would require installation of an air-air heat exchanger, piping of hot air from the heat exchanger to the burners and some changes in the burner air-fuel ratio control system to assure proper air-fuel ratio for the burners. The heat exchanger cost plus installation is estimated to be in excess of \$20,000 and additional cost of piping etc. could exceed \$5000. Hence expected payback would be more than 4 years. This is very optimistic estimate and actual payback period could be as long as 6 to 7 years. Hence this possibility is not considered further in this report.

Recycling Of Incinerator Exhaust Gases Into The Oven

The incinerator exhaust air is at 350 deg. F. (max) while the oven zone temperature varies from 241 deg. F. to 397 deg. F. Differential temperature between the incinerator exhaust gas and lowest zone temperature is about 110 deg. F. Considering the distance of ducts from incinerator exhaust to the first zone where the temperature is 241 deg. F. and possibility of heat losses due to outside exposure of the duct, even with the insulated duct, it would not be economical to justify this project. Hence this was not considered further.

Water Heating Using Incinerator Exhaust Gases

The incinerator exhaust gases are relatively clean and can be used to heat water that can be used in the plant or for use in an absorption chiller system. Depending on the requirement, the water heating can be for once-through flow system in which process make up water from the well or other sources can be heated by a water to air heat exchanger or by a direct contact heat exchanger. If the water is recirculating water from a process or as in case of hot water used for absorption chillers it is necessary to use a water-air heat exchanger. Two possible systems are shown in Figure PH 5 below.

Figure PH 5 Possible methods of heat recovery from incinerator exhaust gases

Typical use of an indirect heat exchanger as shown in Figure PH 5 is economically justifiable when it offers 50% to 60% heat recovery efficiency. Using an efficiency of 55%, the total recovered heat would be 55% of the recoverable heat 37,000 MM Btu/year or 20,350 MM Btu/year with an economic value of \$142,450 per year. Further details of this heat recovery project are given in the recommendations section.

A direct contact heat exchanger as shown in Figure PH 5 is economically justifiable with as high as 85% heat recovery efficiency. In this case hot water can be obtained at as high as 180 deg. F. Using an efficiency of 85%, the total recovered heat would be 31,450 MM Btu/year with an economic value of \$220,150 per year. This is considerably more economical use of incinerator waste heat if hot water is required for once-through flow (non-recirculating) applications. In most cases the total system cost can be justified in less than a one year payback period. However we have not been able to identify such an application in this plant.

Based on preliminary calculations we have made a recommendation of using energy from thermal oxidizer to heat water that can be used a supplementary heat source in absorption chiller system.

Heat Recovery From Melting Furnaces And Fore Hearths

As mentioned earlier in this section a large amount of heat is available for recovery from the melting furnaces and their fore hearths. The heat is in the form of hot gases and it has following characteristics that require special considerations before an appropriate heat recovery system is recommended.

- (i) The gases are at very high temperature, in excess of 2200 deg. F.

- (ii) These gases contain particles as well as condensable volatiles that will condense at various temperatures depending on the composition of charge material.
- (iii) Gases from fore hearths are discharged at several locations over a relatively long (in excess of 50 ft) distance. It will be necessary to collect these gases to avoid use of multiple heat recovery devices.
- (iv) Flow rate of these gases vary however the variation is not very large (say a turn down of more than 5 to 1).
- (v) It is necessary to remove particles and condensable materials before they are discharged into the atmosphere.

In the past the furnace flue gases have been used to preheat combustion air in a regenerator and the industry has learned to deal with the issues of condensation of volatile materials, deposits of particles etc. in a regenerator.

At this plant the melting furnaces use oxy fuel burners hence combustion air preheating is not possible. It is not practical to preheat oxygen to recover the exhaust gas heat. At this time most U.S. glass plants using oxy-fuel fired melting furnaces do not recover flue gas heat.

Several options are considered for flue gas heat recovery from melting furnaces and fore hearths. A list of options considered is given in Figure PH 6.

1	Heat recovery - Melting furnaces and Forehearths
1.1	Heat air in radiation recuperator
1.1.1	Use hot air for ovens
1.1.2	Use hot air for water heating /chiller
1.2	Use flue gases for water heating
1.2.1	Use hot water for chillers
1.3	Use flue gases for batch-charge preheating
1.4	Use flue gases for steam generation
1.4.1	Use steam for chiller
1.4.2	Use steam for plant use

Figure PH 6 List of possible methods of heat recovery from glass melter flue gases

As mentioned earlier the flue gases are hot and contain contaminants that would condense as the gas temperature drops. The plant advised us that the gases must be maintained above 800 deg. F. to avoid condensation of volatile materials and deposit of solid material on heat exchanger surface. Deposits produced from condensation of the volatile material are difficult to remove. Hence it is necessary to avoid heat transfer systems that do not use large cold surfaces where the volatiles may deposit. This eliminates use of a convection type heat exchanger where flue gases are used to heat air or water circulated in multiple tubes with flue gases passing over the tube outside surfaces.

A radiation type recuperator where radiation heat transfer from flue gases containing CO₂ and H₂O can be used to heat air has been widely used for heat recovery from glass

melting furnaces. In this case we cannot use the air for fuel combustion. However it is possible to use hot air to supply heat to the mat oven.

Due to long distances, large duct sizes and possible miss-matching of heat supply vs. heat demand in mat oven we did not pursue this option.

Option of using clean hot air for heating water using a heat exchanger was considered in detail. The hot water can be used to supply heat for an absorption chiller that produces chilled water used in the plant. An alternate method of using flue gas heat for water heating was considered. In this case two options are considered.

In one option, shown in Figure PH 7, the flue gases are cooled below the volatile condensation temperature and the “warm” gases are used to heat water for use in an absorption chiller. Cooling of gases would form particulates that are not “sticky” and can be removed as solid particles in a medium temperature particulates removal device such as a cyclone (more likely) or high temperature bag house (less likely).

Figure PH 7 Possible method of heat recovery from melter flue gases

The gases from particulate removal device may still contain some particles, however the would not be “sticky”. During our conversation with the heat recovery equipment suppliers we were informed that if the gases are cooled below 600 deg. F. to 800 deg. F., almost all volatiles would precipitate out as particulates that can be removed from heat transfer surfaces by using a device similar to a soot blower. The gases would be cooled by mixing them with ambient air. The “warm” and partially cleaned gases are used for heating recirculating water for use in absorption chillers.

Other option includes heating of recirculating water in a specially designed heat exchanger which is a hybrid of a water heater and air heater and it is installed as part of the flue gas stack for melting furnaces. We had considered using water heating by “wrapping” water coils around the stack however the heat recovery device suppliers warned us that this system could result in low surface temperature for the I.D. of the flue gas stack and result in large amount of deposits of condensable solids in the stack. The special design, shown in Figure PH 8, would use water and air heating so that the flue gases would not be cooled below the volatile condensation temperature. It is then

possible to use the hot air in a heat exchanger to heat more water. Obviously this system is somewhat complex and unproven. We have analyzed a case where combination of air-water heating in a stack type of recuperator with secondary heat recovery from heated air as one of the options.

Figure PH 8 Use of radiation recuperator for heat recovery from melter flue gases

Preheating glass batch or charge material is the most efficient method of recovering heat from furnace flue gases. A variety of heating system configurations are used to preheat the batch or cullet used as charge material. It is necessary to make significant changes or modifications to the existing material handling system to handle higher temperature material. It is also necessary to limit the batch temperature to lower than the volatile temperature of the batch components. For this reason preheating of cullet is preferred since there is very little, if any, danger of release of volatiles. Depending on the degree of preheat and percentage of batch material preheated, expected energy savings vary from 15% to 30% of the current energy use. For the Waterville plant almost all charge material consists of fresh batch with very little cullet. The batch material is extremely fine and it will be very difficult to use conventional methods of heating of solids. Due to these reasons it was agreed not to pursue this possibility.

An option of steam generation using flue gases from air-fuel fired glass melting furnaces has been used commonly in Europe and Japan and the U.S. suppliers have shown willingness to supply such a system. However use of oxy-fuel firing at this plant results in reduces the flue gas heat and steam generation is considered practical due to relatively small amount of heat in flue gases. Total waste heat from the two furnaces is about 146,765 MMM Btu/year or 16 MM Btu/hour. Installation of a boiler for recovering heat from contaminated gases that require special design for keeping the boiler tubes and walls clean, is considered economically unjustifiable. Hence this option is not considered or analyzed further.

All of the above comments apply to fore hearth gases also. However collection of flue gases from fore hearth, particularly with the current fore hearth design is very difficult.

Substantial energy savings (about 52%) can be achieved by using oxy-fuel firing of fore hearth.

With oxy-fuel firing the amount of heat in flue gases would drop from current estimated value of 238,800 MM Btu/year to 114,240 MM Btu/year. This still represents an annual fuel cost of \$799,680. If the flue gases are discharged from limited number of openings, perhaps 3 to 4, then it is possible to collect the flue gases and use them for water heating for use in absorption chillers.

Based on these consideration we have made following recommendations in the area of waste heat recovery. They are described in the Assessment Recommendation section of the report.

- (i) Use of the incinerator exhaust air for water heating and use of hot water for chiller system or to preheat water for the plant use
- (ii) Use of melting furnace flue gases for air and water heating and use of hot water for chiller system
- (iii) Use of outdoor air as combustion air and as make-up air in the mat oven to avoid negative pressure in the plant as well as to reduce plant air heating during winter time.
- (iv) Reduction of air by eliminating or reducing air leaks for the mat oven.

Compressed Air System Analysis

Equipment

The plant is equipped with one 4-stage 700-hp Ingersoll-Rand reciprocating compressor with a synchronous motor, a 200-hp Atlas Copco rotary screw compressor, a 75-hp Ingersoll-Rand “Serra” rotary-screw compressor and a 40-hp Atlas Copco rotary screw compressor. Other compressors exist in the plant; however, management indicates they are not used. Compressors are distributed throughout the plant and deliver air to a single compressed air header. A schematic of the compressed air system is shown in the diagram below.

All compressors are water-cooled via the plant's main chilled water system. A central pressure monitor is present in the plant to record plant pressure. Room temperature air, measured to be about 77 F in each compressor location, is used for compression.

700-hp Ingersoll-Rand Reciprocating Compressor

We logged the current draw of the four-stage 700-hp Ingersoll-Rand reciprocating compressor between the hours of 10:00 AM and 3:00 PM. The data, shown below, indicate that the compressor runs at full load most of the time and at 75% of full load some of the time.

700-hp Ingersoll-Rand Reciprocating Compressor

According to these data, the highest current draw was about 596 Amps and the lowest was about 418 Amps. Inherent to synchronous motor operation, these motors generate or absorb reactive power resulting from other inductive loads located in the facility. As such, synchronous motors have a power factor of 1 kW/kVA. This characteristic provides natural power factor correction for the facility. Therefore the highest and lowest power draw is about:

$$\begin{aligned} \text{Highest:} & \quad 596 \text{ A} \times 470 \text{ V} \times 1 \text{ kVA/kW} \times \sqrt{3} / 1,000 \text{ W/kW} = 485 \text{ kW} \\ \text{Lowest:} & \quad 418 \text{ A} \times 470 \text{ V} \times 1 \text{ kVA/kW} \times \sqrt{3} / 1,000 \text{ W/kW} = 340 \text{ kW} \end{aligned}$$

According to these data, the compressor was loaded about 84% of the time and operating at 75% of full-load about 16% of the time. Therefore, the average power draw during the logged period is about:

$$84\% \times 485 \text{ kW} + 16\% \times 340 \text{ kW} = 462 \text{ kW}$$

Assuming the compressor motor is about 95% efficient, the fraction of full-load power draw is about:

$$(462 \text{ kW} / 95\%) / (700 \text{ hp} \times 0.75 \text{ kW/hp}) \times 100\% = 93\%$$

According to nameplate data, the 700-hp Ingersoll-Rand reciprocating compressor has an output capacity of 3,500 scfm. In multistage reciprocating compressors, the fraction of full-load capacity is approximately equal to the fraction of full-load power draw. Therefore, the compressed air output of the 700-hp compressor is about:

$$3,500 \text{ scfm} \times 0.93 = 3,255 \text{ scfm}$$

200-hp Atlas Copco Rotary-Screw Compressor

We logged the current draw of the 200-hp Atlas Copco compressor between 9:10 AM and 2:59 PM, as shown below. According to these data, the compressor appears to be operating in modulation mode.

Over the data period, the compressor modulated between about 192 Amps and 165 Amps. According to a spot measurement, the compressor operates at about 480 Volts. According to National Electrical Manufacturers Association (NEMA) standard-efficiency 200-hp motors have a power factor of about 0.90 kW/kVA when operating near full-load power draw. If so, the power draw of the compressor varied between about:

$$\text{High: } 192 \text{ A} \times 480 \text{ V} \times 0.90 \text{ W/VA} \times \sqrt{3} / 1,000 \text{ W/kW} = 144 \text{ kW}$$

$$\text{Low: } 165 \text{ A} \times 480 \text{ V} \times 0.90 \text{ W/VA} \times \sqrt{3} / 1,000 \text{ W/kW} = 123 \text{ kW}$$

The average power draw was about 138 kW.

According to nameplate data, the 200-hp Atlas Copco utilizes a 217-hp motor. According to NEMA, 200-hp standard efficiency motors are about 93% efficient. If so, the full load power draw would be about:

$$217 \text{ hp} / 93\% \times 0.75 \text{ kW/hp} = 175 \text{ kW}$$

Therefore, the average fraction of full-load power draw is about:

$$\text{FP}_{\text{avg}} = 138 \text{ kW} / 175 \text{ kW} = 0.91$$

According to Atlas Copco literature, a GA160W compressor operates at about

GA160W Pressure (psig) vs Output Capacity (scfm)

$$OC = -3.0526 \times P + 1285.1$$

Where OC is output capacity (scfm) and P is pressure (psig). According to compressor controls, the compressor is operating at about 107 psig. Therefore, the full-load output capacity of the compressor is about:

$$(-3.0526 \text{ scfm/psig} \times 107 \text{ psig}) + 1285.1 \text{ scfm} = 958 \text{ scfm}$$

The fraction of peak output capacity (FC) at which the compressor operates is a function of the average fraction of full-load power (FP). For compressors operating in modulation mode, the typical relationship is:

$$FC_{\text{avg}} = (FP_{\text{avg}} - .70) / (1 - .70)$$

If so, the average fraction of full-load output capacity is about:

$$FC_{\text{avg}} = (0.91 - 0.70) / (1 - 0.70) = 0.70$$

$$0.70 \times 958 \text{ scfm} = 671 \text{ scfm}$$

75-hp Ingersoll-Rand Sierra Rotary Screw Compressor

We logged the current draw of the 75-hp Ingersoll-Rand “Sierra” reciprocating compressor between the hours of 10:00 AM and 3:00 PM. According to logged data, the compressor operates loaded about 15% of the time and unloaded about 85% of the time.

According to NEMA, 200-hp standard efficiency motors are about 93% efficient. Most compressors motors operate at about 105% of rated power when fully loaded and 60% of rated power when unloaded. If so, the loaded, unloaded and average power draw is about:

Loaded: $(75 \text{ hp} \times 0.75 \text{ kW/hp}) \times 1.05 / 93\% = 63 \text{ kW}$
Unloaded: $(75 \text{ hp} \times 0.75 \text{ kW/hp}) \times 0.60 / 93\% = 37 \text{ kW}$
Average: $85\% \times 37 \text{ kW} + 15\% \times 63 \text{ kW} = 41 \text{ kW}$

According to NEMA, 200-hp standard efficiency motors are about 93% efficient; therefore, the average fraction of full-load power draw is about:

$$FP_{avg} = 41 \text{ kW} / 63 \text{ kW} = 0.65$$

The fraction of peak output capacity (FC) at which the compressor operates is a function of the average fraction of full-load power (FP). For a compressor operating in load/unload mode, the typical relation is:

$$FC_{avg} = (FP_{avg} - .60) / (1 - .60)$$

If so, the average fraction of full-load output capacity is about:

$$FC_{avg} = (0.65 - 0.60) / (1 - 0.60) = 0.13$$

Compressors can typically generate about 4.2 scfm per horsepower. At 75 hp, the output would be about 315 scfm. If so, the average quantity of compressed air generated is about:

$$0.13 \times 315 \text{ scfm} = 41 \text{ scfm}$$

40-hp Atlas Copco

We could not measure the power draw of the 40-hp Atlas Copco compressor. However, assuming that on average the compressor motor is 70% loaded and 93% efficient, the average power draw would be about:

$$40\text{-hp} \times 0.75 \text{ kW/hp} \times 70\% / 93\% = 23 \text{ kW}$$

The peak power draw would be about:

$$40\text{-hp} \times 0.75 \text{ kW/hp} / 93\% = 32 \text{ kW}$$

Assuming the compressor is operating in load unload mode, the compressed air output would be about:

$$FC_{avg} = (0.70 - 0.60) / (1 - 0.60) = 0.25$$

Compressors can typically generate about 4.2 scfm per horsepower. At full load power of 40 hp, the output would be about 168 scfm. If so, the average quantity of compressed air generated is about:

$$0.25 \times 168 \text{ scfm} = 42 \text{ scfm}$$

Total

Based on the previous analysis, the total compressed air output of the compressors during normal plant operation is about:

$$3,325 \text{ scfm} + 671 \text{ scfm} + 41 \text{ scfm} + 42 \text{ scfm} = 4,079 \text{ scfm}$$

Assuming the compressors operate continuously, the total electricity use and cost is about.

$$\begin{aligned} 462 \text{ kW} + 138 \text{ kW} + 41 \text{ kW} + 23 \text{ kW} &= 664 \text{ kW} \\ 664 \text{ kW} \times 8,760 \text{ hours/year} &= 5,816,640 \text{ kWh/year} \\ 5,816,640 \text{ kWh/year} \times \$0.039 / \text{kWh} &= \$226,850 / \text{year} \end{aligned}$$

The average unit cost of compressed air is about:

$$\$226,850 / \text{year} / (4,079 \text{ scf/min} \times 60 \text{ min/hr} \times 8,760 \text{ hr/year}) = \$0.106 / \text{thousand scf}$$

Compressed Air and Material Transport

Most of the time, the plant employs dense-phase pneumatic transport to move material from rail cars into silos and from mixers to day bins. The system uses large quantities of compressed air and requires that compressed air be transported from the main plant to the batch house. This in turn requires the use of several desiccant dryers with large parasitic loads to avoid condensation. In addition, according to management, this system encounters frequent operational problems.

An alternative is to employ dilute phase pneumatic transport. Our preliminary understanding is that dilute phase pneumatic transport would require an additional 75-hp compressor in the batch house. However, the system would also use less compressed air, avoid the need to transport compressed air between the main plant and the batch house and the need for desiccant dryers, and reduce many operational problems. Unfortunately, we were not able to collect enough data to quantify the savings potential of this option.

III. Whole Plant Utility Analysis

Electricity, Natural Gas and Water Use

Plant 1 electricity, natural gas and water use from 2003 are shown below. The trends show that monthly energy and water usage is nearly constant throughout the year.

Date	(kWh/mo)	Electricity (\$/mo)	(\$/kWh)	(Mcf/mo)	Nat Gas (\$/mo)	(\$/Mcf)	(kgal/mo)	Water (\$/mo)	(\$/kgal)
1/31/2003	10,215,500	416,282	0.041	95,376	540,982	5.67	1,776	26,020	14.65
2/28/2003	9,250,000	404,247	0.044	86,365	548,499	6.35	1,784	26,230	14.70
3/31/2003	10,458,000	436,726	0.042	89,591	786,071	8.77	1,404	20,675	14.73
4/30/2003	10,405,500	438,696	0.042	85,955	515,408	6.00	1,662	24,417	14.69
5/31/2003	11,396,091	470,659	0.041	90,735	546,326	6.02	1,868	27,270	14.60
6/30/2003	11,352,379	474,870	0.042	85,446	585,049	6.85	1,913	27,476	14.36
7/31/2003	11,697,452	479,976	0.041	88,316	537,417	6.09	2,130	30,676	14.40
8/31/2003	11,187,057	457,551	0.041	84,501	469,741	5.56	2,109	31,052	14.72
9/30/2003	11,266,405	428,123	0.038	87,212	507,940	5.82	1,929	28,375	14.71
10/31/2003	11,348,705	426,711	0.038	92,205	481,531	5.22	1,728	27,379	15.84
11/30/2003	10,803,230	414,520	0.038	89,908	480,985	5.35	1,788	26,333	14.73
12/31/2003	10,873,181	432,753	0.040	95,513	533,428	5.58	1,857	28,051	15.11
Total	130,253,500	5,281,114	0.041	1,071,123	6,533,377	6.10	21,948	323,954	14.76

Avoided Costs

The following unit costs will be used in this report to calculate the cost savings from reducing electricity, natural gas and water use.

Electricity: \$0.039 /kWh

Natural Gas: \$7.00 /Mcf

Water: \$14.76 /kgal

Estimated Electricity Use Breakdown

The plant supplied a one-line electrical diagram and equipment list. Based on this information, an estimated electricity use breakdown is shown below.

Equipment	Qty	Rated Power (hp)	Rated Power (kW)	Fraction Loaded	Average Power (kW)	Run Time (hrs/yr)	Annual Energy (kWh/yr)	Fraction Annual Energy	Annual Cost (\$/yr)
Sub 1									
Hot Well Pumps	2	60	45	50%	23	8,760	197,100	0.2%	8,081
Cold Well Pumps	5	200	150	50%	75	8,760	657,000	0.5%	26,937
Cooling Pond Cooling Tower Fans	4	160	120	81%	97	8,760	851,472	0.7%	34,910
Transfer Well Pumps	5	125	94	48%	45	8,760	394,200	0.3%	16,162
Sub 2									
IR Air Compressor	1	700	525	80%	420	8,760	3,679,200	2.8%	150,847
Sub 5									
Dielectric Oven Zone 1 (East)	1	200	150	100%	150	8,760	1,314,000	1.0%	53,874
Binder Room	1	50	38	50%	19	8,760	164,250	0.1%	6,734
Lanly Tunnel Oven	2	150	113	80%	90	8,760	788,400	0.6%	32,324
Sub 6									
Abatement ID Fan	1	350	263	90%	236	8,760	2,069,550	1.6%	84,852
Batch House	1	150	113	67%	75	8,760	660,285	0.5%	27,072
MMC 66C Batch Feeder	1	30	23	67%	15	8,760	132,057	0.1%	5,414
Zone 5 Boost / Hi Pressure Pump	1	310	233	81%	188	8,760	1,649,727	1.3%	67,639
Finishing	1	100	75	80%	60	8,760	525,600	0.4%	21,550
Pre-Bake Ovens	7	455	341	77%	263	8,760	2,301,800	1.8%	94,374
Electrode Cooling Water System	1	50	38	50%	19	8,760	164,250	0.1%	6,734
Furnace Bubbler Cooling Blower	1	15	11	67%	8	8,760	66,029	0.1%	2,707
Forehearth Blowers	2	250	188	40%	75	8,760	657,000	0.5%	26,937
Tempered Water Pump	1	200	150	40%	60	8,760	525,600	0.4%	21,550
Hot Well Pumps	1	110	83	45%	37	8,760	325,215	0.2%	13,334
9211 Boost MCC	1	90	68	39%	26	8,760	230,607	0.2%	9,455
9211 ECWS MCC	1	40	30	38%	11	8,760	99,864	0.1%	4,094
Sub 10									
Compactor	1	25	19	80%	15	8,760	131,400	0.1%	5,387
Chilled Water Pumps	2	400	300	40%	120	8,760	1,051,200	0.8%	43,099
Chillers	2	600	450	40%	180	8,760	1,576,800	1.2%	64,649
Scrap Conveyors	1	15	11	80%	9	8,760	78,840	0.1%	3,232
Coldwell Pumps	2	150	113	40%	45	8,760	394,200	0.3%	16,162
Cooling Tower Fan	3	60	45	53%	24	8,760	208,926	0.2%	8,566
Glycol Pump & Misc Equipment	1	10	8	50%	4	8,760	32,850	0.0%	1,347
Sub 11A									
10 Bushings - Leg 1	10	730	548	75%	411	8,760	3,597,075	2.8%	147,480
10 Bushings - Leg 2	10	730	548	75%	411	8,760	3,597,075	2.8%	147,480
10 Bushings - Leg 3	10	730	548	75%	411	8,760	3,597,075	2.8%	147,480
10 Bushings - Leg 4	10	730	548	75%	411	8,760	3,597,075	2.8%	147,480
Basement Exhaust Fans	6	450	338	53%	179	8,760	1,566,945	1.2%	64,245
Air Handling Units	3	375	281	80%	225	8,760	1,971,000	1.5%	80,811
Scrap Conveyors	1	170	128	59%	75	8,760	658,971	0.5%	27,018
Misc. Equipment	1	50	38	62%	23	8,760	203,670	0.2%	8,350
Sub 11B									
10 Bushings - Leg 5	10	730	548	75%	411	8,760	3,597,075	2.8%	147,480
10 Bushings - Leg 6	10	730	548	75%	411	8,760	3,597,075	2.8%	147,480
10 Bushings - Leg 7	10	730	548	75%	411	8,760	3,597,075	2.8%	147,480
10 Bushings - Leg 8	10	730	548	75%	411	8,760	3,597,075	2.8%	147,480
Leg 7 & Leg 8 Equip + Misc Equipment	1	50	38	50%	19	8,760	164,250	0.1%	6,734
Sub 14									
Oxygen Plant	1	11,990	8,993	69%	6,205	8,760	54,354,267	41.7%	2,228,525
Estimated Totals							108,623,124	83.4%	4,453,548
Estimated Other							21,630,376	16.6%	886,845
Actual Totals							130,253,500	100.0%	5,281,114

Estimated Natural Gas Use Breakdown

The plant supplied a one-line natural gas use diagram with fraction of use by major branch. Based on this information, an estimated natural gas breakdown is shown below.

Equipment	Qty	Rated Input (mmBtu/hr)	Fraction Loaded	Average Input (mmBtu/hr)	Run Time (hrs/yr)	Annual Energy (mmBtu/yr)	Fraction Annual Energy	Annual Cost (\$/yr)
Furnace 11								
9211 Melter	1	40.0	75%	30.0	8,760	262,800	25%	1,603,080
9211 Forehearth	1	30.0	75%	22.5	8,760	197,100	18%	1,202,310
9211 Other	1	5.0	75%	3.8	8,760	32,850	3%	200,385
Furnace 12								
9212 Melter	1	40.0	75%	30.0	8,760	262,800	25%	1,603,080
9212 Forehearth	1	13.0	75%	9.8	8,760	85,410	8%	521,001
9212 Other	1	5.0	75%	3.8	8,760	32,850	3%	200,385
Mat Line								
9214 Oven	1	24.0	45%	10.8	8,760	94,608	9%	577,109
9214 Incinerator	1	4.0	75%	3.0	8,760	26,280	2%	160,308
9214 Other	1	2.0	75%	1.5	8,760	13,140	1%	80,154
Finishing								
1210 Prebakes	1	0.5	75%	0.4	8,760	3,285	0.3%	20,039
1210 Tunnel	1	0.5	75%	0.4	8,760	3,285	0.3%	20,039
1210 TP Chop	1	0.5	75%	0.4	8,760	3,285	0.3%	20,039
Reclaim								
Reclaim	1	0.5	75%	0.4	8,760	3,285	0.3%	20,039
Other								
T-glass and MUA	1	10.0	75%	7.5	8,760	65,700	6%	400,770
Estimated Totals						1,086,678	101%	6,628,736
Actual Totals						1,071,123	100%	6,533,377
Also Have Boilers...								
Erie City #1	1	2.2	50%	1.1	4,000	4,400	0.4%	26,840
American #3	1	1.7	50%	0.9	4,000	3,400	0.3%	20,740
Cleaver Brooks #5	1	4.2	50%	2.1	4,000	8,400	0.8%	51,240
Kewanee (HW)	1	1	50%	0.5	4,000	2,000	0.2%	12,200
American Red Flash (HW)	1	1.3	50%	0.7	4,000	2,600	0.2%	15,860

2003 Measured Natural Gas Breakdown

		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Year
9211	Melter	18,000	16,800	18,000	20,200	23,406	22,320	22,618	22,557	22,601	23,300	22,896	22,915	255,613
	FH	12,900	11,700	11,500	11,100	11,500	11,000	11,330	11,550	12,142	12,150	12,000	12,000	140,872
	Other	3,000	3,000	2,750	2,500	2,500	2,000	2,000	2,000	2,000	2,000	2,500	2,500	28,750
	Sub Total	33,900	31,500	32,250	33,800	37,406	35,320	35,948	36,107	36,743	37,450	37,396	37,415	425,235
9212	Melter	18,300	16,200	18,000	21,000	22,149	21,960	21,874	20,936	19,174	19,545	18,432	19,344	236,914
	FH	9,230	8,433	8,000	7,600	7,917	7,300	7,833	8,097	8,514	8,589	8,274	8,357	98,144
	Dryers	2,823	2,541	2,546	2,546	2,543	2,068	2,497	2,454	3,122	3,180	2,731	2,546	31,597
	Other	2,500	2,500	2,750	2,500	2,500	2,000	2,000	2,000	2,000	2,000	2,500	2,500	27,750
	Sub Total	32,853	29,674	31,296	33,646	35,109	33,328	34,204	33,487	32,810	33,314	31,937	32,747	394,405
Mat Line	Oven	10,136	9,672	9,895	7,949	9,564	10,349	9,953	7,280	8,987	9,072	7,176	6,552	106,585
	Incinerator	1,527	1,368	1,602	1,490	1,382	1,400	1,822	2,003	2,272	2,448	2,238	2,614	22,166
	Other	500	500	500	250	250	100	100	100	100	100	500	500	3,500
	Sub Total	12,163	11,540	11,997	9,689	11,196	11,849	11,875	9,383	11,359	11,620	9,914	9,666	132,251
Finishing	Prebakes	1,469	1,031	1,575	1,892	724	347	402	335	858	928	1,126	864	11,551
	Tunnel	153	137	153	126	121	95	117	1,160	1,267	1,474	1,320	1,193	7,316
	TPChop	147	147	147	133	135	105	1	1	62	126	115	221	1,340
	Reclaim	662	662	713	682	681	501	632	555	653	657	653	690	7,741
	Undistribut	14,029	11,674	11,460	5,987	5,363	3,901	5,137	3,473	3,460	5,994	7,447	12,717	90,642
	Reclaim	662	662	713	682	681	501	632	555	653	657	653	690	7,741
	Sub Total	17,122	14,313	14,761	9,502	7,705	5,450	6,921	6,079	6,953	9,836	11,314	16,375	126,331
Undistributed		14,029	11,674	11,460	5,987	5,363	3,901	5,137	3,473	3,460	5,994	7,447	12,717	90,642
Total		94,269	85,712	88,429	84,486	90,436	85,400	88,428	83,560	85,678	89,692	88,000	93,925	1,058,015

IV. Assessment Recommendations

The recommendations in this section result from plant visits and analyses by the respective team members. Each recommendation seeks to identify and quantify savings opportunities. However, the savings estimated below are preliminary estimates only. Actual savings and costs will vary depending on operating and other conditions during and following implementation.

The savings recommendations are grouped by function as follows.

Water Treatment

- AR WT1: Eliminate High TOC Streams to Maximize Refinery Condensate Reuse
- AR WT2: Reuse Well Water After Heat Exchangers
- AR WT3: Reduce BOD in Evaporator Condensate
- AR WT4: Use Well Water for Contact Applications

Heat Integration

- AR HI1: Preheat Boiler Feed Water with Process Streams
- AR HI2: Preheat Starch Slurry Feed to Converters on A and B Lines
- AR HI3: Heat the DorrClone Wash Water with Condensing Vapors from WGE Unit
- AR HI4: Heat Integrate Feed to 95-Evaporator
- AR HI5: Use Furnace Scrubber Water as Heating Source in Sucrose Plant Evaporator

Process

- AR P1: Reduce Pumping Costs by Trimming Pump Impellers
- AR P2: Use Model to Adjust Air Feed Conditions to Fiber Pre-dryers

Thermal Oxidizer

- AR TO1: Consider Reducing Excess Air in Thermal Oxidizer
- AR TO2: Consider Purchasing a Regenerative Thermal Oxidizer
- AR TO3: Preheat Boiler Feed Water with Thermal Oxidizer Exhaust Gases

Steam System

- AR SS1: Install Turbine / Generator Set and Generate Electricity
- AR SS2: Use Dryer Exhaust Air For Combustion Air To Boiler

Comments

Adjust Water Flow Rate in the Gluten Flash Dryer Scrubber and Purchase Process Simulation Software For Closer Analysis

M-1: Use Melting Furnace Flue Gases For Absorption Cooling

	Annual Savings			Project Cost	Simple Payback
	Resource	CO2 (lb)	Dollars		
Electricity	3,015,936 kWh	6,936,653	\$123,492	\$335,795 or \$574,819	33 months or 56 months

Recommendation by E3M

Analysis

The Waterville, OH plant #1 has two glass melting furnaces designated as 9211 and 9212. Both of these furnaces use natural gas as fuel and oxygen as combustion oxidant ("air"). Flue gases from the furnaces leave furnace at 2400 to 2500 deg. F. Each of this furnace has a fore hearth that is used to maintain molten glass temperature before the glass is used for fiberization process. As mentioned earlier the furnaces and fore hearths use 80% of the natural gas used by the plant and discharge flue gases at very high temperature. Hence a large amount of energy is wasted in exhaust gases. Any action in reducing and recovering heat of exhaust gases would reduce cost of natural gas for the plant. We have discussed several options in an earlier section and concluded that the best method of recovering heat from the furnace and fore hearth exhaust gases is to heat water that can be used for an absorption cooling system to replace existing water chillers that use electricity.

The following analysis is done for two furnaces since each furnace has a single exhaust stack. The same system can be used for exhaust gases from fore hearths if the exhaust gases can be collected and passed through one heat exchanger per fore hearth. This would require significant changes in the fore hearth heating system design. At this time we have not considered fore hearth in detail however we will report possible benefits if the fore hearth gases can be collected after they are mixed with ambient air and then used for water heating.

It is realized that exhaust gases from melting furnaces and fore hearths contain volatile material that condenses at lower temperature and it is impractical to use conventional designs of heat recovery devices such as air preheaters or water heaters. However it is possible to use heat recuperators with relatively large cross sections to avoid condensation and deposit of materials on the cooler surfaces. We have considered use of a specially designed stack recuperator that, as shown in Figure PH 2 – 1 below, can be used to preheat air as well as water. Please note that use of directly cooled stack could result in condensation of volatile material on the inside surface of the stack recuperator hence it is necessary to heat water in a semi-indirect design. In this design bundle of tubes are inserted in an annulus around the stack and at the same time air is passed through the annulus in the open space between water cooled tubes. This design avoids "undercooking" of the stack gases and still allows recovery of heat from exhaust gases. The design will also use hot air to "preheat" water in a separate heat exchanger so that the heat of preheated air can also be used efficiently.

Figure M-1 – 1 Schematic of a radiation recuperator with water cooled tubes

The following calculations (Figure M-1 – 2 and M-1 -3) assume a combined heat exchanger performance.

Parameter	Units	Furnace 9212 Single effect	furnace - 9211 Single effect
Percent O2 in furnace burners		100%	100%
Fuel use	MM Btu/hr	31.00	29.50
Fuel use	scfh	31,000	29,500
Oxygen or air -fuel ratio		2	2
Air or Oxygen use	scfh	62,000	59,000
Total flue gas volume	scfh	93,000	88,500
Temperature of gases	deg. F.	2,400	2,400
Available heat for flue gases	%	71.41	71.41
Possible heat use in furnace	MM Btu/hr	22.14	21.07
Heat in flue gases - calculated	MM Btu/hr	8.86	8.43
Assume heat losses from flue gases		10%	10%
Possible heat in flue gases	MM Btu/hr	7.98	7.59

Figure M-1 -2 Estimate of heat contained in furnace flue gases

The calculations are for a heat exchanger design where the gases are cooled from 2400 deg. F. and 65% of its heat is recovered to raise water temperature by 20 to 25 deg. F. The water can be used for absorption cooling system heat water. This is a combined efficiency of heat recovery from the stack recuperator as well. The assumption of 65% efficiency allows the gases to cool to a temperature range of 800 deg. F. to 875 deg. F. Final temperature depends on the hot gas temperature entering the recuperator and the above numbers are for initial gas temperature of 2400 deg. F. We were informed by the plant that the gases can be cooled to 800 deg. F. before any problems related to

condensation of the volatiles arise. Hence the assumption of 65% heat recovery after accounting for heat losses is reasonable and practical.

Parameter	Units	Furnace 9212 Single effect	furnace - 9211 Single effect
Hot water production			
Expected water heater efficiency	%	65	65
Heat going to water	MM Btu/hr	5.18	4.93
Temp. Diff for water	deg. F.	18	18
Hot water inlet temp to chiller system	deg. F.	200	200
Hot water outlet temp to chiller system	deg. F.	182	182
Mass flow of water	lbs/min	4,800	4,568
Hot water flow rate	gpm	575	547

Figure M-1 -3 Calculations for hot water available from a recuperator – water heater

Recommendation

It is recommended that the plant consider use hot water as energy source for absorption cooling system to produce chilled water. The plant uses a large amount of chilled water produced by using conventional compressor refrigeration system. As mentioned in an earlier section that describes details of the chilled water system, the plant uses three chillers for furnace 9212 and two chillers for furnace 9211. It is likely that one of these chillers would be replaced. Use of recirculating hot water produced by using furnace flue gas heat recovery system should be considered to replace or supplement current chiller system.

A schematic of the proposed system is shown in Figure M-1 – 4. The melter flue gases are cooled in a combined heat exchanger (radiation recuperator followed by (if necessary) an auxiliary heat exchanger. The following calculations are based on use of radiation recuperator only.

Figure M-1 – 4 Melter flue gas heat recovery system using absorption chiller

The absorption chiller uses absorption cycle in which refrigeration effect is produced through the use of two fluids and some quantity of heat input, rather than electrical input as in the commonly used vapor compression cycle systems. Absorption chillers work in primarily the same manner as conventional compressor based systems with the exception that the compressor is replaced by an absorber, a solution pump and a “generator” These systems use hot water, steam or direct fired burner as a source of heat as source of energy to replace compressor system. More details of the system is given at the end of this section.

Following calculations are based on a COP of 0.7 with 65% heat recovery efficiency (as efficiency of the absorption unit. This is very practical, doable and conservative. The system offers combined refrigeration capacity of about 531 tons. An electrical – compressor based system would be replaced by using this unit. It is assumed that the compressor system has a COP of 5. Based on this, electrical savings for the system would be 374 kWh per hour with potential savings of \$123,492 per year in electricity cost based on marginal electricity cost of 4.1 cents/kWh. .

Parameter	Units	Furnace 9212 Single effect	furnace - 9211 Single effect
Hot water production			
Expected water heater efficiency	%	65	65
Heat going to water	MM Btu/hr	5.18	4.93
Temp. Diff for water	deg. F.	18	18
Hot water inlet temp to chiller system	deg. F.	200	200
Hot water outlet temp to chiller system	deg. F.	182	182
Mass flow of water	lbs/min	4,800	4,568
Hot water flow rate	gpm	575	547
COP		0.7	0.7
Theoretical tons of refrigeration	tons	302	288
"Efficiency " of the unit		90%	90%
Available chiller capacity		272	259
Combined capacity of two furnaces	tons	531	
COP for the electrical - compressor unit		5	
Electrical energy required	kWh	374	
Cost of electricity	cents/kWh	4	
Operating hours/year	hrs/year	8,064	
Potential savings	\$/year	\$123,492	

Figure M-1 – 5 Calculations for savings for the absorption chiller heat recovery system

In summary, the savings would be about 374 kW x 8,064 hrs/year = 3,015,936 kWh per year for the two furnaces. CO₂ emission reductions would be about:

$$3,015,936 \text{ kWh/year} \times 2.3 \text{ lb CO}_2/\text{kWh} = 6,936,653 \text{ lb CO}_2/\text{year}$$

Estimated Implementation Cost

The cost of implementing this recommendation is based on cost data obtained from several sources. As shown in Figure M-1 – 6, basic absorption system cost for a unit of

530 TR capacity is about \$400/TR (ton-refrigeration) delivered to JM site in Ohio. This cost figures are obtained from Thermax, Inc. which is considered as a reputable international supplier of such units.

Figure M-1 – 6 Cost of the absorption chiller system

The heat exchanger – recuperator cost is estimated to be \$99,952 and \$70,795 respectively for furnaces 9212 and 9211. These costs are for standard units and they have been corrected for their special features used in this application. This cost is included in installation cost. All other costs relate to absorption system and its installation etc. are also included in installation cost line item. As shown in Figure M-1 – 7, overall cost for the heat recovery system and the absorption chiller is estimated at about \$1,076 per TR. This can be considered as reasonable cost.

Parameter	Units	Furnace 9212 Single effect	furnace - 9211 Single effect
Cost of the chiller system	\$/TR	400	
	\$/unit	212,465	
Cost associated with water heater etc.		\$ 99,952	\$ 70,795
Cost of installation			
Percentage of the unit cost		50%	
Installation cost for the system incl controls etc.		\$191,606	
Total cost - installed		\$574,819	
Cost of conventional units	\$/TR	300	
Cost of the chiller system		50%	
Installed cost of conventional system		\$239,024	
Cost difference (absorption - conventional)		\$335,795	
Payback period - new installation or replacement of older unit	Years	2.72	
Payback period - replacement of operating unit	Years	4.65	

Figure M-1 – 7 Cost estimate for the absorption chiller system heat recovery system

Estimated Simple Payback

The payback period is calculated by using estimated costs of savings and cost of the system. Installed cost for the system for two furnaces is \$574,819 and estimated savings in electricity cost based on 4.1 cents/kWh are \$123,492 per year.

We have used two different scenarios for estimating payback period. In one case it is assumed that the plant needs a new unit and use of absorption system would require premium over the cost of electrical – compressor based unit. The premium (difference between absorption unit cost and conventional unit cost) is used as basis for payback calculations. This payback period is 2.72 years. If the plant has to replace a well functioning unit and did not have to buy a new unit for replacement, then the cost is assumed as the total cost of the absorption cooling system as basis for payback calculations. This payback period is 4.65 years.

M-2: Increase Electric Boost in Melting

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy	17,844	-3,013,026	-4,914,000	\$7,400	None	Immediate

Analysis

The plant's melting furnaces use heat from both oxy-fuel firing and from electric resistance boost. According to plant personnel, electric boost consists of about 200 kW in each of five zones in both melting furnaces; for a total of 2,000 kW in melting furnace electric boost.

In AR FH1: Convert 9211 Forehearth to Oxy-fuel Burners, we estimated that oxy-fuel fired heat is about 67% efficient. Electric resistance heating, however, is 100% efficient. Since the plant's negotiated electrical energy rate is low, and since the price of natural gas is currently high and rising, it is economical to increase electric boost in the melting furnaces as much as possible and reduce natural gas use.

Recommendation

The quantity of heat added by the electric boost is limited by the lifetime of the electric heating elements. Based on this analysis, we recommend increasing electric boost in the plant's melting furnaces as much as possible.

Estimated Savings

If electric boost was increased by 20% in the melting furnaces, additional electrical energy use for the year would be about:

$$2,000 \text{ kW} \times 20\% \times 8,760 \text{ hours/year} = 3,504,000 \text{ kWh/year}$$

Because natural gas heat from oxy-fuel in the melting furnaces is about 67% efficient, the reduction in natural gas usage would be about:

$$[3,504,000 \text{ kWh/year} \times 3,412 \text{ Btu/kWh} \times (\text{mmBtu} / 10^6 \text{ Btu})] / 67\% = 17,844 \text{ mmBtu/year}$$
$$17,844 \text{ mmBtu/year} \times \$7.00 / \text{mmBtu} = \$124,908 / \text{year}$$

This would reduce CO₂ emissions by about:

$$17,844 \text{ mmBtu/year} \times 113 \text{ lb CO}_2 / \text{mmBtu} \approx 2,016,000 \text{ lb CO}_2 / \text{year}$$

The higher heating value of natural gas is 23,900 mmBtu/lb. For stoichiometric oxy-fuel combustion, four lb of oxygen is needed for one lb of natural gas. If the forehearth operated at 20% excess oxygen, the reduction of oxygen use would be about:

$$17,844 \text{ mmBtu/year} \times 10^6 \text{ Btu/mmBtu} \times (\text{lb} / 23,900 \text{ Btu}) \times 4 \text{ lb-O}_2 / \text{lb-NG} \times 120\%$$
$$= 3,583,732 \text{ lb/year}$$

At standard conditions, the density of oxygen is 0.0827 lb/ft³. Thus, the annual volume of oxygen production would be reduced by about:

$$3,583,732 \text{ lb/year} / 0.0827 \text{ lb/ft}^3 \times (\text{mcf} / 1,000 \text{ ft}^3) = 43,334 \text{ mcf/year}$$

According to the Melter Efficiency Analysis section of the report, about 11.33 kWh of electricity is needed to produce 1 mcf of oxygen. Thus, the annual savings in electrical energy consumption would be about:

$$43,334 \text{ mcf/year} \times 11.33 \text{ kWh/mcf} = 490,974 \text{ kWh/year}$$

The overall net increase in electrical energy use and cost would be about:

$$3,504,000 \text{ kWh/year} - 490,974 \text{ kWh/year} = 3,013,026 \text{ kWh/year}$$

$$3,013,026 \text{ kWh/year} \times \$0.039 / \text{kWh} = \$117,508 / \text{year}$$

This would increase CO₂ emissions by about:

$$3,013,026 \text{ kWh/year} \times 2.3 \text{ lb CO}_2/\text{kWh} \approx 6,930,000 \text{ lb CO}_2 / \text{year}$$

The net cost savings would be about:

$$\$124,908 / \text{year} - \$117,508 / \text{year} = \$7,400 / \text{year}$$

The net increase in CO₂ emissions would be about:

$$6,930,000 \text{ lb CO}_2 / \text{year} - 2,016,000 \text{ lb CO}_2 / \text{yr} \approx 4,914,000 \text{ lb CO}_2 / \text{year}$$

Estimated Implementation Cost

Increasing electric boost in the melting furnaces would require no significant implementation cost.

Estimated Simple Payback

Immediate

FH-1: Convert 9211 Forehearth to Oxy-fuel Burners

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy	114,730	-2,266,397	7,751,000	\$714,721	\$2,000,000	34 months

Analysis

The 9211 forehearth currently uses atmospheric air to mix with natural gas for combustion. The plant's melting furnaces were recently retrofitted with oxy-fuel burners, which use pure oxygen to mix with natural gas for combustion. Oxy-fuel is more energy-efficient and provides a higher flame temperature than burners that use atmospheric air because atmospheric air contains nitrogen which does not contribute to the combustion chemical reaction but only acts as a heat sink. Oxygen is produced in an oxygen plant and sent to the melting furnaces.

Within the next year, the 9211 forehearth will be rebuilt. According to management, it is possible to retrofit the 9211 forehearth with oxy-fuel burners during rebuild.

Recommendation

We recommend converting the 9211 forehearth to use oxy-fuel burners during rebuild and expanding the oxygen plant to provide for oxy-gas for the 9211 forehearth.

Estimated Savings

The temperature of exhaust from the 9211 forehearth is about 2,700 F. The spreadsheet below is an output screen from CombEff.xls for the combustion of atmospheric air with natural gas. CombEff.xls incorporates combustion chemical equations and heat energy balances to calculate combustion efficiency. According to the spreadsheet, the combustion efficiency in the forehearth is currently about 28.0%.

Input Data	
EA = excess air (0=stoch, 0.1 = optimum)	0.20
Tca = temperature combustion air before burner (F)	70
Tex = temperature exhaust gasses (F)	2,700
Constants (for Natural Gas)	
LHV = lower heating value (Btu/lb)	21,500
HHV = higher heating value (Btu/lb)	23,900
C _{pp} = specific heat of products of exhaust (Btu/lb-F)	0.260
Afs = air/fuel mass ratio at stoichiometric conditions	17.20
Calculated Values	
T _c = temp combustion (F) = $Tca + LHV / [(1 + (1 + EA)(Afs))C_{pp}]$	3,891
Efficiency = $[1 + (1 + EA)(Afs)] * C_{pp} * (Tc - Tex) / HHV$	28.0%

The graph below demonstrates the combustion efficiency improvement from using pure O₂ as the oxidizer in natural gas combustion. According to the graph, the forehearth combustion efficiency would increase to about 67%.

Source: North American Combustion Handbook 3rd ed. North American Manufacturing Company. 1995

According to management's records, the 9211 forehearth uses about 197,100 mmBtu per year. Thus, the annual natural gas savings would be about:

$$197,100 \text{ mmBtu/year} \times [1 - (28\% / 67\%)] = 114,730 \text{ mmBtu/year}$$

$$114,730 \text{ mmBtu/year} \times \$7.00 / \text{mmBtu} = \$803,110 / \text{year}$$

This would reduce CO₂ emissions by about:

$$114,730 \text{ mmBtu/year} \times 113 \text{ lb CO}_2 / \text{mmBtu} \approx 12,964,000 \text{ lb CO}_2 / \text{year}$$

To generate oxygen for oxy-fuel, additional electrical energy would be needed. The annual natural gas used in the forehearth would be about:

$$197,100 \text{ mmBtu/year} - 114,730 \text{ mmBtu/year} = 82,370 \text{ mmBtu/year}$$

The higher heating value of natural gas is about 23,900 mmBtu/lb. For stoichiometric combustion, four lb of oxygen is needed for one lb of natural gas. If the forehearth operated at 20% excess oxygen, the annual mass of oxygen used would be about:

$$82,370 \text{ mmBtu/year} \times 10^6 \text{ Btu/mmBtu} \times (\text{lb-NG} / 23,900 \text{ Btu}) \times 4 \text{ lb-O}_2 / \text{lb-NG} \times 120\% \\ = 16,542,929 \text{ lb/year}$$

At standard conditions, the density of oxygen is 0.0827 lb/ft³. Thus, the annual volume of oxygen used by the forehearth would be about:

$$16,542,929 \text{ lb/year} / 0.0827 \text{ lb/ft}^3 \times (\text{mcf} / 1,000 \text{ ft}^3) = 200,035 \text{ mcf/year}$$

According to the Melter Efficiency Analysis section of the report, about 11.33 kWh of electricity is needed to produce 1 mcf of oxygen. Thus, the annual increase in electrical energy consumption would be about:

$$200,035 \text{ mcf/year} \times 11.33 \text{ kWh/mcf} = 2,266,397 \text{ kWh/year} \\ 2,266,397 \text{ kWh/year} \times \$0.039 / \text{kWh} = \$88,389 / \text{year}$$

This would increase CO₂ emissions by about:

$$2,266,397 \text{ kWh/year} \times 2.3 \text{ lb CO}_2 / \text{kWh} \approx 5,213,000 \text{ lb CO}_2 / \text{year}$$

The net annual savings would be about:

$$\$803,110 / \text{year} - \$88,389 / \text{year} = \$714,721 / \text{year}$$

The net reduction in CO₂ emissions would be about:

$$12,964,000 \text{ lb CO}_2 / \text{year} - 5,213,000 \text{ lb CO}_2 / \text{year} \approx 7,751,000 \text{ lb CO}_2 / \text{year}$$

Estimated Implementation Cost

According to management, it would cost about \$2,000,000 to convert the 9211 forehearth to use oxy-fuel and expand the oxygen plant to provide enough oxy-gas.

Estimated Simple Payback

$$(\$2,000,000 / \$714,721 / \text{year}) \times 12 \text{ months/year} = 34 \text{ months}$$

FH-2: Reposition Mullite Zone Dividers in Forehearths

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy	15,900		1,797,000	\$111,000	Negligible	Immediate

Analysis

The forehearths are divided into a series of zones. Each zone is separated by an insulated mullite board which extends from the ceiling of the flow passage to just a few inches above the molten glass. Each zone has about 48 burners and two exhaust ports. The exhaust ports are located across from each other on the top surface of the forehearths. A thermostat located in each zone controls the firing rate of the burners. This design enables fairly precise control of the temperature of the glass as it flows through the forehearths.

The position of the mullite boards could not be exactly determined from the available construction blueprints. However, operators thought that the boards may be located at the midpoint between each pair of exhaust ports as shown below. For simplicity, the diagram below shows 11 burners per zone, even though there may be up to 48. The distance between each burner is about 1 foot. The arrow shows the direction of glass flow. If the mullite boards are positioned as shown below, then the total horizontal travel distance of the combustion gasses from burner to exhaust port in each zone is:

$$2 \times (5 + 4 + 3 + 2 + 1) = 30 \text{ feet}$$

If the mullite boards were repositioned as shown in the figure below, then the total travel horizontal travel distance between each burner and the exhaust port would be increased to about:

$$(10 + 9 + 8 + 7 + 6 + 5 + 4 + 3 + 2 + 1) = 55 \text{ feet}$$

This increased travel distance would increase the heat transfer between the hot combustion gasses and the molten glass. As a result, the combustion gasses would leave the exhaust ports at a lower temperature and the overall efficiency of the heating process would improve.

Recommendation

During the next forehearth rebuild, we recommend repositioning the mullite boards that divide forehearth zones to maximize the total travel distance of the hot exhaust gasses, and move the gasses in a direction counterflow to the flow of the molten glass.

Estimated Savings

During 2003, the 9211 and 9212 forehearths operated continuously and used about 141,000 and 98,000 mcf of gas respectively. The actual temperature of exhaust gasses leaving the forehearth vents is about 2,700 F. Assuming 20% excess air, the combustion efficiency, as calculated by HeatSim (Kissock and Carpenter, 2004) is about 28%.

The increase in heat transfer and efficiency from repositioning the mullite zone dividers is difficult to quantify because the proportion of heating from radiation and convection is unknown. However, as seen above, the total contact distance between the hot gasses and the glass will increase significantly and the heat transfer should also increase significantly. If the combustion efficiency increases by only 2%, from 28% to 30%, the savings would be about:

$$(141,000 \text{ mmBtu/yr} + 98,000 \text{ mmBtu/yr}) \times (1 - 0.28 / 0.30) = 15,900 \text{ mmBtu/yr}$$
$$15,900 \text{ mmBtu/year} \times \$7.00 / \text{mmBtu} = \$111,000 / \text{year}$$

This would reduce CO₂ emissions by about:

$$15,900 \text{ mmBtu/year} \times 113 \text{ lb CO}_2 / \text{mmBtu} \approx 1,797,000 \text{ lb CO}_2 / \text{year}$$

Estimated Implementation Cost

We assume that during a rebuild, the cost of repositioning the mullite zone dividers would be negligible.

Estimated Simple Payback

Immediate

FH-3: Install Electric Resistance Heating in 9211 Forehearth During Rebuild

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy	197,100	-4,108,440	12,823,000	1,219,471	\$1,000,000	10 months

Analysis

The 9211 forehearth employs natural gas combusted with atmospheric air for heating. Within the next year, the 9211 forehearth will be rebuilt. In AR FH1: Convert 9211 Forehearth to Oxy-fuel Burners, we calculated natural gas heat in glass melting operation using atmospheric air as the oxidizer to be about 28% efficient. Electric resistance heating, however, is 100% efficient. Since the plant's negotiated electrical energy rate is low, and since natural gas price is currently high and rising, it is economical to install electric resistance in the 9211 forehearth during rebuild.

In addition, the forehearth can be rebuilt shorter and narrower than its current dimensions if it were to use electric resistance instead of natural gas heat. The reason for this is because the space directly above molten glass where combustion products are currently present would no longer be needed. According to "High-Temperature-Resistance Heating Elements in the Glass Industry" (Frisk and Linder, American Ceramic Society Bulletin, November 2001), the surface area of a forehearth could be reduced by as much as 55% if rebuilt with electric resistance.

Recommendation

We recommend considering installing electric resistance in the 9211 forehearth during rebuild to replace natural gas firing.

Estimated Savings

According to plant records, the average natural gas input to the 9211 forehearth is about 22.5 mmBtu/hour, or 197,100 mmBtu/year. If so, the energy cost to operate the forehearth is about:

$$197,100 \text{ mmBtu/year} \times \$7.00 / \text{mmBtu} = \$1,379,700 / \text{year}$$

Eliminating natural gas use in the forehearth would reduce CO₂ emissions by about:

$$197,100 \text{ mmBtu/year} \times 113 \text{ lb CO}_2 / \text{mmBtu} \approx 22,272,000 \text{ lb CO}_2 / \text{year}$$

Because the combustion efficiency of the forehearth is about 28%, the energy lost from exhaust gasses is about:

$$22.5 \text{ mmBtu/hour} \times (100\% - 28\%) = 16.2 \text{ mmBtu/hour}$$

In addition to this loss, energy is also lost via radiation through the exhaust ports. If the forehearth was heated by electric resistance, there would be no need for exhaust ports.

The forehearth has eight stages, and each stage has four exhaust ports; for a total of 32 exhaust ports. Each exhaust port is about 1 ½ feet long and about 4 inches wide. Thus, the combined area of all exhaust ports is about:

$$1 \frac{1}{2} \text{ ft} \times \{4 \text{ in} \times (1 \text{ ft} / 12 \text{ in})\} \times 32 \text{ ports} = 16 \text{ ft}^2$$

The temperature of molten glass in the forehearth is about 2,700 F. We measured ambient temperature in the forehearth area to be about 150 F. Assuming the molten glass radiates as a blackbody, from the Stephen-Boltzmann Law, radiation energy lost through the exhaust ports is about:

$$(1.714 \times 10^{-9} \text{ Btu/hr-ft}^2\text{-R}^4) \times 16 \text{ ft}^2 \times [(2,700 + 460 \text{ R})^4 - (150 + 460 \text{ R})^4] \\ \times (\text{mmBtu} / 10^6 \text{ Btu}) = 2.7 \text{ mmBtu/hr}$$

The remaining energy is about:

$$22.5 \text{ mmBtu/hour} - 16.2 \text{ mmBtu/hour} - 2.7 \text{ mmBtu/hour} = 3.6 \text{ mmBtu/hour}$$

Because the forehearth does not heat molten glass, but keeps it at temperature, all of this remaining energy ends up being transferred through the forehearth wall via conduction, convection, and radiation. To verify this result, we used the heat loss simulation program HeatSim (Kissock, 2001) which can be downloaded free of charge off of the UDIAC website www.udayton.edu/udiac. HeatSim applies convection, conduction, and radiation heat transfer relations to determine heat lost from hot surfaces. The plan's forehearths we estimate are, on average, about four feet high and about six feet wide. We estimate the overall length to be about 300 feet. We measured the surface temperature of its outer walls to be about 400 F. The HeatSim output screen simulating heat lost through the forehearth wall is shown below. According to HeatSim, about 4.3 mmBtu is lost through the wall per hour. This is slightly higher than the 3.6 mmBtu/hour calculated above. However, we conservatively assume 3.6 mmBtu/hour.

HeatSim Output Screen for Heat Transfer Through Forehearth Wall

If the forehearth surface area were reduced by 55%, the reduction in heat lost through the wall would be about:

$$3.6 \text{ mmBtu/hour} \times 55\% = 2.0 \text{ mmBtu/hour}$$

The electric power needed in the forehearth would be about:

$$(3.6 \text{ mmBtu/hour} - 2.0 \text{ mmBtu/hour}) \times 293 \text{ kWh/mmBtu} = 469 \text{ kW}$$

The annual electrical energy used by the forehearth would be about:

$$469 \text{ kW} \times 8,760 \text{ hours/year} = 4,108,440 \text{ kWh/year}$$

$$4,108,440 \text{ kWh/year} \times \$0.039 / \text{kWh} = \$160,229 / \text{year}$$

This would increase CO₂ emissions by about:

$$4,108,440 \text{ kWh/year} \times 2.3 \text{ lb CO}_2 / \text{kWh} \approx 9,449,000 \text{ lb CO}_2 / \text{year}$$

Annual cost savings would be about:

$$\$1,379,700 / \text{year} - \$160,229 / \text{year} = \$1,219,471 / \text{year}$$

The net increase in CO₂ emissions would be about:

$$22,272,000 \text{ lb CO}_2 / \text{year} - 9,449,000 \text{ lb CO}_2 / \text{year} \approx 12,823,000 \text{ lb CO}_2 / \text{year}$$

Estimated Implementation Cost

Management estimates that the additional cost during rebuild to install electric resistance equipment in the 9211 forehearth would be about \$1,000,000.

Estimated Simple Payback

$(\$1,000,000 / \$1,219,471 \text{ /year}) \times 12 \text{ months/year} = 10 \text{ months}$

FH-4: Insulate Forehearths

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy	135,000	0	15,255,000	\$945,000	\$112,000	1 month

Analysis

The forehearths transport the molten glass from the melters for about 350 feet to the bushings. The first 30 feet or so of the forehearths is called the conditioning channel, where no heat is added and the molten glass is allowed to degas. Gas burners are located along the remaining length of the forehearths, and are fired to keep the molten glass hot. The function of the forehearths was described by an operator to be “controlled cooling”; the glass leaves the melters at about 2,400 F and is delivered to the bushings at about 2,200 F.

In 2003, the 9211 and 9212 melters used about 256,000 mcf and 237,000 mcf of gas respectively. The 9211 and 9212 forehearths used about 141,000 and 98,000 mcf of gas respectively, which is about half as much gas as the melters. This is an astonishing number, since the forehearth gas provides no useful purpose other than making up for heat loss through the walls and openings of the forehearths.

This suggests that adding additional insulation to the forehearths is a major cost saving opportunity. The opportunity is especially significant since a one Btu reduction in heat loss will generate at least four Btus of gas savings, due to the low combustion efficiency of the gas heaters.

Recommendation

We recommend adding insulation to the forehearths. Any type of increased insulation, including fiberboard, Kaowool insulating fiber or additional layers of refractory brick would help. We measured the surface temperature of the forehearths to be between 200 F and 800 F. The insulation will be most effective if placed over the hottest surfaces.

Estimated Savings

During 2003, the 9211 and 9212 forehearths operated continuously and used about 141,000 and 98,000 mcf of gas respectively. The average hourly gas use rates were about:

9211: 141,000 mmBtu/yr / 8760 hr/yr = 16.0 mmBtu/hr

9212: 98,000 mmBtu/yr / 8,760 hr/yr = 11.2 mmBtu/hr

Even if an infinite counter-flow heat exchanger, the temperature of the exhaust gasses leaving the forehearth could only approach 2,400 F. The actual temperature of exhaust gasses leaving the forehearth vents is about 2,700 F. Assuming 20% excess air, the combustion efficiency, as calculated by HeatSim (Kissock and Carpenter, 2004) is about 28%.

Thus, the average rate of heat loss through the walls of the forehearths is about:

9211: $16.0 \text{ mmBtu/hr} \times 28\% = 4.48 \text{ mmBtu/hr}$

9212: $11.2 \text{ mmBtu/hr} \times 28\% = 3.14 \text{ mmBtu/hr}$

According to construction blueprints, the width, height and length of the 9211 forehearth are about 6 feet by 6 feet by 390 feet, and the width, height and length of the 9212 forehearth are about 4 feet by 4 feet by 340 feet. The average temperature of the glass inside the forehearths is about 2,300 F. We measured surface temperatures of the forehearths to vary between 200 F and 800 F. The average surface temperature of each forehearth can be determined by varying the average surface temperature in HeatSim, which calculates heat loss from hot surfaces, until the heat loss equals the actual hourly heat loss. The HeatSim results, shown below, indicate that the average surface temperatures of the 9211 and 9212 forehearths are about 301 F and 333 F respectively.

Any type of additional insulation on the forehearths would result in significant savings. As an example, we will use 1-inch Kaowool ceramic fiber blanket, which has an R-value of 2.1 ft²-hr-F/Btu at 500 F and costs about \$2 per square foot. If two inches on this insulation were installed over the forehearths, the reduction in heat loss through the walls of the 9211 and 9212 forehearths would be about 2.44 mmBtu/hr and 1.87 mmBtu/hr respectively.

HeatSim results for 9211 forehearth.

HeatSim results for 9212 forehearth.

The gas savings would be about:

$$(2.44 \text{ mmBtu/hr} + 1.87 \text{ mmBtu/hr}) \times 8,760 \text{ hrs/yr} / 0.28 = 135,000 \text{ mmBtu/yr}$$

$$135,000 \text{ mmBtu/year} \times \$7.00 / \text{mmBtu} = \$945,000 / \text{year}$$

This would reduce CO₂ emissions by about:

$$135,000 \text{ mmBtu/year} \times 113 \text{ lb CO}_2 / \text{mmBtu} \approx 15,255,000 \text{ lb CO}_2 / \text{year}$$

Estimated Implementation Cost

One inch Kaowool insulation costs about \$2 per square foot. Assuming two inches of insulation would cost about \$10 per square foot installed, the total implementation cost, from HeatSim, would be about:

$$\$70,920 + \$41,120 = \$112,000$$

Estimated Simple Payback

$$(\$112,000 / \$945,000 / \text{year}) \times 12 \text{ months/year} = 1 \text{ month}$$

MO-1: Use Mat Oven Incinerator Exhaust Air

	Annual Savings			Project Cost	Simple Payback
	Resource	CO ₂ (lb)	Dollars		
Natural gas	16,716 MMBtu	1,889,000	\$117,000	\$182,000	18 months

Recommendation by E3M

Analysis

The plant uses a thermal oxidizer (TO) or an incinerator for destruction of volatile materials contained in the mat oven exhaust air. The TO uses regenerative heat recovery system to preheat incoming exhaust air by using heat from the TO exhaust gases. This is a very efficient system however the exhaust gases from the TO are still at about 350 deg. F. and contain substantial amount of heat. Based on data for natural gas use for the TO, as discussed in a previous section, the expected average air flow is about 765,476 scfh. This is based on natural gas use and % O₂ measured in the TO exhaust gases. These gases are assumed to be at average temperature of 350 deg. F.

Recommendation

We recommend that the plant consider use of exhaust air heat for heating water as illustrated in Figure MO-1 - 1. The hot water can be used for plant or can be supplied to a water tank that supplies water to absorption cooling system.

The following calculations are based on assumption that the flue gas heat will be used for absorption cooling application. In this case a water to gas heat exchanger will be used to heat water. Water enters the heat exchanger at 182 deg. F. and leaves at 202 deg. F., a rise of 20 deg. F.

Figure MO-1– 1 Heat recovery system for incinerator flue gases

We suggest that the system be designed to drop gas temperature to 220 deg. F. This will avoid condensation of water vapor from the TO exhaust air. Note that the TO exhaust air contains a large amount of excess air and its dew point will be much lower than the dew point of stoichiometric products of natural gas combustion. As shown in Figure MO-1 – 2 , this will result in an LMTD of 80.90 deg. F.

Figure MO-1 – 2 LMTD calculations for incinerator flue gas – water heat exchanger

A simple tube and shell type heat exchanger is considered for these calculations. The water side heat transfer coefficient is very high (in excess of 50 to 100 Btu/hr-ft² F.) and in most designs using finned tubes it is possible to get over all heat transfer coefficient of 10 Btu/(hr.ft² F). Heat transfer calculations in Figure MO-1 – 3 show that it would be possible to get heat transfer rate of 1.99 MM Btu/hour. With this heat transfer rate it is possible to heat 105 GPM water by 20 deg. F. (from 182 deg. F. to 202 deg. F.) as required for operation of an absorption cooling system. .

Incinerator gas inlet temp	350	Deg. F.
Water inlet temp	182	Deg. F.
Water outlet temp	202	Deg. F.
Expected gas outlet temp	220	Deg. F.
LMTD	80.90	Deg. F.
Exhaust air (flue Gas) flow	765,476	scfh
Heat recovered	1,990,238	Btu/hr
Water heated	105	gpm

Figure MO-1 – 3 Heat transfer calculations for gas-water heat exchanger

The calculations also show that the heat exchanger would require “bare” tube surface area of 2,460 ft². The surface area can be enhanced by using finned tubes. A simple shell and tube heat exchanger using carbon steel shell and water tubing without fins would cost approximately \$72,900. This cost is obtained from a commonly used cost estimator tool that give costs in terms of year 2003. We have applied a price escalation factor of 5% to estimate cost for the year 2007. The heat exchanger installation and piping cost is estimated to be 50% of the cost of simple heat exchanger. Premium for finned tube heat exchanger and an ID fan is estimated to be at 50% of the “bare tube” heat exchanger. Thus the total cost for the installed system is \$182,326. Based on estimated savings for recovered heat the payback period is 1.56 years. Details are given in Figure MO-1 – 4.

Overall heat transf. Coeff.	10	Btu/(hr-ft ² F)
Area	2,460	ft ²
Heat exchanger cost	60,000	2003 cost
Heat exchanger cost	72,930	2007 cost
Instalation etc.	72,930	100%
Premium for finned tubes	50%	apply to HX cost
Total cost - installed	\$ 182,326	
Value of recoverd heat		
Gas cost	\$ 7.00	per MM Btu
Heat recovered	1.99	MM Btu/hr
No. of hours	8,400	Hrs/year
Value of recoverd heat	\$ 117,026	
Payback	1.56	Years

Figure MO-1 – 4 Payback calculations for water heater system

Savings

In actual application the payback period would be different if we would use the recovered heat to replace electrical energy used for running compressor based water chiller system. It is assumed that the heat or hot water will be used in the main absorption cooling system that uses furnace exhaust gases in absorption cooling system. For this calculation it is assumed that the single effect of absorption cooling system has a Coefficient of Performance (COP) of 0.7, the unit operating hours of 8400 hours (24 hours/day, 7 days/week and 50 weeks/year) and COP of compressor based system as 5.0. As shown in Figure MO-1 – 5, at an electric rate of 4.1 cents/kWh, estimated savings is \$28,116 per year. Use of the hot water heat for refrigeration also requires additional cooler capacity and its cost is assumed to be \$600 per ton installed. Payback period based on cost of heat exchanger as well as the added cost of absorption cooling system is 7.67 years.

Approximate COP	0.7	
Potential addition to refrigeration	116	Tons refrigeration (TR)
COP for conventional cooling system	5	
Potential electricity savings	81.64	kWh
No. of hours per year (24/7/50)	8400	hrs./year
Total electric savings	685,766	kWh/year
Avoided (marginal) elect. Cost	4.1	Cents/kWh
Potential cost savings	\$ 28,116	per year
Added installed cost of absorption system		
Unit cost per TR of refrigerant	\$ 600	per TR
Added cost for cooling system	\$ 69,658	
Payback based on electric savings	7.67	Years

Figure MO-1 – 5 Payback calculations based on electric energy replacement

Reduction in CO₂ emissions, based on energy savings is about:

$$16,716 \text{ MM Btu/year} \times 113 \text{ lb CO}_2/\text{MM Btu} \approx 1,889,000 \text{ lb CO}_2/\text{year}$$

Implementation Cost

As shown in Figure MO-1 – 4, cost of implementing this recommendation depends on two possible uses of the recovered heat. In one case hot water is used in the plant and in other case it is used for added refrigeration capacity. Cost of implementation for the first case is \$182,386 and in other case it is additional \$69,658 or total cost of \$252,044.

Savings for each of these two cases are: \$117,026 and \$145,142 respectively.

Estimated Simple Payback

Estimated savings are calculated by using two different methods. The savings based on equivalent gas cost for the recovered heat is estimated to be \$117,000 and based on potential electric savings is \$28,116. Pay back periods based on these two cost savings are 1.6 years and 7.67 years respectively. For summary we have used payback period for heating water and its use in the plant.

MO-2: Use Outdoor Air As Combustion And Make-Up Air In Mat Oven

	Annual Savings			Project Cost	Simple Payback
	Resource	CO ₂ (lb)	Dollars		
Natural Gas	6,662 MMBtu/year	753,000	\$46,600	\$32,300	8 months

Analysis

The plant uses combustion air for burners fired in fore hearths of two melting furnaces (9211 and 9212), mat drying oven and several other small batch ovens. At the same time a large volume of make up is used for the mat oven and other ovens. All of this air is drawn from the building housing the furnaces and ovens. The plant location requires that the plant building be closed during cold weather experienced from approximately November to April of a year. Use of plant air as combustion air and make-up air creates negative pressure inside the plant building that results in infiltration of cold outdoor air in the building from any opening such as doors in the building. This results in number of problems such as cold spots in the building that require use of gas fired make up air heaters and/or perhaps localized electric heaters.

Amount of air required for burners in fore hearths (furnaces 9211 and 9211) and mat oven plus make up air for the amt oven is calculated by using available data. Air requirement for the batch ovens used in the plant is not considered. The melting furnaces use oxy-fuel burners and hence do not require combustion air but it is likely that a fair amount of plant air infiltrates in the furnaces due to negative pressure in the furnace. This is ignored also.

The data used for calculating plant air requirement is includes:

- Exhaust air from mat oven going to thermal oxidizer
- Burner operating data for the mat oven and energy use in mat oven
- Energy use in fore hearths for two furnaces

This data is analyzed in earlier sections and is summarized in Figure MO-2 – 1.

Air used for mat oven burners and makeup air		
Average firing rate based on heat balance and measurements	12.1	MM Btu/hr
Combustion air at 20% excess air	145,200	scfh
Estimated total exhaust air volume from ovens	765,476	scfh
Estimated make up air for the mat oven	620,276	scfh

Source	MM Btu/year used for calculations
9211 fore hearth use (MMBtu per year).	197,100
9212 fore hearth use (MMBtu per year)..	85,085
Total n. gas used in the fore hearths (MMBtu per year).	282,185
Combustion air for forehearth with 10% excess air SCFH	369,528

Figure MO-2 – 1 Summary of air used by heating equipment from the plant

Combustion air requirement for the fore hearth burners is based on use of 10% excess air while the combustion air for mat oven burners is estimated based on 20% excess air for burners due to lower temperature operation of the oven burners.

Recommendation

We recommend that combustion air for the fore hearth burners and mat oven burners as well as make-up air for the mat oven be taken directly from outside the building to reduce cold air draft in the building as well as to reduce particulates mixing in the air intake to the air supply blowers. It is suggested that two ducts be used to make up and combustion air for the mat oven while two separate ducts be used to supply combustion air for each of the fore hearth combustion air supply.

Typical and possible arrangement for air distribution to mat oven is shown in Figure MO-2 – 2. Similar system can be used for the fore hearth air supply also.

Figure MO-2 – 2 Schematic of out door air distribution system for the mat oven

Note that the system would use suction pressure of the combustion air blower and zone recirculating fan to draw air from the outdoor. An alternate scheme is to use one or more low pressure air fans to bring air to the blower and recirculating fans. We felt that it is not necessary to use external fan for ambient air. However if the plant engineering personnel feels that it is necessary it would be necessary to perform additional calculations and cost analysis.

Savings calculations are carried out using weather data for Toledo, OH obtained from U. S. National Weather Bureau. The data is average values of 30 years, from the years 1971 to 2000. The data is shown in Figure MO-2 - 3.

Figure MO-2 – 3 Monthly average temperature for Toledo, OH

Calculations are carried out to estimate possible savings resulting from use of out door air. The assumptions are listed below.

- The make up air heaters heat air to 100 deg. F. Note that in most cases the air heaters discharge air at a higher temperature than the room temperature
- The make up air heating is used when average temperature is below 50 deg. F. This means for the plant it is used between the months of November to April.
- Average temperature is used for calculating hourly gas sue for the make up air heater.
- Make up air heater efficiency is 85% which is normal.
- Volume of air heated is equal to the volume of air used in mat oven and fore hearths. It is likely that reducing or eliminating negative pressure in the building would result in reduction of larger volume of air
- Cost savings associated with other possible advantages are not considered.

Calculations shown in Figure MO-2 – 4 indicate annual gas cost savings of \$46,600 at a gas cost of \$7 per MM Btu.

Month	Average Temp (F)	days/month	Temp. rise (F) required*	Heat required MM Btu/hr.**	Heat used MM Btu/month	Cost savings ***
Jan	23.90	31.00	76.10	1.88	1,396	\$ 9,770
Feb.	27.00	28.00	73.00	1.80	1,209	\$ 8,465
Mar	37.20	31.00	62.80	1.55	1,152	\$ 8,062
Apr	48.30	30.00	51.70	1.27	918	\$ 6,423
May	59.60					
Jun	68.80					
Jul	73.00					
Aug	70.80					
Sep	63.50					
Oct	51.80					
Nov	40.50	30.00	59.50	1.47	1,056	\$ 7,392
Dec	29.20	31.00	50.80	1.25	932	\$ 6,522
Estimated total annual savings						\$ 46,634

Notes:

- * Based on heating air to 100 deg. F. in the make-up air heater
- ** Based on air heater operating at 85% efficiency
- *** Based on gas cost of \$ 7.00 per MM Btu for natural gas

Figure MO-2 – 4 Calculations for energy savings by using outdoor air

This would reduce CO₂ emissions by about:

$$6,662 \text{ mmBtu/year} \times 113 \text{ lb CO}_2/\text{mmBtu} \approx 753,000 \text{ lb CO}_2/\text{year}$$

Implementation Cost

Cost of implanting this recommendation includes cost of duct, dampers, support and installation. It is assumed that total six air intakes and ducts would be required. Two large ducts (4 ft. diameter) and eight smaller ducts (2 ft. diameter) would be used to supply make up air to mat oven, eight smaller (2 ft. diameter) ducts would be used to supply combustion air to the oven combustion air blowers and two ducts of 2 ft. diameter each would used to supply combustion air to the air blowers for fore hearths.

Estimated duct lengths and cost are given in Figure MO-2 – 5. Cost of duct and installation is assumed to be \$40 per ft. for 2 ft. diameter. Duct and \$75 per ft. for 4 ft. diameter duct.

Work to install ducts for using outside air	No of ducts	Approx. size (ft. dia)	Approx. length-ft	Approx. cost
Install outside air ducts for combustion air fans ¹	8	2	40	\$ 12,800
Install outside air ducts for zone makeup air ²	8	2 and 4	see note 2	\$ 15,500
Install outside air ducts for fore hearth combustion fans ³	2	2	50	\$ 4,000
Total cost				\$ 32,300
Note: 1 Eight ducts of 2 ft. diameter and 40 ft. length each 2 Main ducts two of 4 ft. dia. - 50 ft. long, eight ducts of 2 ft. diameter, 25 ft. long one for each zone recirculation fan 3 Two ducts of 2 ft. diameter and 50 ft. length each Cost of duct (installed with damper etc.) 2 ft. diameter - \$40 per ft. 4 ft. diameter - \$75 per ft.				

Figure PH 3- 5 Cost analysis for out door air distribution system

Total estimated cost for implementation of this recommendation is estimated to be \$32,300. Payback period based on savings and cost of implantation as discussed above, would be about:

$\$32,300 / \$46,600 \text{ /yr} = 0.7 \text{ years or 8 months.}$

MO-3: Reduce Air Entering Mat Oven

	Annual Savings			Project Cost	Simple Payback
	Resource	CO ₂ (lb)	Dollars		
Natural Gas	6,158 MMBtu/year	696,000	\$43,100	\$20,000	6 months

Analysis

The mat oven is used for drying mat of fiberglass impregnated in binder that may contain organic materials and, in some cases, moisture. The mat in the wet form is laid on a belt that carries the material through the oven where moisture and part of the binder is removed. The oven is divided in 8 zones each with one burner and two air recirculating fans to provide convection heat to the mat. A large amount of make-up air is introduced in the oven zones to maintain required level of moisture and solvent concentration in the recirculating air. The make up air and combustion products from the burners are collected at each end of the oven and taken to a regenerative thermal oxidizer (RTO) where the air temperature is raised to about 1400 deg. F. to assure destruction of organic materials before the air is discharged into the atmosphere.

The oven zone temperature varies from 250 deg. F. to 400 deg. F. Measurement of temperature, oxygen and CO concentration in the oven zones were carried out during the assessment. Figure MO-3 – 1 gives values at the time of measurements.

Zone	1	2	3	4	5	6	7	8
Temperature (F)	246	271	329	369	385	394	396	397
O ₂ %	18.9	17.7	16.5	15.2	13.2	12.5	12.3	12.8
CO ppm	12	21	40	87	165	136	213	242

Figure MO-3 – 1 Measurements of temperature and gas analysis for mat oven zones

This measurements show that the front zones are getting relatively large amount of air and at the same time very little solvent vapors. It may also suggest that the air entry into the oven is properly balanced. Normally the end zone should have relatively low amount of CO or combustibles since all volatile material or binder should have come out of the mat before it exists the oven.

We estimated amount of exhaust air based on energy use and measurement of oxygen in oven exhaust gases. The calculations are discussed a previous section. These calculations show that average exhaust air volume is 765,476 scfh. This volume includes combustion products gases from the burners as well as make up air. The combustion products volume is calculated by performing a detail heat balance for the oven. The heat balance shows that total average heat input in the oven should be about 12.1 MM Btu/hr. This number is very close to the once estimated by the pant and discussed earlier in this report 106,600 MCF per year or 12.7 MM Btu/hr. Based on this heat input the combustion products volume is 145,200 scfh. This is estimated on the assumption of 20% excess air for combustion and 10,000 scf air per MM Btu/hr firing rate.

Total combustion air = $12.1 \times 10,000 \times 1.2 = 145,200$ scfh

Make up air volume = 765,476 – 145,200 = 620,276 scfh.

Oven heat load calculations		
Wt. Of the mat (at entry)	180	lbs./min
% moisture	20%	
Approx. moisture	36	lbs./min
Heat for belt (assumed)	240	lbs./min
Avg. exhaust temp	400	Deg. F.
Heat in water vapor	1240	Btu/lb.
Total heat for moisture	2,678,400	Btu/hr
Heat required for mat	734,400	Btu/hr
Heat required for belt	1,224,000	Btu/hr
Total heat load excl. air	4,636,800	
Estimated make-up air flow	600,236	scfh
Heat for the make-up air	4,081,606	Btu/hr
Total heat use estimated	8,718,406	
Wall loss calculations		
Oven width (outside)	15	ft
Oven length (outside)	184	ft
Oven height	10	ft
Total area - outside	6740	ft^2
Approx. "skin" temperatur	130	deg. F.
Heat loss/ft^2 - estimate	119	Btu/hr-ft^2
Total wall heat loss	802,060	Btu/hr

Heat balance		
Total heat for moisture	2,678,400	Btu/hr
Heat required for mat	734,400	Btu/hr
Heat required for belt	1,224,000	Btu/hr
Heat for the make-up air	4,081,606	Btu/hr
Total wall heat loss	802,060	Btu/hr
Total estimated heat load	9,520,466	Btu/hr
Misc. losses @5% of total	476,023	
Av. Heat for the burners	82%	
Gross heat required	12,086,348	Btu/hr

Figure MO-3 – 2 Heat balance for the oven

A preliminary heat balance was carried out using data given in Figure MO-3 – 1 above. The heat balance shown in Figure MO-3 - 2 and Figure MO-3 - 3 indicates that more than 70% of the total heat input into the oven is used for heating make-up air and moisture removal. These two areas should be considered as the main targets for reducing energy use on the mat oven.

Figure MO-3 – 3 Distribution of heat in mat oven

Recommendation

The general recommendation is to reduce amount of uncontrolled volume of air entering the oven by taking following actions or steps.

- Close the unnecessary and frequently used manually adjustable openings by using various methods such as using flexible seals or keeping the doors closed. The openings include areas where the belt is returned and other surrounding areas.
- Reduce and control suction (negative pressure) in the first zone at the entry and exit end of the oven. Possible methods include: use better (smaller?) damper that can be closed and opened to adjust negative pressure in the zones; use a smaller exhaust fan and/or install a variable speed motor and use it to control negative pressure. This would eliminate or reduce “fluttering” of the web or mat at it enters or exists the oven.

During the visit we observed that there is a large opening at the entrance and exit end of the oven where the belt is returned. The measurements indicate the opening is 6” high and 13 ft. wide on the top and bottom of the belt at the entrance and exit. In addition to this the door on the left side of the oven (looking from the entrance side) was also left open to allow air into the zone 1 to avoid excessive air entering the oven at the level where mat enters the oven. The oven pressure was clearly negative since air was entering the oven. Although we could not measure exact value of the oven negative pressure, it seems that it was in the range of 0.05 to 0.1 inch w.c. We calculated approximate volume of air that could enter the oven through the oven openings and cost of heating this air or potential savings by closing these openings. The results for 0.1 inch negative pressure and two openings with gap of 6” above and below the belt at the oven entrance and exit end and 12.5 ft. width (total area of 300 sq. inch) is shown in Figures MO-3 - 4 below.

Effect of 0.05 inch negative pressure with the same opening size is shown in Figure MO-3 - 5.

Parameter	Scenario	
	Baseline	Efficiency Measure
Equipment Load and Annual Use Inputs		
1. Connected load (MBtuh)	45,900	
2. Operating time (hrs/yr)	8,400	
3. Load Factor	26%	
4. Equivalent full load hours (hrs/yr)	2,218	
5. Annual Gas Use (therms/yr)	1,017,878	
Temperature and % Oxygen Inputs		
6. Flue gas temperature (F)	400	400
7. Oxygen (O ₂) in flue gas (% dry)	3.5%	3.5%
8. Combustion air temperature (F)	80	80
Gas Savings from Reducing Air Infiltration on Induced Draft Combustion		
15. Induced draft (inches of W.C. of vacuum)	0.050	
16. Total opening area (square inches)	300	
17. Air infiltration (scfh)	89,917	
18. Annual gas use (therms/yr)	1,017,878	
19. Gas savings (%)	6.1%	
20. Gas savings (therms/year)	61,585	
Cost Savings from Reducing Air Infiltration on Induced Draft Systems		
21. Gas rate (\$/therm)	\$0.700	
22. Annual cost savings (\$/year)	\$43,109	

Figure MO-3 – 5 Potential savings for 0.05 inch negative pressure in the oven

Effect of 0.1 inch negative pressure with the same opening size is shown in Figure MO-3 - 6.

Parameter	Scenario	
	Baseline	Efficiency Measure
Equipment Load and Annual Use Inputs		
1. Connected load (MBtuh)	45,900	
2. Operating time (hrs/yr)	8,400	
3. Load Factor	26%	
4. Equivalent full load hours (hrs/yr)	2,218	
5. Annual Gas Use (therms/yr)	1,017,878	
Temperature and % Oxygen Inputs		
6. Flue gas temperature (F)	400	400
7. Oxygen (O ₂) in flue gas (% dry)	3.5%	3.5%
8. Combustion air temperature (F)	80	80
Gas Savings from Reducing Air Infiltration on Induced Draft Combustion		
15. Induced draft (inches of W.C. of vacuum)	0.100	
16. Total opening area (square inches)	300	0
17. Air infiltration (scfh)	127,162	0
18. Annual gas use (therms/yr)	1,017,878	930,785
19. Gas savings (%)	8.6%	
20. Gas savings (therms/year)	87,094	
Cost Savings from Reducing Air Infiltration on Induced Draft Systems		
21. Gas rate (\$/therm)	\$0.700	
22. Annual cost savings (\$/year)	\$60,966	

Figure MO-3 – 6 Potential savings for 0.1 inch negative pressure in the oven

These calculations show that it is possible to save \$43,000 to \$61,000 by closing these openings and reducing make up air entering the furnace. It should be noted that the air leakage would be reduced by 89,000 scfh for 0.05" w.c negative pressure to 127,000 scfh

for 0.05" w.c negative pressure case. This represents 14% to 20% of the total estimated make up air (620,276 scfh) used at this time.

The savings are reported for the lower negative pressure (i.e. 0.05 inch w.c.) conditions and they are \$43,000 per year.

This would reduce CO₂ emissions by about:

$$6,158 \text{ mmBtu/year} \times 113 \text{ lb CO}_2/\text{mmBtu} \approx 696,000 \text{ lb CO}_2/\text{year}$$

Implementation Cost

The cost of implementing this recommendation is relatively small if the only action taken is to seal the openings at the front and back end. This can be achieved by designing an adjustable seal that might need frequent adjustment or, preferably, use of "soft" seal using a "curtain" of flexible material such as high temperature fabric commonly used in dust collector bags or ceramic cloth with wire reinforcement so that the material can "move" with the belt without scratching or damaging the belt.

In addition to this it may be helpful to change the damper on the zone so that it allows better control of pressure in the end zones. It may also help to rebalance the oven so that effect of suction in the neighboring zones (7 and 2) are minimized.

We have assumed cost of \$20,000 (perhaps on the high side) for implementation of this recommendation.

Estimated Simple Payback

As mentioned earlier the savings for the lower negative pressure conditions are \$43,000 per year and implementation cost is \$20,000.

$$\text{Simple payback period} = (\$20,000/\$43,000) \times 12 \text{ months/year} = 5.6 \text{ months.}$$

It is very likely that this period could be much shorter.

Annual gas savings = 61,580 Therms per year or 6,158 MM Btu/year.

CO₂ savings = 696,000 lbs./year

PC-1: Install Variable Speed Drives on Transfer Pumps

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy		281,196	342,500	\$10,967	\$25,950	28 months

Analysis

The plant's cooling system includes a 9211 forming room hot well with two 50-hp pumps (only one operates), a 9212 hot well with two 30-hp pumps (only one operates), and a transfer well with five 30-hp pumps (only three operate). The fluid flow from each of these wells is controlled by a throttling valve coupled with a level indicator.

Fluid power output of a pump equals the product of volumetric fluid flow and fluid head. Throttling fluid reduces volumetric flow and therefore causes a pump to use less power in most cases. However, throttling almost always increases pump head requirements, resulting in an energy-inefficient means of flow control. An energy-savings alternative to throttling flow is installing a variable speed drive (VSD) on the motor. A variable speed drive slows pump speed down enough to meet flow requirements while simultaneously reducing fluid head output, resulting in significant energy savings.

Recommendation

We recommend replacing the throttling valve flow control with VSD flow control on the 9212 hot well pumps, the 9211 forming room hot well pumps, and the transfer well pumps.

Estimated Savings

9211 Forming Room Hot Well Pump

According to ASHRAE HVAC Systems and Equipment Handbook (2000), fluid systems with modulating two-way control valves are usually designed with pumps having "flat" characteristic curves. Fluid head across pumps with "flat" characteristic curves is relatively constant across the pumps' fluid flow operating range. Thus, a nearly linear relationship exists between percent of rated fluid flow capacity and percent of full-load output power at which the pump operates.

Management estimates that the fluid from the 9211 forming room hot well pump is throttled to about 75% of flow capacity. (We need to verify this percentage.) If so, the output power from the single 50-hp pump that operates is about:

$$50 \text{ hp} \times 75\% = 37.5 \text{ hp}$$

According to "Pumping Energy and Variable Frequency Drives" (Bernier and Bourret, ASHRAE Journal, December 1999) typical VSD and motor part-load efficiencies can be calculated using the following equations.

$$\eta_{\text{VSD}} = 50.87 + 1.283 \text{ pl} - 0.0142 \text{ pl}^2 + (5.834 \times 10^{-5}) \text{ pl}^3$$
$$\eta_{\text{motor}} = 94.187(1 - e^{-0.0904 \text{ pl}})$$

At 75% of full-load power, the motor efficiency is about:

$$\eta_{\text{motor}} = 94.187(1 - e^{-0.0904 (75\%)}) = 94\%$$

Thus, the power input to the motor is about:

$$(37.5 \text{ hp} \times 0.75 \text{ kW/hp}) / 94\% = 29.9 \text{ kW}$$

To calculate savings from reducing pump flow, engineers frequently rely on the pump/fan affinity laws. According to the pump/fan affinity relationship, the motor power output ratio (P_2/P_1) is equal to the cube of volumetric flow ratio (V_2/V_1) as follows.

$$P_2/P_1 = (V_2/V_1)^3 \rightarrow P_2 = P_1 (V_2/V_1)^3$$

For a volumetric flow ratio (V_2/V_1) of 75%, P_2 would be about:

$$P_2 = 50 \text{ hp} (75\%)^3 = 21.1 \text{ hp}$$

The percent of full-load power at which this 50-hp motor would operate at would be about:

$$21.1 \text{ hp} / 50 \text{ hp} = 42\%$$

At 42% of full-load power, the VSD and motor efficiencies would be about:

$$\eta_{\text{VSD}} = 50.87 + 1.283 (42\%) - 0.0142 (42\%)^2 + (5.834 \times 10^{-5}) (42\%)^3 = 84\%$$
$$\eta_{\text{motor}} = 94.187(1 - e^{-0.0904 (42\%)}) = 92\%$$

The input power to the motor would be about:

$$(21.1 \text{ hp} \times 0.75 \text{ kW/hp}) / (84\% \times 92\%) = 20.4 \text{ kW}$$

Annual electrical energy savings would be about:

$$(29.9 \text{ kW} - 20.4 \text{ kW}) \times 8,760 \text{ hours/year} = 83,220 \text{ kWh/year}$$

9212 Hot Well Pump

Management estimates that the fluid from the 9212 hot well pump is throttled to about 75% of flow capacity. (Again, we need to verify this percentage.) If so, the output power from the single 30-hp pump that operates is about:

$$30 \text{ hp} \times 75\% = 22.5 \text{ hp}$$

At 75% of full-load power, the motor efficiency is about:

$$\eta_{\text{motor}} = 94.187(1 - e^{-0.0904 (75\%)}) = 94\%$$

Thus, the power input to the motor is about:

$$(22.5 \text{ hp} \times 0.75 \text{ kW/hp}) / 94\% = 17.9 \text{ kW}$$

For a volumetric flow ratio (V_2/V_1) of 75%, P_2 would be about:

$$P_2 = 30 \text{ hp} (75\%)^3 = 12.7 \text{ hp}$$

The percent of full-load power at which this 30-hp motor would operate at would be about:

$$12.7 \text{ hp} / 30 \text{ hp} = 42\%$$

At 42% of full-load power, the VSD and motor efficiencies would be about:

$$\eta_{\text{VSD}} = 50.87 + 1.283 (42\%) - 0.0142 (42\%)^2 + (5.834 \times 10^{-5}) (42\%)^3 = 84\%$$
$$\eta_{\text{motor}} = 94.187(1 - e^{-0.0904 (42\%)}) = 92\%$$

The input power to the motor would be about:

$$(12.7 \text{ hp} \times 0.75 \text{ kW/hp}) / (84\% \times 92\%) = 12.3 \text{ kW}$$

Annual electrical energy savings would be about:

$$(17.9 \text{ kW} - 12.3 \text{ kW}) \times 8,760 \text{ hours/year} = 49,056 \text{ kWh/year}$$

Transfer Well Pump

Management estimates that the fluid from the transfer well pump is throttled to about 75% of flow capacity. (Again, we need to verify this percentage.) If so, the output power from the three 30-hp pumps that operate is about:

$$30 \text{ hp/pump} \times 3 \text{ pumps} \times 75\% = 67.5 \text{ hp}$$

At 75% of full-load power, the efficiency of the motors is about:

$$\eta_{\text{motor}} = 94.187(1 - e^{-0.0904 (75\%)}) = 94\%$$

Thus, the power input to the motors is about:

$$(67.5 \text{ hp} \times 0.75 \text{ kW/hp}) / 94\% = 53.9 \text{ kW}$$

For a volumetric flow ratio (V_2/V_1) of 75%, P_2 would be about:

$$P_2 = 30 \text{ hp/pump} \times 3 \text{ pumps} \times (75\%)^3 = 38.0 \text{ hp}$$

The percent of full-load power at which these motors would operate at would be about:

$$38.0 \text{ hp} / (30 \text{ hp/motor} \times 3 \text{ motors}) = 42\%$$

At 42% of full-load power, the VSD and motor efficiencies would be about:

$$\eta_{\text{VSD}} = 50.87 + 1.283 (42\%) - 0.0142 (42\%)^2 + (5.834 \times 10^{-5}) (42\%)^3 = 84\%$$

$$\eta_{\text{motor}} = 94.187(1 - e^{-0.0904 (42\%)}) = 92\%$$

The input power to the motors would be about:

$$(38.0 \text{ hp} \times 0.75 \text{ kW/hp}) / (84\% \times 92\%) = 36.9 \text{ kW}$$

Annual electrical energy savings would be about:

$$(53.9 \text{ kW} - 36.9 \text{ kW}) \times 8,760 \text{ hours/year} = 148,920 \text{ kWh/year}$$

Total

The total electrical energy savings would be about:

$$83,220 \text{ kWh/year} + 49,056 \text{ kWh/year} + 148,920 \text{ kWh/year} = 281,196 \text{ kWh/year}$$

$$281,196 \text{ kWh/year} \times \$0.039 / \text{kWh} = \$10,967$$

This would reduce CO₂ emissions by the electric utility by about:

$$148,920 \text{ kWh/year} \times 2.3 \text{ lb CO}_2/\text{kWh} \approx 342,500 \text{ lb CO}_2 / \text{year}$$

Estimated Implementation Cost

According to quotes from a VSD manufacturer, the cost of a VSD for a 50-hp motor is about \$4,700, and the cost of a VSD for a 30-hp motor is about \$3,150. Thus, the materials cost of the VSD's would be about:

$$(\$4,700 / \text{VSD} \times 1 \text{ VSD}) + (\$3,150 / \text{VSD} \times 4 \text{ VSD's}) = \$17,300$$

Generally, the installation cost of a VSD is about 50% of the purchase cost. Thus, installation cost would be about:

$$\$17,300 \times 50\% = \$8,650$$

Total cost would be about:

$$\$17,300 + \$8,650 = \$25,950$$

Estimated Simple Payback

$$(\$25,950 / \$10,967 / \text{year}) \times 12 \text{ months/year} = 28 \text{ months}$$

PC-2: Install Variable Speed Drives on Cold Well Building Pumps

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy		306,600	705,200	\$11,957	\$14,175	14 months

Analysis

Cooling water is pumped from the plant's cold well through the 9211 area, the 9212 area, and other areas of the plant including air compressors and air conditioners. This water is pumped at a constant 1,500 gpm flow rate by three 30-hp pumps. Water leaves the cold well at a temperature of 85 F and rises to a temperature of about 86 F as it passes through the 9211 and 9212 areas. This is only a 1 F temperature rise. Management believes that this temperature rise has room to be increased, thus flow could be reduced. To regulate flow, temperature and differential pressure indicators could be installed in the cooling line immediately after the areas where cooling is needed. These indicators could be coupled with variable speed drives (VSD's) to speed up or slow down the pumps to meet the cooling load. VSD's would be good for this cooling loop because it includes air conditioners whose cooling load varies throughout the year.

Recommendation

We recommend installing VSD's on the three 30-hp cold well building pumps. These VSD's would be controlled by temperature indicators immediately downstream of cooling loads and by differential pressure sensors between the supply and return lines. Based on temperature and differential pressure, VSD's could adjust fluid output to meet the plant's cooling demand.

Estimated Savings

The cold well building pumps in operation consist of three 30-hp pumps. (We need to verify this HP rating.) Assuming the pumps have 93% premium efficiency motors, the power input to the motors is about:

$$(30 \text{ hp/pump} \times 3 \text{ pumps} \times 0.75 \text{ kW/hp}) / 93\% = 72 \text{ kW}$$

To calculate savings from reducing pump flow, engineers frequently rely on the pump/fan affinity laws. According to the pump/fan affinity relationship, the motor power input ratio (P_2/P_1) is equal to the cube of volumetric flow ratio (V_2/V_1) as follows.

$$P_2/P_1 = (V_2/V_1)^3 \rightarrow P_2 = P_1 (V_2/V_1)^3$$

Management estimates that fluid flow could be reduced to about 75% on average while still meeting cooling demand. (We need to verify this percentage.) If so, the average motor power output would be about:

$$P_2 = (30 \text{ hp/pump} \times 3 \text{ pumps}) \times (75\%)^3 = 38 \text{ hp}$$

The percent of full-load power at which these three 30-hp motors would run at would be about:

$$38 \text{ hp} / (30 \text{ hp/motor} \times 3 \text{ motors}) = 42\%$$

According to “Pumping Energy and Variable Frequency Drives” (Bernier and Bourret, ASHRAE Journal, December 1999) typical VSD and motor part-load efficiencies can be calculated using the following equations.

$$\eta_{\text{VSD}} = 50.87 + 1.283 \text{ pl} - 0.0142 \text{ pl}^2 + (5.834 \times 10^{-5}) \text{ pl}^3$$
$$\eta_{\text{motor}} = 94.187(1 - e^{-0.0904 \text{ pl}})$$

At 42% load, the efficiencies would be about:

$$\eta_{\text{VSD}} = 50.87 + 1.283 (42\%) - 0.0142 (42\%)^2 + (5.834 \times 10^{-5}) (42\%)^3 = 84\%$$
$$\eta_{\text{motor}} = 94.187(1 - e^{-0.0904 (42\%)}) = 92\%$$

The input power to the motors would be about:

$$(38 \text{ hp} \times 0.75 \text{ kW/hp}) / (84\% \times 92\%) = 37 \text{ kW}$$

Annual electrical energy savings would be about:

$$(72 \text{ kW} - 37 \text{ kW}) \times 8,760 \text{ hours/year} = 306,600 \text{ kWh/year}$$
$$306,600 \text{ kWh/year} \times \$0.039 / \text{kWh} = \$11,957 / \text{year}$$

This would reduce CO₂ emissions by the electric utility by about:

$$306,600 \text{ kWh/year} \times 2.3 \text{ lb CO}_2/\text{kWh} \approx 705,200 \text{ lb CO}_2 / \text{year}$$

Estimated Implementation Cost

According to quotes from a VSD manufacturer, the cost of a VSD for a 30-hp motor is about \$3,150. Thus, the materials cost of the VSD's would be about:

$$\$3,150 / \text{VSD} \times 3 \text{ VSD's} = \$9,450$$

Generally, the installation cost of a VSD is about 50% of the purchase cost. Thus, installation and total cost would be about:

$$\text{Installation: } \$9,450 \times 50\% = \$4,725$$
$$\text{Total: } \$9,450 + \$4,725 = \$14,175$$

Estimated Simple Payback

$$(\$14,175 / \$11,957 / \text{year}) \times 12 \text{ months/year} = 14 \text{ months}$$

PC-3: Install Variable Speed Drive on Chilled Water Supply Pumps

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy		841,157	1,935,000	\$32,805	\$18,600	7 months

Analysis

The 9211 and 9212 areas each are equipped with an air handling unit that maintains incoming makeup air to the plant at a temperature of 50 F. Makeup air flow is constant to the air handlers. Each air handler has an associated cooling loop consisting of two 500-ton variable speed chillers and two 75-hp cooling water supply pumps. The cooling loop in the 9211 area also serves a binder that has a steady year-round 70-ton cooling load. Incoming air temperature to the plant from the air handlers is controlled by 3-way valves that either direct cooling water to the air handlers or bypass cooling water around the air handlers. This is demonstrated in the figure below.

The cooling loops in both the 9211 and 9212 areas are constant flow systems. Therefore, pumping energy is constant regardless of season of year or cooling demand from the air handlers. Rather than bypassing fluid that is not needed for cooling, pumping energy would be saved if fluid was instead slowed down when cooling load was less than maximum. This could be achieved through the use of a variable speed drive (VSD) that is coupled to the existing temperature sensors in the air handling unit.

Recommendation

We recommend 1) shutting the valves on the bypass loops and 2) installing variable speed drives (VSD's) on the 9211 and 9212 cooling supply pumps. The VSD's would vary cooling water flow through air handling units according to outdoor air temperature during periods of the year when cooling is needed. During the four months of the year when outdoor air temperature is consistently below 50 F (typically December through March) and cooling water is not needed in air handling units, we recommend 3) shutting off the 9212 cooling supply pump. The 9211 cooling supply pump, however, could not be turned off because it would need to service the binder year-round.

Estimated Savings

Only one 75-hp supply pump operates at a time in each cooling loop. Assuming the pumps have 93% premium efficiency motors, the combined power input to the motors is about:

$$(75 \text{ hp/pump} \times 1 \text{ pump/loop} \times 2 \text{ loops} \times 0.75 \text{ kW/hp}) / 93\% = 121 \text{ kW}$$

The annual electrical energy consumed by the supply pumps is about:

$$121 \text{ kW} \times 8,760 \text{ hours/year} = 1,059,960 \text{ kWh/year}$$

To calculate savings from reducing pump flow, engineers frequently rely on the pump/fan affinity laws. According to the pump/fan affinity relationship, the motor power input ratio (P_2/P_1) is equal to the cube of volumetric flow ratio (V_2/V_1) as follows.

$$P_2/P_1 = (V_2/V_1)^3 \rightarrow P_2 = P_1 (V_2/V_1)^3$$

According to "Pumping Energy and Variable Frequency Drives" (Bernier and Bourret, ASHRAE Journal, December 1999) typical VSD and motor efficiencies when operating at a percentage of full-load power (pl) can be calculated using the following equations.

$$\eta_{\text{VSD}} = 50.87 + 1.283 \text{ pl} - 0.0142 \text{ pl}^2 + (5.834 \times 10^{-5}) \text{ pl}^3$$
$$\eta_{\text{motor}} = 94.187(1 - e^{-0.0904 \text{ pl}})$$

The equation to find electrical power input to a motor with a VSD is as follows.

$$E_2 \text{ (kW)} = [P_2 \text{ (hp)} \times 0.75 \text{ kW/hp}] / (\eta_{\text{VSD}} \times \eta_{\text{motor}})$$

If VSD's were installed on the cooling supply pumps, pump speed would vary throughout the year depending on outdoor temperature. To estimate pump speed as a function of temperature, we assume the pumps would be at maximum speed during high summer temperatures of around 100 F and would be at minimum speed when temperature drops down to around 50 F. Thus, we developed the following linear relationship of volumetric flow ratio (V_2/V_1) as a function of outdoor air temperature, T (F).

$$V_2/V_1 = [T (F) / 50] - 1$$

In a variable speed HVAC cooling loop, fluid must always be moving, and volumetric flow rate cannot fall below 30% of maximum. Thus, V_2/V_1 must always equal 30% or higher. An exception would be during winter if the pump for the 9212 area were shut off, in which case its flow rate would, of course, fall to zero. The pump for the 9211 area would also be able to shut off during winter if a dedicated chiller was installed for the binder. This option is explored in AR PC 4: Install 100-ton Dedicated Chiller for Binder Room.

We used the preceding equations with TMY2 hourly temperature data (<http://rredc.nrel.gov/>) for Toledo, OH over a year to simulate hour-by-hour electrical energy consumed by the cooling supply pumps if VSD's were installed. According to the simulation, the annual electrical energy consumption of the 9211 pump would be about 129,337 kWh, and the annual consumption of the 9212 pump would be about 89,466 kWh. Annual electrical energy savings would be about:

$$1,059,960 \text{ kWh/year} - (129,337 \text{ kWh/year} + 89,466 \text{ kWh/year}) = 841,157 \text{ kWh/year}$$

$$841,157 \text{ kWh/year} \times \$0.039 \text{ /year} = \$32,805 \text{ /year}$$

This would reduce CO₂ emissions by the electric utility by about:

$$841,157 \text{ kWh/year} \times 2.3 \text{ lb CO}_2/\text{kWh} \approx 1,935,000 \text{ lb CO}_2 \text{ /year}$$

Estimated Implementation Cost

According to quotes from a VSD manufacturer, the cost of a VSD for a 75-hp motor is about \$6,200. Thus, the materials cost of the VSD's would be about:

$$\$6,200 \text{ /VSD} \times 2 \text{ VSD's} = \$12,400$$

Generally, the installation cost of a VSD is about 50% of the purchase cost. Thus, installation cost would be about:

$$\$12,400 \times 50\% = \$6,200$$

Total cost would be about:

$$\$12,400 + \$6,200 = \$18,600$$

Estimated Simple Payback

$$(\$18,600 / \$32,805 \text{ /year}) \times 12 \text{ months/year} = 7 \text{ months}$$

PC-4: Install 100-ton Dedicated Chiller for Binder Room

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy		910,212	2,093,000	\$35,498	\$99,900	34 months

Analysis

This AR is an alternative to “AR PC 3: Install Variable Speed Drive on Chilled Water Supply Pumps”.

The 9211 area has two 500-ton variable-speed chillers that service the section’s air handling unit and the binder. According to management, the binder operates year-round with a cooling load of 70 tons. Because cooling is supplied to the air handling unit only during warm summer months, a 500-ton chiller must supply cooling solely to the 70-ton binder load for much of the year. When doing so, the chiller operates significantly underloaded.

The chillers serving the 9211 area have variable speed compressors, thus their part-load efficiency is higher than standard chillers. However, the efficiency of even variable speed chillers drops significantly when operating below 40% of capacity, as shown in the diagrams below.

Source: “All-Variable Speed Centrifugal Chiller Plants: Can We Make Our Plants More Efficient?” T. Hartman. www.automatedbuildings.com

Source: “Variable Speed Drive Turbo Chiller NART-I Series Featuring World’s Highest Efficiency.” Seki, Ueda, Taito, Shirakata, and Koga. Mitsubishi Heavy Industries, Ltd. Technical Review. Vol. 41 No. 1. February, 2004.

If a 100-ton dedicated chiller met the binder’s 70-ton cooling load, the chiller would operate near its full-load capacity year-round, thus being more efficient than a 500-ton chiller running significantly underloaded during winter months. In addition, a 100-hp dedicated chiller would allow the 75-hp cooling supply pump in the 9211 area to be shut off during winter months, similarly to what we recommended in AR X for the 9212 cooling supply pump.

Recommendation

We recommend installing a 100-ton dedicated chiller to meet the binder’s 70-ton cooling load in the 9211 area. The performance of a chiller without variable speed capabilities does not severely degrade until it is underloaded below about 60% of capacity, as shown in the first graph. Thus, a standard chiller would be appropriate as a dedicated chiller operating close to full capacity to meet a steady cooling load.

In addition, we recommend installing variable speed drives (VSD’s) on the 9211 and 9212 cooling supply pumps and shutting the valves on the bypass loops in the air handling units; as also recommended in AR X. A dedicated chiller would allow the supply pump for the 9211 cooling loop as well as the pump for the 9212 loop to be shut off during the winter when no cooling needs to be supplied to the air handling units.

Estimated Savings

Two types of energy savings would result from installing a dedicated chiller: 1) improved chiller efficiency and 2) pumping energy savings from cooling supply pumps. To quantify savings from improved chiller efficiency, we consider only four months out of the year when outdoor air temperature is consistently below 50 F and no cooling load is demanded from the air handlers. In reality, there are more periods during the year when

cooling load from the air handlers is low and additional savings would be realized. Thus, the following savings calculations are slightly conservative.

When only serving the 70-ton load of the binder, the percent of full-load capacity at which the 500-ton chiller operates is about:

$$70 \text{ tons} / 500 \text{ tons} = 14\%$$

According to the middle few performance lines in the second chart above, the COP of a VSD chiller is about 6 when near 20% loaded and about 8 when near full-load. Thus, we assume the current chiller has a COP of 6 during four winter months, and the COP of a dedicated chiller would improve to 8. If so, the current power input to the 500-ton chiller during winter is about:

$$(70 \text{ tons} / 6) \times 3.52 \text{ kW/ton} = 41 \text{ kW}$$

The power input to a 100-ton dedicated chiller would be about:

$$(70 \text{ tons} / 8) \times 3.52 \text{ kW/ton} = 31 \text{ kW}$$

Electrical energy savings from improved chiller efficiency would be about:

$$(41 \text{ kW} - 31 \text{ kW}) \times 4 \text{ months/year} \times 30.4 \text{ days/month} \times 24 \text{ hours/day} = 29,184 \text{ kWh/year}$$

Based on chiller performance curve show above (Hartman), we calculate that the 9211 and 9212 cooling supply pumps together consume about 1,059,960 kWh per year in electrical energy. We also calculate that the annual electrical energy consumed by the 9212 pump would decrease to about 89,466 kWh if equipped with a VSD and shut off during four winter months. If a dedicated chiller were installed, the annual consumption of the 9211 pump would also decrease to about 89,466 kWh. Thus, pumping electrical energy savings would be about:

$$1,059,960 \text{ kWh/year} - (89,466 \text{ kWh} + 89,466 \text{ kWh}) = 881,028 \text{ kWh/year}$$

An additional cooling supply pump would be needed for the 100-ton chiller. However, we assume that additional energy consumed by the new pump would offset the energy saved from the existing 75-hp supply pump due to its decrease in pumping load.

The annual electrical energy savings would be about:

$$29,184 \text{ kWh/year} + 881,028 \text{ kWh/year} = 910,212 \text{ kWh/year}$$
$$910,212 \text{ kWh/year} \times \$0.039 \text{ /year} = \$35,498 \text{ /year}$$

This would reduce CO₂ emissions by the electric utility by about:

$$910,212 \text{ kWh/year} \times 2.3 \text{ lb CO}_2\text{/kWh} \approx 2,093,000 \text{ lb CO}_2 \text{ /year}$$

Estimated Implementation Cost

According to RS Means Mechanical Cost Data 2002, a 100-ton chiller costs about \$63,000, and installation costs about \$18,300. According to quotes from a VSD manufacturer, the cost of a VSD for a 75-hp motor is about \$6,200. Generally, the installation cost of a VSD is about 50% of the purchase cost. The cost of new equipment would be about:

$$\$63,000 + (\$6,200 / \text{VSD} \times 2 \text{ VSD's}) = \$75,400$$

The cost of installation would be about:

$$\$18,300 + (\$6,200 / \text{VSD} \times 50\% \times 2 \text{ VSD's}) = \$24,500$$

Total project cost would be about:

$$\$75,400 + \$24,500 = \$99,900$$

Estimated Simple Payback

$$(\$99,900 / \$35,498 / \text{year}) \times 12 \text{ months/year} = 34 \text{ months}$$

CA-1: Purchase Compressor Sequencer

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy		499,320	1,148,000	\$19,473	\$20,000	12 months

Analysis

In the Compressed Air System Analysis section of this report, we estimated actual and maximum compressed air output and power use of the compressors based on measured current draw of the compressors and manufacturers specifications. A summary of these estimates is shown below.

Compressor (hp)	Cactual (scfm)	Cmax (scfm)	Pactual (kW)	Pmax (kW)
700	3,325	3,500	462	485
217	671	958	138	175
75	41	315	41	63
40	42	168	23	32
Total	4,079	4,941	664	755

The data show that on average the compressors generate about 4,079 scfm of compressed air and have a peak capacity of about 4,941 scfm. This indicates that if the compressors were properly staged, or sequenced, the average compressed air demand could be met by the 700 hp and 217 hp compressors, and the 75-hp and 40-hp compressors could be turned off rather than running part loaded. This would result in significant energy savings, with no adverse production effects.

To stage the compressors using activation pressures, the base load compressor would have the highest activation pressure, followed by the first lag compressor, the second lag compressor and the third lag compressors. The lag compressors would turn on only when the plant air pressure dropped low enough to start them, and would turn off after the plant air pressure increased high enough that they were no longer needed. In addition, compressors with poor part load efficiency, such as the 217-hp modulating compressor should be base loaded, and compressors with good part-load efficiency, such as the 700-hp reciprocating compressor should be designated as the lag compressors. Finally, the lag compressors should be equipped with automatic shutoff control, so that they completely turn off after running unloaded for more than about 10 minutes.

In plants with a single compressor room and compressed air storage tank, this is relatively easy to do. Unfortunately, your plant has distributed compressors and storage tanks. Thus, properly staging these compressors will probably require a dedicated sequencer, and may require centralized compressed air storage.

Recommendation

We recommend contacting a compressed air system company to install a sequencer and recommend where and how to install the pressure controls.

Estimated Savings

According to the Compressed Air System Analysis section of this report the plant compressed air demand is about 4,079 scfm. According to product literature, total compressed air capacity of the 700-hp Ingersoll-Rand and 200-hp Atlas Copco compressors is about:

$$3,500 \text{ scfm} + 958 \text{ scfm} = 4,458 \text{ scfm}$$

This indicates that compressed air demand could be accommodated solely by these two compressors and reduce the amount of compressors operating a part-load operation. Ideally, the 700-hp compressor should operate as the base load and the 217-hp compressor as the first lag compressor. However, the 217-hp compressor employs modulation control, which results in relatively poor part-load efficiency. In comparison, the part load efficiency of the 700-hp reciprocating compressor is very good. Thus, we recommend running the 217-hp compressor as base load and the 700-hp compressor as the first lag compressor. If this were done, the average compressed air generated by the 700-hp compressor would be about:

$$4,079 \text{ scfm} - 958 \text{ scfm} = 3121 \text{ scfm}$$

For the 700-hp reciprocating compressor, the fraction of full load power, FP, is about equal to the fraction of full-load capacity, FC. Thus, the power draw of the 700-hp compressor at this output capacity would be about:

$$FC = C_{\text{actual}} / C_{\text{max}} = 3,121 \text{ scfm} / 3,500 \text{ scfm} = 0.89$$

$$FP = P_{\text{actual}} / P_{\text{max}} = FC = 0.89$$

$$P_{\text{actual}} = FP \times P_{\text{max}} = 0.89 \times 485 \text{ kW} = 433 \text{ kW}$$

The total power draw by the 217-hp compressor at full load and the 700-hp compressor at 89% of full load would be about:

$$175 \text{ kW} + 432 \text{ kW} = 607 \text{ kW}$$

Currently, the compressor power draw is about 664 kW. According to management, the compressors operate continuously, thus, the power and energy savings would be about:

$$664 \text{ kW} - 607 \text{ kW} = 57 \text{ kW}$$

$$57 \text{ kW} \times 8,760 \text{ hours/year} = 499,320 \text{ kWh/year}$$

$$499,320 \text{ kWh/year} \times \$0.039 / \text{kWh} = \$19,473 / \text{year}$$

This would reduce CO₂ emissions by the electric utility by about:

$$499,320 \text{ kWh/year} \times 2.3 \text{ lb-CO}_2 / \text{kWh} = 1,148,000 \text{ lb-CO}_2 / \text{year}$$

Estimated Implementation Cost

Compressor sequencers with controls may cost up to \$20,000. Additional compressed air storage may also be needed, but we included an estimate for the cost of additional storage in Assessment Recommendation CA-2.

Simple Payback

$\$20,000 / \$19,473 \text{ /year} \times 12 \text{ months/year} = 12 \text{ months}$

CA-2: Increase Compressed Air Storage and Decrease Pressure

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy		601,037	1,400,000	\$23,440	\$20,000	10 months

Analysis

A 700-hp Ingersoll-Rand reciprocating, 200-hp Atlas Copco rotary-screw, a 75-hp Ingersoll-Rand “Sierra” rotary-screw and a 40-hp Atlas Copco rotary-screw compressor create compressed air for the plant. The setpoint pressures of each compressor are shown in the table below.

Compressor	Loaded/Modulating Pressure
700-hp Ingersoll-Rand	107 psig
200-hp Atlas Copco	105 psig
75-hp Ingersoll-Rand	105 psig
40-hp Atlas Copco	105 psig

Taking spot measurements of compressed air line pressure throughout the plant indicates that the average line pressure is about 100 psig. According to management, the plant pressure requirement is 80 psig, but the line pressure is kept artificially high to accommodate for periods of high compressed air demand. This indicates a that the pressure is kept about 20 psig higher than necessary to accommodate pressure swings caused by variable loads.

Installing more compressed air storage near sensitive loads should enable you to decrease the activation pressures of the compressors by about 10 psig. This would reduce compressor energy use, reduce the compressed air leak load and reduce wear and tear on the compressors and pneumatic equipment. In addition, as the following graph shows, the quantity of compressed air that a compressor can supply increases at lower pressures.

GA160W Pressure (psig) vs Output Capacity (scfm)

Recommendation

We recommend adding up to 500 gallons of compressed air storage and reducing the activation pressures of the air compressors by about 10 psig each.

Estimated Savings

The cost penalty for operating at a high system pressure is found using fractional savings. The fractional savings for operating at a reduced average discharge pressure, P_{2low} , compared to a high average discharge pressure, P_{2high} , when the inlet air pressure is P_1 is about:

$$\text{Fractional Savings} = \frac{(P_{2high} / P_1)^{0.286} - (P_{2low} / P_1)^{0.286}}{(P_{2high} / P_1)^{0.286} - 1}$$

700-hp Ingersoll-Rand Reciprocating Compressor

The current discharge pressure of the 700-hp Ingersoll-Rand reciprocating compressor is about 107 psig. This indicates an average pressure drop between compressor discharge and plant pressure to be about:

$$107 \text{ psig} - 100 \text{ psig} = 7 \text{ psig}$$

To maintain plant pressure at 80 psig, this indicates the discharge pressure of the 700-hp compressor could be reduced to:

$$80 \text{ psig} + 7 \text{ psig} = 87 \text{ psig}$$

Thus, the fractional savings by reducing the 700-hp compressor's discharge pressure from 107 psig to 87 psig would be about:

$$(P_{2high}/P_1)^{0.286} = [(107 \text{ psig} + 14.7 \text{ psia}) / 14.7 \text{ psia}]^{0.286} = 1.83$$
$$(P_{2low}/P_1)^{0.286} = [(87 \text{ psig} + 14.7 \text{ psia}) / 14.7 \text{ psia}]^{0.286} = 1.74$$

$$(1.83 - 1.74) / (1.83 - 1) = 10.8\%$$

Therefore, the average power savings by reducing discharge pressure would be about:

$$462 \text{ kW} \times 10.8\% = 49.9 \text{ kW}$$

200-hp Atlas Copco Rotary Screw Compressor

The current modulating pressure of the 200-hp Atlas Copco rotary-screw compressor is about 105 psig. This indicates an average pressure drop between compressor discharge and plant pressure to be about:

$$105 \text{ psig} - 100 \text{ psig} = 5 \text{ psig}$$

Thus, the fractional savings by reducing the 200-hp compressor's discharge pressure from 105 psig to 85 psig would be about:

$$(P_{2\text{high}}/P_1)^{0.286} = [(105 \text{ psig} + 14.7 \text{ psia}) / 14.7 \text{ psia}]^{0.286} = 1.82$$
$$(P_{2\text{low}}/P_1)^{0.286} = [(85 \text{ psig} + 14.7 \text{ psia}) / 14.7 \text{ psia}]^{0.286} = 1.73$$

$$(1.82 - 1.73) / (1.82 - 1) = 10.8\%$$

Therefore, the average power savings by reducing discharge pressure would be about:

$$138 \text{ kW} \times 10.8\% = 14.9 \text{ kW}$$

75-hp Ingersoll-Rand Rotary Screw Compressor

The current modulating pressure of the 75-hp Ingersoll-Rand "Sierra" rotary-screw compressor is about 105 psig. This indicates an average pressure drop between compressor discharge and plant pressure to be about:

$$105 \text{ psig} - 100 \text{ psig} = 5 \text{ psig}$$

Thus, the fractional savings by reducing the 75-hp compressor's discharge pressure from 105 psig to 85 psig would be about:

$$(P_{2\text{high}}/P_1)^{0.286} = [(105 \text{ psig} + 14.7 \text{ psia}) / 14.7 \text{ psia}]^{0.286} = 1.82$$
$$(P_{2\text{low}}/P_1)^{0.286} = [(85 \text{ psig} + 14.7 \text{ psia}) / 14.7 \text{ psia}]^{0.286} = 1.73$$

$$(1.82 - 1.73) / (1.82 - 1) = 10.8\%$$

Therefore, the average power savings by reducing discharge pressure would be about:

$$40 \text{ kW} \times 10.8\% = 4.0 \text{ kW}$$

Total

The aggregate power savings by reducing loaded or discharge pressure from each respective compressor would be about:

$$49.9 \text{ kW} + 14.9 \text{ kW} + 4.0 \text{ kW} = 68.8 \text{ kW}$$

According to management, the compressors operate 24 hours per, 7 days per week and 52 weeks per year. If so, the annual energy and cost savings would be about:

$$68.8 \text{ kW} \times 24 \text{ hours/day} \times 7 \text{ days/week} \times 52 \text{ weeks/year} = 601,037 \text{ kWh/year}$$
$$601,037 \text{ kWh/year} \times \$0.039 / \text{kWh} = \$23,440 / \text{year}$$

This would reduce CO₂ emissions by the electric utility by about:

$$601,037 \text{ kWh/year} \times 2.3 \text{ lb-CO}_2/\text{kWh} = 1,400,000 \text{ lb-CO}_2/\text{year}$$

Estimated Implementation Cost

Compressed air storage tanks cost about \$4 per gallon. The cost of adding 500 gallons of storage would be about \$20,000.

Simple Payback

$$\$20,000 / \$23,440 \text{ /year} \times 12 \text{ months/year} = 10 \text{ months}$$

CA-3: Institute Program to Fix Compressed Air Leaks

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy		85,848	197,500	\$3,348	\$500	2 months

Analysis

We walked through the plant and found several compressed air leaks. Leaks were most prevalent in the winder machine area, and were also present in the batch house and bag house. The following lists the location of each leak we found.

Winder #3
Winder #301
Winder #302
Winder #102
Winder #106
Winder #108
Bag house
Batch house near Zurn air & gas dryer
Batch house on mixed batch transportation vessel
Air dryer near Gardner Denver compressor

We placed an orange tag marked “DOE” near each leak. Leaks in the compressed air system unnecessarily increase plant compressed air demand, thereby requiring compressors to draw more power. Regularly inspecting and maintaining leaks in the system would reduce compressed air demand and save electricity.

In the following analysis, we calculate that about 70 scfm of compressed air are consumed by the leaks we found. According to the Air Compressor Analysis section of the report, the plant’s air compressors generate an average of about 4,046 scfm of compressed air. Thus, the leaks we found constitute about 2% of compressed air generation. Generally, leaks consume about 10% - 30% of a typical plant’s air demand. Thus, the plant appears to be doing a good job identifying and fixing leaks.

Recommendation

We recommend repairing compressed air leaks we identified with orange tags. In addition, we recommend instituting a program to inspect the compressed air system and repair leaks every month. Leaks can easily be identified with an ultrasonic leak detector.

Estimated Savings

Based on the Moss Equation (Ingersoll-Rand Air Compressors: Condensed Air Power Data) the volumetric flow rate of compressed air through holes can be calculated as:

$$V \text{ (scfm)} = 8.8356 \times D^2 \text{ (in)} \times P \text{ (psia)}$$

where $P \text{ (psia)} = P \text{ (psig)} + 14.7 \text{ psi}$. We measured the average compressed air pressure in the plant to be 100 psig.

All of the leaks were medium-to-large, hence we estimate that on average they have an equivalent hole diameter of about $1/12''$. The volumetric flow rate through a $1/12''$ hole at 100 psig is about:

$$V \text{ (scfm)} = 8.8356 \times (1/12'')^2 \times (100 \text{ psig} + 14.7 \text{ psi}) = 7.0 \text{ scfm}$$

The entire volumetric flow rate of the compressed air through the ten leaks we identified is about:

$$7.0 \text{ scfm/leak} \times 10 \text{ leaks} = 70 \text{ scfm}$$

According to the Compressed Air Analysis section of the report, the 700-hp Ingersoll-Rand compressor generates about 3,325 scfm of compressed air, which is about 82% of total plant demand. We measured the full-load input power to this compressor to be 485 kW, and found its full-load generation capacity to be 3,500 scfm. Thus, its ratio of input power to output generation ratio is 0.14 kW/scfm. The power saved from reducing its load by 70 scfm would be about:

$$70 \text{ scfm} \times 0.14 \text{ kW/scfm} = 9.8 \text{ kW}$$

The energy saved over the plant's 8,760 annual hours of operation would be about:

$$\begin{aligned} 9.8 \text{ kW} \times 8,760 \text{ hours/year} &= 85,848 \text{ kWh/year} \\ 85,848 \text{ kWh/year} \times \$0.039 \text{ /kWh} &= \$3,348 \text{ /year} \end{aligned}$$

This would reduce CO₂ emissions by the electric utility by about:

$$85,848 \text{ kWh/year} \times 2.3 \text{ lb CO}_2\text{/kWh} \approx 197,500 \text{ lb CO}_2 \text{ /year}$$

Estimated Implementation Cost

Maintenance estimates the cost to repair a leak is about \$50. The cost to fix 10 leaks would therefore be about:

$$\$50 \text{ /leak} \times 10 \text{ leaks} = \$500$$

Estimated Simple Payback

$$(\$500 / \$3,348 \text{ /year}) \times 12 \text{ months/year} = 2 \text{ months}$$

L-1: Replace HID Lights with High-Bay Fluorescent Lights

	Annual Savings				Project	
	Fuel (mmBtu)	Elec (kWh)	CO ₂ (lbs)	Dollars (\$)	Cost (\$)	Payback
Energy		920,165	2,116,000	\$35,886	\$113,500	38 months

Analysis

We counted about 380 400-W metal halide (MH) fixtures and about 70 400-W high pressure sodium (HPS) fixtures in the plant. New six-lamp T8 high-bay fluorescent (HBF) fixtures (shown in photo below) are available that consume about 50% less energy than MH and HPS fixtures, while maintaining a higher light output over time. In addition, T8 fluorescent lamps have a color rendition index (CRI) of around 80, compared to a CRI of 60 for MH lamps and 22 for HPS lamps. The CRI indicates the ability of a light source to accurately show the true color of an object; a measure of the quality of light. The improved clarity and visibility due to the higher CRI makes it seem like there is more light. Thus, high-bay fluorescent fixtures would both reduce energy use and improve lighting quality in the plant.

A 6-lamp high-bay fluorescent luminaire

Another advantage of the HBF fixtures is their quick re-strike time. Conventional HID fixtures take about 5 to 10 minutes to fully illuminate when cold and up to 20 minutes when hot. Thus, it is not practical to turn off MH lights using lighting controls such as photocells or motion sensors. However, HBF fixtures fully illuminate within seconds, making them very compatible with most types of lighting controls. The following photo shows a plant with HBF fixtures installed at a height of about 30 feet.

Photo of high-bay fluorescent lights in another plant.

Maintenance informed us that the 9211 and 9212 forehearth areas are only occupied for about 1 hour per week, and the incinerator room is only occupied for about 8 hours per day. If HBF fixtures replaced the current HID fixtures in those areas, motion sensors could be installed to reduce lighting level in unoccupied areas.

Recommendation

We recommend replacing all of the plant's high-bay 400-W MH and 400-W HPS fixtures with six-lamp T8 high-bay fluorescent fixtures. We also recommend installing motion sensors in the 9211 and 9212 forehearth areas and the incinerator room.

Estimated Savings

Electricity Savings

According to manufacturers' data, a 400-W MH fixture draws about 460 Watts, and a 400-W HPS fixture draws about 465 Watts. Thus, the total electrical energy consumed by the HID lights during the plant's 8,760 annual operating hours is about:

$$[(380 \text{ MH fixtures} \times 0.460 \text{ kW/fix.}) + (70 \text{ HPS fixtures} \times 0.465 \text{ kW/fix.})] \times 8,760 \text{ hours/year} = 1,816,386 \text{ kWh/year}$$

According to manufacturers' data, a HBF fixture with six F32T8 fluorescent lamps draws about 234 Watts per fixture. If 450 HBF fixtures replaced all of the HID fixtures and operated continuously throughout the entire year, their annual electrical energy consumption would be about:

$$450 \text{ fixtures} \times 0.234 \text{ kW/fixture} \times 8,760 \text{ hours/year} = 922,428 \text{ kWh/year}$$

If a motion sensor were installed on a HBF fixture, two of the three ballasts could be shut off during unoccupied periods, allowing each fixture to draw 1/3 of its full electrical power.

About 18 HID fixtures are located in the 9211 forehearth area, and two HID fixtures are located in the incinerator room. According to maintenance, the forehearth area is occupied about one hour per week, and the incinerator room is occupied about eight hours per day. If motion sensors were installed to turn off HBF lights in these areas when unoccupied, additional energy savings would be about:

$$[(18 \text{ fixtures} \times (168 \text{ hours/week} - 1 \text{ hour/week}) \times 52 \text{ weeks/year}) + (2 \text{ fixtures} \times (24 \text{ hours/day} - 8 \text{ hours/day}) \times 365 \text{ days/year})] \times 0.234 \text{ kW/fixture} \times 2/3 = 26,207 \text{ kWh/year}$$

Annual electrical energy savings would be about:

$$(1,816,386 \text{ kWh/year} - 922,428 \text{ kWh/year}) + 26,207 \text{ kWh/year} = 920,165 \text{ kWh/year}$$
$$920,165 \text{ kWh/year} \times \$0.039 / \text{kWh} = \$35,886 / \text{year}$$

This would reduce CO₂ emissions by the electric utility by about:

$$920,165 \text{ kWh/year} \times 2.3 \text{ lb CO}_2/\text{kWh} \approx 2,116,000 \text{ lb CO}_2 / \text{year}$$

Estimated Implementation Cost

Based upon quotes provided to us from various manufacturers (listed below), the cost of a high-bay fluorescent fixture with lamps and associated materials is about \$200 per fixture. A fixture would cost about \$250 if equipped with a motion sensor. If 430 HBF fixtures without motion sensors and 20 fixtures with motion sensors were installed, the cost would be about:

$$(430 \text{ fixtures} \times \$200 / \text{fixture}) + (20 \text{ fixtures} \times \$250 / \text{fixture}) = \$91,000$$

We estimate that it would cost about \$50 in labor to replace a fixture. If so, the labor cost of the project would be about:

$$450 \text{ fixtures} \times \$50 / \text{fixture} = \$22,500$$

The total project cost would be about:

$$\$91,000 + \$22,500 = \$113,500$$

Estimated Simple Payback

$$(\$113,500 / \$35,886 / \text{year}) \times 12 \text{ months/year} = 38 \text{ months}$$

Contacts

- Orion Lighting
(920) 892-9340

www.orionlighting.com

- Stonco Lighting
(856) 546-5500
www.stoncolighting.com
- Techbrite
(513) 772-5070
www.techbrite.com
- 1st Source Lighting
www.1stsourcelight.com
- Columbia Lighting
www.columbialighting.com

V. Implementation and Replication Plan

This section summarizes Johns Manville's plans for implementing assessment recommendations (ARs) made in the report. In addition, it summarizes potential additional savings through replicating these recommendations in other areas of the plant and throughout the industry.

Replication of the results would be carried at four levels. The first level would be at the Waterville facility that includes two plants. The Plant-1 where the assessment would be carried out would disseminate the information derived from the assessment to Plant 2 through communications to the plant operators, maintenance staff and engineering personnel. This is expected to happen practically as the implementation is carried out at Plant-1.

JM Corporate energy management would carry out the second level and third level efforts. In the second level activities JM Engineered Products Group (EPG) group would use a variety of communication tools (News Letters, Training Seminars, visits by the key personnel etc.) to inform the other three plants located in Ohio, Tennessee and Texas about the assessment results and specific actions taken by the Waterville plant as implementation of the assessment recommendations. The plants would share knowledge gained in operating and maintenance practices and availability of applicable technologies used at the Waterville facilities. In the third level activities applicable project results for commonly used energy systems (compressed air, motors, fans/blowers, pumps etc.) would be communicated to all plants through corporate energy conservation activities. A variety of means such as corporate web site, internal meetings and training programs would be used.

The fourth level replication efforts for the glass industry would be carried out through participation of The Glass Manufacturing Industry Council (GMIC) technical director, John Brown and Executive Director Michael Greenman. GMIC is a trade association of the U.S. Glass Industry that includes among its members, representatives of all four sectors: Flat, Container, Fiber and Specialty. GMIC's mission is to facilitate, organize and promote the interests and growth of the U.S. glass industry through cooperation in the areas of technology, productivity and the environment. The council develops, selects and coordinates activities and information dissemination related to energy efficiency improvement and replication of the programs with the goal of strengthening the competitive position of the U.S. glass industry in material markets.

The council promotes energy efficiency improvement programs to its membership, associated companies such as equipment suppliers, R&D organizations and consultants to adapt best practices and new technologies through its web site (www.gmic.org), special interest seminars, conferences and collaborative programs and case studies.

GMIC will actively participate in the proposed project and would use its resources and facilities to promote replication of the recommendations and results achieved in this program.

Replication within Corporation

The company has total 35 plants located within USA however four of these plants produce similar products as the Waterville plants. These four plants are located in Ohio, Tennessee and Texas. These plants are very similar and use seven melting furnaces together with other energy consuming equipment. The following energy savings estimates are for the four plants.

No. of furnaces at Plant 1 in Waterville – 2

Estimated primary energy savings at Waterville Plant 1 - 247 Billion Btu/year

Number of furnaces at four plants – 7

Estimated savings at the four plants due to replication of results

$$= (7/2) * 247 = 864.5 \text{ Billion Btu/year}$$

Energy cost, particularly electricity cost, varies considerably from plant to plant. Assuming an average gas cost of \$6 per MM Btu and electricity cost of 4.5 cents/kWh, and average energy use distribution (80% gas, 20% electricity) for the primary energy use in a plant, it is estimated that the cost of primary energy would be \$5.66 per MM Btu. Using this cost per million Btu of the primary energy the energy cost savings are estimated to be \$4,890,000 per year for the four plants of JM. Please note that we have not accounted for the possible savings resulting from replication of energy saving recommendations in the other energy user areas (ovens, compressed air, pumps, material handling systems and other machine drives including those used for cooling systems) mentioned in Table 2.

Replication throughout The Glass Industry

In any glass plant energy is used in four major areas: Batch preparation; Melting and refining; Forming and Post-forming (finishing). Distribution of energy used in these areas is shown in Figure 2 using data given in Reference 1. Almost all glass manufacturing operations use batch preparation and melting as the first step followed by different methods for forming. Energy intensity (Btu/ton) for these two processes varies in a narrow range. Overall energy use analysis for glass manufacturing shows that more than 70% of the energy is supplied by fuels, primarily natural gas and it is used for process heating (furnaces, ovens, annealing and tempering furnaces, glass lens, polishing etc.). A conservative estimate of average energy savings of 10% for process heating systems is used to calculate industry wide energy savings through implementation of recommendations made in this assessment. The same percentage (10%) is used for all other energy user systems.

The glass industry can save large amount of energy by adapting several recommendations made during this assessment. Industry wide replication efforts would be supported GMIC and State of Ohio. Energy and cost savings resulting from such replication efforts are calculated by using following assumptions. The calculations and results are summarized in Table 4.

▪ Total glass melting – production	22.11 million tons/yr. (1999 data)
▪ Natural gas savings	10% of the current use
▪ Electricity savings	10% of the current use

- Replication of energy saving methods in the in the glass industry – 15% during the next 5 to 7 years after completion of this project.

Figure 2 Energy Use Distribution and Major Sources in Glass Manufacturing

Please note that the four JM plants account for about 1.5% of the Industry glass production. The industry wide replication with limited application as mentioned above can result in energy savings of 3.01 Trillion Btu (Tbtu) per year with the energy cost savings of \$416.58 million per year.

Table 4. Calculations for Potential Energy and Cost Savings for the Glass Industry

	Values	Units	Comments
Glass production)	20.12	Million tons/year	Year 1999 data
Average energy used for melting	9.00	MM Btu/ton	Average for all types of glass
Total energy use for melting	181.08	Trillion Btu (TBtu) per year	
Potential energy savings	10	Percent	Seven years after project completion
Replication in the industry	15	Percent	
Industry growth rate	1.50	Percent per year	
Potential energy savings	3.01	TBtu/year	Avg. 1.5% production growth/year
Cost of primary energy	\$5.50	Per MM Btu	
Total energy cost savings	\$16,580,084	Per Year	Assume - no increase in energy price

VI. Project Participants

<p>Richard LeBoeuf Senior Director of Manufacturing Technology 303-978-2109 leboeuf@jm.com</p>
<p>James McCormick Plant Manager Waterville Ohio Plant 1 6050 River Road Waterville, Ohio 43566 419-878-1463 mccormickj@jm.com</p>
<p>Tom Allen Plant Engineer Waterville Ohio Plant 1 6050 River Road Waterville, Ohio 43566 419-878-1573 allentr@jm.com</p>
<p>Neil Farrow Electrical Engineer Waterville Ohio Plant 1 6050 River Road Waterville, Ohio 43566 419-878-1198 neil.farrow@jm.com</p>
<p>Arvind Thekdi Assessment Team Co-Leader E3M 15216 Gravenstein Way North Potomac, MD 20878 240-715-4333 athekdi@e3minc.com</p>
<p>Kelly Kissock, Ph.D., P.E. Assessment Team Co-Leader Energy Information Systems 208 Lonsdale Ave. Dayton, OH 45419 937-294-6101 kkissock@udayton.edu</p>
<p>Kevin Carpenter and Bill Eger Energy Information Systems 208 Lonsdale Ave. Dayton, OH 45419 937-294-6101 kkissock@udayton.edu</p>