PLANT WIDE ASSESSMENT

for

SIFCO INDUSTRIES INC.

**970 EAST 64TH STREET
CLEVELAND, OHIO 44103**

Report No. DOE-SIFCO – 1
DOE Contract No. DOE-GO-14031

Co-funded by

SIFCO

and

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

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July 6, 2005

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	6
III. Energy Use Overview	8
Electric Rate Schedules.....	9
Electricity Use Summary	10
Electricity Cost Verification and Break Down	11
Lean Energy Analysis of Electricity Use.....	12
Natural Gas Use Summary.....	14
Lean Energy Analysis of Natural Gas Use	15
Avoided Costs	17
III. Compressed Air System Analysis.....	18
Compressed Air Requirements For The 45000 MKG Counter Blow Hammer.....	18
Distribution System Requirements	22
Summary Recommendations for Convertin Hammers to Compressed Air.....	23
IV. Process Heating Analysis	25
V. Steam System Analysis.....	30
Boiler Gas Use	30
Combustion Efficiency	31
Energy and Mass Balance.....	31
Condition of Steam Leaving Hammers.....	33
VI. Assessment Recommendations.....	35
AR B1: Reduce Excess Combustion Air to Process Boilers	36
AR B2: Mechanically Clean and De-scale Tubes in Process Boilers.....	41
AR B3: Install Economizers on Process Boilers.....	43
AR B4: Redirect Condensate Return to Deaerator	49
AR S1: Install Additional Heat Exchanger to Reclaim Heat From Exhaust Steam	52
AR S2: Insulate Steam Supply Piping	54
AR FO1: Preheat Combustion Air to Forging Furnaces.....	56
AR FO2: Reduce Excess Air in Forging Furnaces	60
Payback Period.....	64
AR FO3: Use Furnace Flue Gases for Preheating Load and/or Dies	65
AR CA1: Reduce Compressed Air Demand.....	71
AR CA2: Eliminate Compressor Cooling Water.....	74
AR M1: Use High Temperature Die Materials and Lubricating Coatings	79
AR P1: Convert Steam Hammers to Compressed Air Hammers	81
VII. Implementation and Replication Plan	87
VIII. Project Participants	89

I. Executive Summary

In May 2003, Sifco Industries submitted a proposal to the U.S. Department of Energy for a plant wide assessment of the Cleveland forging plant under RFP# DE-PS07-03ID14489. The project was selected for an award and a project kick-off meeting took place at the plant on April 14, 2004. During the next year, personnel from EIS, E3M, DPS and BuyCastings.Com visited the plant. With the help of Sifco plant facilities and maintenance personnel, the team collected energy use, construction, process, equipment and operational information about the plant.

Based on this information, the team identified 13 energy savings opportunities. Near term savings opportunities have a total potential savings of about \$1,329,000 per year and a combined simple payback of about 11 months. Implementation of these recommendations would reduce CO₂ emissions by about 16,000,000 pounds per year, which would reduce overall plant CO₂ emissions by about 45%. These totals do not include another \$830,000 per year in potential savings with an estimated 9-month payback, from converting the forging hammers from steam to compressed air.

These savings opportunities are summarized in the table below. The body of the report contains more detailed descriptions of the process, how energy is used in the plant, how to track and measure savings, and the energy saving opportunities identified.

Table ES1. Summary of Assessment Recommendations (AR)

Boiler

- B1 Reduce Excess Combustion Air to Process Boilers
- B2 Mechanically Clean and Descale Tubes in Process Boilers
- B3 Install Economizers on Process Boilers
- B4 Redirect Condensate Return to Deaerator

Steam Systems

- S1 Install Additional Heat Exchanger to Reclaim Heat From Exhaust Steam
- S2 Insulate Steam Supply Piping

Furnaces and Ovens

- FO1 Preheat Combustion Air to Forging Furnaces
- FO2 Reduce Excess Air in Forging Furnaces
- FO3 Use Furnace Flue Gasses For Preheating Load and/or Dies

Compressed Air

- CA1 Reduce Compressed Air Demand
- CA2 Eliminate Compressor Cooling Water

Materials

- D1 Use High Temperature Die Materials and Lubricating Coating

Process

- P1 Convert Steam Hammers to Compressed Air Hammers

AR Number	Projected Resource Savings			Projected Cost Savings			Implement Status
	Fuel 10 ⁶ Btu/yr	Electricity kWh/yr	Emissions* lb CO ₂ /yr	Annual Savings \$/yr	Capital Cost \$	Payback (months)	
Boiler							
B1	22,367		2,527,000	\$165,068	\$0	Immediate	
B2	3,399		384,000	\$25,085	\$24,000	11	
B3	9,225	-13,426	1,018,516	\$66,932	\$60,000	11	
B4	1,334	-3,371	142,774	\$10,385	\$13,600	16	
Steam							
S1	9,722		1,099,000	\$71,748	\$10,000	2	
S2	602		68,000	\$5,923	\$2,880	6	
Furn/Ovens							
FO1	41,527		4,775,605	\$306,469	\$664,725	26	
FO2	10,194		1,172,310	\$75,232	\$28,000	4	
FO3	41,240	-74,592	4,507,262	\$297,041	\$265,106	11	
Compress Air							
CA1		142,957	328,800	\$13,438	\$13,500	12	
CA2				\$47,424	\$100,000	25	
Materials							
D1				\$245,000		Immediate	
Process							
P1	165,200	-4,051,093	6,206,802	\$831,098	\$600,000	9	
Total**	139,610	51,568	16,023,267	\$1,329,745	\$1,181,811	11	

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

** Does not include savings or implementation costs from Process Recommendation (P1)

II. Plant Overview

SIFCO Forge Group is a forging company based in Cleveland, Ohio with fully integrated facilities such as forging, pressing, heat treatment, die manufacturing, CNC machining, pickling, magnetic particle inspection, chemical milling and fluorescent penetrate equipment. The Cleveland metal forging and heat treating plant produces forged and finished OEM components for various industries including the aerospace industry in a wide variety of alloys such as titanium, stainless steel, alloy steels, nickel based alloys, copper based alloys, monel and carbon steels. Energy cost represent approximately 11% of the value-added sales of this company and are typical of similar forge and heat-treating operations.

The company was founded in 1913 and has served the US and International market for the last 90 years. This plant produces forged and finished OEM components for various industries including the aerospace industry in a wide variety of alloys such as titanium, stainless steel, alloy steels, nickel based alloys, copper based alloys, monel and carbon steels. This is a fully integrated forging facility that includes forging hammers of various sizes, hydraulic presses, forge heating furnaces, in-house heat-treatment facilities, die manufacturing equipment, CNC machines and modern equipment for pickling, magnetic particle inspection, chemical milling and fluorescent penetrant inspection.

General layout of the plant located on a 14 acres site is shown below.

General layout of the SIFCO Forge plant

The plant employs 100 hourly workers and 37 salary people. The forging process carried out at the plant includes a number of steps that are shown below.

Forging process steps carried out at SIFCO Forge plant

III. Energy Use Overview

Primary sources of energy for the plant are natural gas and electricity. Natural gas and electricity use from July 2003 to June 2004 are shown below. This represents approximately 11% of the value added sales and a significant part of the company's bottom line profit. Natural gas is used for process heating in furnaces and ovens and for steam generation in boilers to power the steam forging hammers and for space heating. Electricity is used for machine drives through use of electric motors, process cooling for cryogenic heat treatment and induction heating in the heat treating processes.

	Annual Usage	Average Demand	Annual Cost	Unit Cost	Annual CO ₂ Emissions**
Electricity	5,210,181 kWh	1,454 kW	\$500,972	\$0.098 /kWh	11,983,000 lb
Natural Gas	201,386 mmBtu*		\$1,465,801	\$7.38 /mmBtu	22,757,000 lb
Total			\$1,966,773		34,740,000 lb

(mmBtu = million Btu, mcf = thousand cubic feet, klb = thousand pounds)

* Annual total extrapolated from 336 days of data. Assumes 1mmBtu/mcf.

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

Electric Rate Schedules

Cleveland Public Power Large Commercial Winter Schedule (November – May)

Energy: Commercial Energy Adjustment (varies by month)
\$0.058919 /kWh average for 2002

Plus,

\$0.0288 /kWh	0 - 40,000 kWh
\$0.0173 /kWh	40,001 - 100,000 kWh
\$0.0140 /kWh	100,001 kWh and above

Plus,

kWh Tax (varies by month)
\$0.003651 /kWh average for 2003

Demand: \$7.28 /kW-month 0-50 kW
\$6.33 /kW-month 51 kW and above

Plus,

Voltage Supply Discount (varies by month)
-\$0.05 /kW average for 2003

Reactive Power: (varies by month)
\$0.193 /kVAr average for 2003

Large Commercial Summer Schedule (June – October)

Energy: Commercial Energy Adjustment (varies by month)
\$0.052117 /kWh average for 2002

Plus,

\$0.0331 /kWh	0 - 40,000 kWh
\$0.0207 /kWh	40,001 - 100,000 kWh
\$0.0166 /kWh	100,001 kWh and above

Plus,

kWh Tax (varies by month)
\$0.003655 /kWh average for 2002

Demand: \$7.99 /kW-month 0-50 kW
\$6.92 /kW-month 51 kW and above

Reactive Power: (varies by month)
\$0.193 /kVAr average for 2003

Electricity Use Summary

Meter Reading Date	Days	Consumption (kWh/period)	Avg. Daily Consumption (kWh/day)	Demand (kW)	Load Factor	Total Cost (\$/period)
06/30/03	30	398,110	13,270	1,393	0.40	\$42,619
07/31/03	31	308,287	9,945	1,310	0.32	\$34,850
08/31/03	31	404,928	13,062	1,379	0.39	\$43,077
09/30/03	30	404,139	13,471	1,461	0.38	\$42,860
10/31/03	31	423,868	13,673	1,357	0.42	\$38,243
11/30/03	30	410,569	13,686	1,544	0.37	\$40,831
12/31/03	31	434,168	14,005	1,433	0.41	\$39,039
01/31/04	31	505,672	16,312	1,532	0.44	\$45,532
02/29/04	29	528,669	18,230	1,688	0.45	\$48,184
03/31/04	31	510,996	16,484	1,460	0.47	\$46,155
04/30/04	30	404,402	13,480	1,506	0.37	\$39,541
06/01/04	32	386,372	12,074	1,382	0.36	\$40,041
Total/Average	367	5,120,181	13,974	1,454	0.40	\$500,972

- **Load factor** is defined as [(kWh/period) / (peak kW x hr/period)]. Load factor represents the average fraction of the peak electrical capacity used by the plant. High load factors indicate multiple operating shifts and generally correlate with lower average prices per kWh.

Electricity Cost Verification and Break Down

The table below shows the actual cost of electricity from the electricity bill and the cost we calculated using the published rate schedules. The difference between the actual and calculated costs was only about 0.01%, which indicates that the rate schedule is being properly applied.

Meter Reading Date	Days	Actual Cost (\$/period)	Calculated Energy Cost (\$/period)	Calculated Demand Cost (\$/period)	Calculated kVAr Cost (\$/period)	Calculated Total Cost (\$/period)	Difference (\$)	% Difference
06/30/03	30	\$42,619	\$32,802	\$9,625	\$192	\$42,619	\$0.33	0.00%
07/31/03	31	\$34,850	\$25,632	\$9,056	\$160	\$34,849	\$1.42	0.00%
08/31/03	31	\$43,077	\$33,378	\$9,526	\$174	\$43,077	-\$0.50	0.00%
09/30/03	30	\$42,860	\$32,932	\$9,748	\$180	\$42,861	-\$0.26	0.00%
10/31/03	31	\$38,243	\$29,507	\$8,571	\$165	\$38,242	\$0.38	0.00%
11/30/03	30	\$40,831	\$30,895	\$9,746	\$193	\$40,834	-\$3.33	-0.01%
12/31/03	31	\$39,039	\$29,759	\$9,047	\$162	\$38,967	\$71.74	0.18%
01/31/04	31	\$45,532	\$35,507	\$9,668	\$141	\$45,317	\$34.00	0.07%
02/29/04	29	\$48,184	\$37,518	\$10,648	\$179	\$48,346	-\$161.83	-0.34%
03/31/04	31	\$46,155	\$36,847	\$9,216	\$154	\$46,218	-\$63.40	-0.14%
04/30/04	30	\$39,541	\$29,861	\$9,505	\$174	\$39,540	\$1.14	0.00%
06/01/04	32	\$40,041	\$30,313	\$9,551	\$178	\$40,041	\$0.10	0.00%
Total/Average	367	\$500,972	\$384,952	\$113,907	\$2,052	\$500,912	\$60.35	0.01%

Based on this analysis, total electricity cost can be broken into demand, energy and power factor components. The figure below shows that the kWh energy charge is the largest component of the total cost of electricity.

Lean Energy Analysis of Electricity Use

The following statistical analyses of energy use were performed with Energy Explorer (Kissock, 1999). The electricity and temperature data are shown in the table below. The average temperatures during each billing period were compiled from average daily temperatures in Cleveland, Ohio from the archive maintained by the University of Dayton (<http://www.engr.udayton.edu/weather/>).

Month	Day	Year	Elec (kWh/day)	T (F)
6	30	2003	13270	-99
7	31	2003	9945	72
8	31	2003	13062	73
9	30	2003	13471	64
10	31	2003	13673	52
11	30	2003	13686	48
12	31	2003	14005	34
1	31	2004	16312	22
2	29	2004	18230	30
3	31	2004	16484	40
4	30	2004	13480	49
6	1	2004	12074	63

The figure below shows a time series plot of electricity use (kWh/day) with average outdoor air temperature (°F).

The figure below shows an XY plot of electricity use (kWh/day) versus average outdoor air temperature (°F).

The best-fit relationship between electricity use and outdoor air temperature can be modeled as:

$$E\left(\frac{\text{kWh}}{\text{day}}\right) = 19,399\left(\frac{\text{kWh}}{\text{day}}\right) - 108\left(\frac{\text{kWh}}{\text{day} \cdot ^\circ \text{F}}\right) \times T_{\text{oa}}$$

Natural Gas Use Summary

Meter Reading Date	Days	Wholesale (MCF/period)	Wholesale Cost (\$/period)	Trans (MCF/period)	Trans (MCF/day)	Trans Cost (\$/period)	Total Cost (\$/period)	Total Unit Cost (\$/MCF)
07/31/03	31	8,990	61,978	8,039	259	7,177	69,155	8.60
08/31/03	31	19,995	122,783	16,349	527	14,480	137,263	8.40
09/30/03	30	20,010	107,383	20,538	685	18,162	125,545	6.11
10/31/03	31	20,987	137,114	19,303	623	17,077	154,190	7.99
11/30/03	30	18,000	93,585	18,908	630	16,730	110,315	5.83
12/31/03	31	19,003	119,447	17,547	566	15,534	134,981	7.69
01/31/04	31	23,994	148,767	20,731	669	22,056	170,822	8.24
02/29/04	29	20,996	115,398	22,013	759	19,458	134,856	6.13
03/31/04	31	23,994	156,273	24,245	782	21,419	177,692	7.33
04/30/04	30	17,010	112,807	17,741	591	15,704	128,511	7.24
05/31/04	31	15,996	108,321	15,972	515	14,149	122,470	7.67
Total/Average	336	208,975	1,283,856	201,386	601	181,945	1,465,801	7.38

Lean Energy Analysis of Natural Gas Use

The analysis below is performed with Energy Explorer (Kissock, 1999). The data used in the analysis are shown below.

Mo	Dy	Yr	NG (mmBtu/day)	T (F)
6	30	2003	13270	-99
7	31	2003	9945	73
8	31	2003	13062	73
9	30	2003	13471	64
10	31	2003	13673	52
11	30	2003	13686	48
12	31	2003	14005	34
1	31	2004	16312	22
2	29	2004	18230	30
3	31	2004	16484	40
4	30	2004	13480	49

The figure below shows a time series plot of natural gas use (mmBtu/day) with average outdoor air (°F).

The figure below shows an XY plot of natural gas use (mmBtu/day) versus outdoor temperature (°F).

The best-fit relationship between natural gas use and outdoor air temperature can be modeled as:

$$NG \left(\frac{\text{mmBtu}}{\text{day}} \right) = 557.73 \left(\frac{\text{mmBtu}}{\text{day}} \right) - 2.96 \left(\frac{\text{mmBtu}}{\text{day}} - ^\circ \text{F} \right) \times (X_{\text{CP}} - 73.46^\circ \text{F})$$

Based on this model, the estimated breakdown of process and space heating natural gas use is:

Process Use: $557.73 \text{ mmBtu/day} \times 336 \text{ days/year} = 187,404 \text{ mmBtu/year}$

Space Conditioning Use: $-2.96 \text{ (kWh/day-}^\circ\text{F)} \times (47^\circ \text{F} - 73.46^\circ \text{F}) \times 303 \text{ days/year} = 23,731 \text{ mmBtu/year}$

Avoided Costs

The avoided cost of energy represents the cost savings you would realize by reducing your energy usage by a given unit. For example, the avoided cost of electrical energy represents the cost savings from reducing electrical energy usage by one kWh assuming that everything else (demand, power factor, etc.) stayed the same. The avoided cost of electrical energy should be calculated based on the appropriate block structure if applicable, and should include average fuel cost adjustments and taxes. Similarly, the avoided cost of electrical demand is the cost savings realized by reducing peak demand by one kW, assuming that everything else stayed the same. The avoided-cost of demand represents the cost savings from reducing demand by one kW. The avoided cost of demand should include provisions for power factor charge and block structures if applicable.

Electrical Demand: \$6.33 /kW-month (Winter Month)
 \$6.92 /kW-month (Summer Month)

Electrical Energy: \$0.0140 /kWh (Winter Month)
 \$0.0166 /kWh (Summer Month)

Natural Gas: \$7.38 /mmBtu

III. Compressed Air System Analysis

Compressed Air Requirements For The 45000 MKG Counter Blow Hammer

This portion of our report will cover SIFCO's immediate concerns regarding compressed air requirements for the recently acquired 45000 MKG counter blow hammer. The recommendations offered take into consideration the potential that SIFCO will convert their existing steam driven hammers to compressed air. The comparison of hammer energy sources is left to an unbiased evaluator. The methodology used in this report incorporated tools available to both the forging industry and those used by the compressed air industry. Data loggers recorded compressor power consumption and pressure profiles. This data was used to establish an operational profile and to gain a better understanding of the secondary compressed air demands associated with normal hammer operation. The evaluation was complicated by the fact the hammer currently being installed came with little more than stroke capabilities, previous operational allegories and installation drawing.

A paper by Erie Press Systems titled *Steam Drop Forging Hammers, Conversion from Steam to Air Drive* was provided by SIFCO. This paper outlined the process of converting Steam Driven hammers to Compressed Air. The well-written paper provides *Average Air Consumption per Rated Hammer Size*. This document will come in very handy for the existing steam driven hammers. The challenge presented with the new 45000 MKG counter blow hammer is that it is not rated in Pounds.

To establish an equivalent rating in pounds for the 45000 MKG counter blow hammer it was agreed that the hammer cylinder volumes would be compared. The table below summarizes that Erie Press Systems data provided by SIFCO and that available on the 45000 MKG.

[Volume in Cubic feet = $(\pi \times \text{Diameter}^2 \text{ in inches} \times \text{Stroke in inches})/1748$]

RATING	CYLINDER DIAMETER (in)	CYLINDER STROKE (in)	CYLINDER VOLUME (in 3)
20000 Pounds	26	53	1998
25000 Pounds	30	58	2345
45000 MKG	45	30	3976
35000 Pounds	33	60	4276
50000 Pounds	40	72	7540

Based on this simple evaluation the 45000 MKG Counter Blow Hammer would have compressed air requirements ~32-33,000 Pound Drop Hammer. Based on the explanations provided on the counter blow technology this appears very conservative considering the counter weights differential is only 5000 Pounds.

Erie Press recommends 35 cubic feet of free air per minute for each 100 Pounds of rated hammer size. Erie also recommend receivers equal to 20% capacity of the free air volume. The table below summarizes the *Average Air Consumption and Recommended Receiver Sizing per Erie*. (1 cubic foot of air =7.48 gallons)

RATING	AVERAGE AIR CONSUMPTION	RECOMMNEDE RECIEVER
20000 Pound	7000 CFM	10,472 gallons
25000 Pound	8750 CFM	13,090 gallons
45000 MKG	11,550 CFM*	17,278 gallons
35000 Pound	12,250 CFM	18,326 gallons
50000 pound	17,5000 CFM	26,180 gallons

* based on 33,000 Pound equivalent

Recent work done by the Compressed Air Challenge in conjunction with the US Department of Energy allows us to quantify and fine-tune the recommendations provided by Erie Press.

Flow Rate Into a Cylinder

The flow rate into a cylinder is based upon the amount of air needed to move the piston load and to force out exhaust air from the other side of the cylinder at a specified speed. This can be calculated by first determining the volume of the cylinder:

$$V = \text{Area} \times \text{Stroke}$$

Next we calculate the compression ratio:

$$[\text{Pressure at cylinder (psig)} + \text{abs pressure (psia)}] / \text{Abs pressure at site (psia)}$$

Now we are ready to calculate the flow rate. Flow rate in pneumatic systems is rated in cubic feet per minute (cfm). We know that a cubic foot of air can be under various pressures. A cubic foot of air can be at 90 psi, 100 psi, or 60 psi. But the cubic foot of air referred to in the term cfm generally indicates a cubic foot of air at the vicinity of the compressor intake. Free air is a term, which refers to the condition of the air supply available to a compressor. This is the air of the ambient or the surrounding air at the compressor's inlet.

After this air is compressed, it will have a smaller volume depending on how much it has been compressed. As this reduced volume passes through system piping, it is still referred to as a cubic foot of air since that is what it is under normal conditions. Since atmospheric conditions vary from day to day and from place to place, free air varies widely in its properties and therefore this is not a good term to use when comparing flow rates between various systems or for flow ratings of compressors. For this reason, cubic feet of free air is usually converted mathematically to cubic feet of standard air. A standard cubic foot of air is defined as air at a barometric pressure of 29.92 inches of mercury (this would be sea level) with a temperature of 68°F and a relative humidity of 0%

Because relative humidity and absolute temperature do not vary widely from one installation to another, an abbreviated equation is used:

$$\text{CFM} = V \text{ (in}^3\text{)} \times \text{Compression Ratio}$$

$$\text{Time to fill cylinder (secs)} \times 28.8$$

45000 MKG Counter Blow Hammer

The hammer properties are:

Bore = 4.5 inch bore

Stroke = 30 inch stroke

100 psig to cylinder

0.5 sec actuation time.

$$V \text{ (in}^3\text{)} = \text{Area} \times \text{Stroke} = (3.14 \times 4.5^2) \text{ in}^2 \times 30 \text{ in} = 47,689 \text{ (in}^3\text{)} = 27.6 \text{ ft}^3$$

$$\text{Compression ratio at 100 psig} = 100 + 14.5/14.5 = 7.9$$

Plugging numbers into the CFM formula, we get the equivalent of 26,163 CFM peak flow during the actuation. This should be the value used to size the proper regulator.

If we want to size a dedicated receiver to handle the actuation of the cylinder, we would use the following formula:

$$V_r = V_s \times P_a / dP$$

V_r = receiver size (cubic feet)

V_s = Usable stored air (cubic feet)

P_a = Atmospheric pressure (psia)

dP = Allowable pressure drop

The cylinder consumes 218 (27.6×7.9) cubic feet of air per actuation. To hold this air in storage and keep the pressure drop to within ten pounds would require:

$$V_r = 218 \times 14.5 / 10 = 316 \text{ cu. ft of storage}$$

Since there are 7.48 gallons per cubic foot, the size of the vessel would be:

$$316 \times 7.48 = 2,364 \text{ gallons.}$$

A 2,364 gallon tank would support one stroke of the cylinder.

The large CFM consumption and receiver sizes are directly related to the impact of refilling the cylinders quickly. We previously calculated the cylinder volume of the 45000 MKG counter blow hammer at 27.61 cubic feet. At 100 PSIG the cylinder would require 316 cubic

feet (27.91 x 7.9) of compressed air per stoke. If the hammer can operate at 20 blows per minute the compressed air system would need to replace 6320 (316 x 20) cubic feet of air per minute. At a minimum 4.5 CFM per HP, the average compressor house power needed to run the 45000 MKG at 20 blows per minute is 1400 HP (6320/4.5). The peak compressor CFM required was identified earlier @ > 26,000 CFM (>5800 HP). Without storage the compressed air system would be required to have system capacity that could satisfy the peaks.

Storage is used to establish a buffer between the supply and demand. By allowing a defined pressure drop in the local receiver you can 'use' a portion of the stored air. The calculation above confirmed a 2364 gallon tank would support one stoke of the cylinder and limit the pressure drop to 10 PSIG. At a rate of 20 strokes per minute you would need 47,280 gallons (2364 x 20) of storage to limit the pressure drop to the same 10 PSIG if you were not putting any compressed air into storage.

With enough compressor capacity on-line to replace the 6320 CFM the hammer would use in one minute we have the ability to determine the proper storage requirements to maintain a 10 psig pressure drop. The critical factor now is the dwell time between blows. In other words, how much time does the compressor have to replace the compressed air used. The fastest cycle time reported during our site work was 17 strokes in 14 seconds. Assuming the same 0.5 seconds (0.083 minutes) used earlier, the following formula can be used to better define the minimum system storage requirements.

$$V = \frac{t(Q_{out} - Q_{in}) \times P_a}{P_1 - P_2} = \frac{0.5/60 \times (26,163 - 6,320) \times 14.5}{100 - 90} = 241 \text{ ft}^3$$

With 7.48 gallons per cubic feet: 241 x 7.48 = 1802 gallons.

In summary, The 45000 MKG Counter Blow Hammer requires 218 cubic feet air for each 'blow'. With an actuation time of 0.5 seconds the 'peak' compressed air demand is 26,163 CFM. Assuming the maximum blows per minute of 20 the 'average' compressed air demand is 6320 CFM. 316 cubic feet (2364 gallons) of useable compressed air is required for each blow to limit the instantaneous pressure drop to 10 PSIG. Roughly 1800 gallons of system storage is required to limit the pressure drop to 10 PSIG during a single blow with 6320 CFM being added. At 20 blows per minute possible, the system would require 36,000 gallons (4813 cubic feet) of storage total.

Erie recommended receivers equal to 20% capacity of the free air volume. This equates to 17,944 gallons (2399 cubic feet). The information presented by Erie Press recommends 35 CFM per 100 # of Hammer Rating. The cylinder size of the 45000 MKG is equivalent to roughly a 33,000 pound hammer. Using the Erie recommendations SIFCO should be looking at 11,550 CFM.

COMPRESSOR CFM AND STORAGE REQUIREMENTS		
	ERIE	COMPRESSOR INDUSTRY
COMPRESSOR CFM	11,500 CFM	6,320 CFM
STORAGE REQUIREMENTS	18,000 GALLONS	36,000 GALLONS

There are a number of factors that will determine the combination of compressor horsepower and storage that will work for SIFCO. As the previous calculations show, both of these parameters are dependent on cycle time. Changing the blows per minute or the recovery time can dramatically changes both the average and peak CFM demand.

It is my opinion that the recommendations made by Erie are the most conservative, have the highest first cost, and potentially the highest operating costs. They do have the benefit of many years successful conversions.

The lower CFM and larger storage recommendations based on the compressed air industry calculations are based on certain undocumented assumptions. Those are blows per minute, recovery time, and acceptable local pressure drops. Based on my discussions with SIFCO personnel the first two are dependent on the product to be shaped and the efficiency of the counter blow operators.

Based on the dynamic nature of the compressed air demands of the forging industry the compressors selected must operate using load no-load controls and have automated controls to minimize the on-line horsepower. Without the ability to model the operating profile of the 45000 MGK counter blow hammer I would recommend SIFCO initially purchase and install three compressors each rated @ 3200 CFM or greater along with 30,000 gallons of storage. As additional hammers are converted from steam to compressed air additional compressors and storage can be added.

Distribution System Requirements

Investments in Compressed Air Distribution Piping should take the future demands into consideration. The information below is a small portion of Table 13.23 Loss of Air Pressure Due to Friction Compressed Air & Gas Handbook, Fifth Edition. Using the following formula you can confirm that a 1000 foot 12 inch distribution 'loop' can efficiently support >30,000 CFM flow. The pressure drop would be < 0.72 PSIG in a looped system.

$$\text{Pipe Loss (PSIG)} = \frac{\text{Correction Factor}}{\text{Compression Ratio}^*} \times \frac{\text{Pipe Length}}{1000}$$

Where:

compression ratio = discharge pressure in psia/inlet pressure in psia

For example

110 psig discharge = $110 + 14.5 = 124.5 / 14.5 = 8.59$

PIPE DIAMETER IN INCHES			
Pump cfm free air	8	10	12
5000	1.47	0.44	0.17
10000	4.77	1.43	0.57
15000	13.20	3.97	1.53
20000	23.60	7.00	2.74
22000	28.50	8.50	3.33
24000	33.80	10.00	3.85
26000	397.00	11.90	4.65
28000	46.20	13.80	5.40
30000	53.00	15.90	6.17

Summary Recommendations for Convertin Hammers to Compressed Air

Based on the information currently available I would make the following recommendations assuming SIFCO plans to move forward with conversion of all their hammers to compressed air.

- Install a 12-inch loop in the foundry area. Material will need to handle fully saturated compressed air. Piping may need to be insulated.
- Use existing piping for instrument air.
- Add heated desiccant dryer and filters to match 100 HP screw compressor to support non-foundry requirements
- Install 10,000 gallons of storage on drop feeding 45000 MKG Hammer
- Install additional storage per Erie Press recommendations on drops of each hammer converted to compressed air
- Install 20,000 gallons of primary storage after compressor room header
- Install additional primary storage in 10,000 gallon increments as additional hammers are converted to compressed air
- All tanks and drip legs to have zero air loss condensate drains. Outdoor tanks require drains equipped with freeze protection.
- Install 3 oil-free screw compressors with load no-load controls rated @ 3250 CFM @ 100-110 PSIG. Compressor should be without after-coolers.
- All compressors should have kW transducers to monitor future power consumption
- Install Flow Meter in compressor room discharge piping to monitor flow
- Install Pressure Transducer in Compressor Room and at farthest point on loop

- Sequencing shall be integral to Microprocessors controlling compressor
- Compressor manufacturer shall have available 24/7 monitoring service that links complete installation to an automated watchdog and data warehouse system.

The recommendations above are based on utilizing oil-free rotary screw compressors without after-coolers. This approach eliminates the environmental and safety risks associated with oil carryover from lubricated designs and allows for Load No-Load control to most efficiently match compressed air demand. The 3250 CFM (700 HP) is a functional building block for a compressed air system that would support the entire forging area.

IV. Process Heating Analysis

Natural gas is used for process heating in furnaces, for steam generation in boilers and space heating. Electricity is used for machine drives through use of electric motors, process cooling for cryogenic heat treatment and induction heating in the heat treating processes. Figure 1 shows a role and location of heating system (fired furnaces and boilers) in the production line for the plant. The heating furnaces include steel reheating furnaces for forging, heat treating furnaces for forged parts and boilers that supply steam for the hammers and presses. The heat treating furnaces use direct fired burners and, in few cases, induction heating. It should be heated that induction heating is used only during few hours of the year and hence has not been considered for this report.

Figure 1. Production operations and location of furnaces - boilers

A list of boilers and furnaces using significant energy and their rated capacity are given in Figure 2. It is apparent that boilers are the major users of natural gas. A large percentage of total natural gas used in the plant is used in boilers for steam generation, and the boilers together with a few rotary and pusher furnaces use more than 90% of the total gas used in the plant.

Figure 2. Rated heat consumption capacities of Boilers and Furnaces

During this assessment we were given log-sheets for the boilers and we had taken spot-check of furnace operations for all large gas fired furnaces to measure firing rate and percentage firing. The readings are used to estimate annual energy used by boilers and furnaces. These estimates should be considered only typical since the furnace readings were for a particular time and there is no guarantee that the furnaces operate at the same conditions all the time. However this “spot-check” gives us an idea about the trend in energy use for fired equipment at the plant. Figure 3 shows energy use distribution in terms of MMBtu/year and estimated fuel cost/year and Figure 4 shows graphical distribution for natural gas use among these fired equipment based on above mentioned survey.

Equipment	MM Btu/year	Gas cost**/year	Cumulative %
Two Boilers	165,188	\$ 1,219,084	65%
2154 PUSHER	21,594	\$ 159,364	73%
2174 PUSHER	12,810	\$ 94,538	78%
8001 BOX	10,400	\$ 76,752	82%
3553 ROTARY	6,900	\$ 50,922	85%
3700 ROTARY	6,750	\$ 49,815	88%
3556 ROTARY	6,720	\$ 49,594	90%
5546 ROTARY	5,063	\$ 37,361	92%
3554 ROTARY	4,200	\$ 30,996	94%
3555 BOX	3,864	\$ 28,516	95%
3552 ROTARY	3,240	\$ 23,911	97%
2175 PUSHER	2,928	\$ 21,609	98%
168 BOX	2,160	\$ 15,941	99%
5782 BOX	960	\$ 7,085	99%
2157 PIT	838	\$ 6,183	99%
2063 PIT	812	\$ 5,993	100%
380 BOX	635	\$ 4,687	100%
Total	255,061	\$ 1,882,349	
Note: Cost based on \$7.38/MM Btu nat. gas cost			

Figure 3. Estimated energy use and energy cost for the boilers and furnaces

The total gas cost estimated here is more than the gas cost given in earlier section since the cost given in this table is based on \$7.38 per MM Btu and the cost given in the earlier section may be based on lower costs for earlier years. It is also likely that the assumed operating

hours and/or the firing rate for the furnaces may be different from what we have accounted for here. In any case, it is apparent that almost 2/3rd of the natural gas cost is for operating the boilers, and 90% of the total cost is for two boilers with six furnaces used for steel reheating. In view of this, any significant reduction in fuel use must consider all opportunities in energy reduction for the boiler operations.

Figure 4. Graphical distribution for natural gas use

The boiler operation was reviewed with the plant personnel including two boiler operators. They supplied the boiler gas flow and steam flow logs that were analyzed to study heat requirement for steam generation. Figure 5 shows a graph of gas use vs. steam production.

Figure 5: Natural gas use vs. steam production for boilers

Figure 6. Energy used per lb. of steam for SIFCO boilers

Estimates of heat use for steam generation, based on flow meter readings for natural gas and steam flow rates (Figure 6) indicate that one lb. of steam uses 969 Btu/lb. This is lower than what we can expect from theoretical considerations. For example total heat for saturated steam at 115 psig pressure is about 1200 Btu/lb. and heat of feed water at about 250 deg. F. is 210 Btu/lb. The net heat required for steam generate would be 990 Btu/lb. With boiler efficiency of approximately 75%, actual heat required for boiler is $(990/0.75 =) 1320$ Btu/lb. This is much greater than the calculated value of 966 Btu/lb. using the boiler log sheets. Hence it is likely that the gas meter or steam flow meter readings are in question. They should be checked for accuracy and or application of proper correction factors if applicable.

The furnace operations were reviewed by using services of Conrad Kacsik Instrument Systems, Inc. who is providing instrumentation calibration and other support for SIFCO. They documented operating data for most of the operating furnaces. The results of the survey are shown in Figure 7. The table indicates furnace name or designation,

SIFCO - HEAT TREAT DEPARTMENT							
FURNACE #	O2 %	COMB %	CO ppm	EXH TEMP	OPER TEMP	OUTPUT %	BTU RATING
3555 BOX	5.5	0	0	1128	1055	23	4,200,000
168 BOX	7.3	0	0	1554	1650	20	2,700,000
380 BOX	7.7	0	0	1480	1500	21	756,000
5782 BOX	11.5	0	0	645	1200	24	1,000,000
8001 BOX	3.9	0	0	1588	1640	100	2,600,000
2063 PIT	12.5	0	0	1140	1200	35	580,000
2157 PIT	12	0	0	590	1000	59	355,000
534 PIT	Readings unavailable for this furnace due to exhaust setup.						

Figure 7. Operating data for forging furnaces (spot-check)

SIFCO - FORGE DEPARTMENT							
FURNACE #	O2 %	COMB %	CO ppm	EXH TEMP	OPER TEMP	OUTPUT %	BTU RATING
3552 ROTARY	3.6	0	0	1700	2100	48	2,700,000
2175 PUSHER	8.9	0	0	2020	2250	8	14,640,000
2174 PUSHER	4	0	0	1900	2250	35	14,640,000
125 SMITH	8.9	0	0	1700	2200	37	Unknown
2154 PUSHER	8.8	0	6	1890	2275	59	14,640,000
5546 ROTARY	4.7	0	47	2110	2275	15	13,500,000
3556 ROTARY	4	0	0	1900	2250	35	7,680,000
3700 ROTARY	7.8	0	0	2070	2150	30	9,000,000
3553 ROTARY	6.5	0	15	1960	2200	23	12,000,000
3554 ROTARY	4	0	0	1900	2250	35	4,800,000

Figure 7. Operating data for forging furnaces (spot-check)

As mentioned earlier this data is a spot-check of furnace operations and these estimates should be considered only typical since the furnace readings were for a particular time and there is no guarantee that the furnaces operate at the same conditions all the time. However, in view of continuously variations in furnace loading and operation of the plant we have used this data to make calculations for the recommendations related to these furnaces. These readings are used in analyzing energy savings and related economics in the following sections.

V. Steam System Analysis

In 1712 Thomas Newcomen developed a practical way to convert heat into work, and the world was forever changed. Newcomen's steam engine was used to lift water from coal mines, but more globally it was the catalyst of the industrial revolution. The era of limited power from humans, horses, waterwheels and windmills would soon be displaced by an era where seemingly unlimited power could be generated from fuels. According to historian L. Rolt, "In the whole history of technology it would be difficult to find a greater single advance than this, nor one with a greater significance for all humanity." Almost three hundred years later, most electricity is still generated from steam and Sifco is still generating power for large forging hammers from steam.

Boiler Gas Use

The plant employs three 60,000 pound per hour, water-tube boilers manufactured in 1968 to provide steam to the steam forging hammers. The plant also employs two small package boilers for space heating during the four winter months. This analysis focuses on the large water-tube boilers.

According to the boiler operator, total gas consumption in 2003 of the large 60,000 lb/hr boilers was 110,437 mcf and total steam production was 112,138,000 lb. From the Natural Gas Use Summary section of the report, annual natural gas use from July 2003 to June 2004 was about 218,768 mcf. Thus, the fraction of total gas use by the boilers is about:

$$110,437 \text{ mcf/yr} / 218,768 \text{ mcf/yr} = 50\%$$

At the 2003/2004 cost of \$7.38 /mcf, the annual cost of natural gas for the boilers is about:

$$110,437 \text{ mcf} \times \$7.38 / \text{mcf} = \$815,025 / \text{yr}$$

Using 2003 gas consumption and steam production numbers, the average gas input per pound of steam produced was about:

$$110,437 \text{ mcf/yr} \times 1,000,000 \text{ Btu/mcf} / 112,138,000 \text{ lb/yr} = 985 \text{ Btu/lb}$$

The temperature of water supplied to the boiler is about 224 F and the pressure of steam supplied to the plant is about 115 psig. At 115 psig (~130 psia), the saturation temperature of the steam is about 347 F and the enthalpy of evaporation is about 874 Btu/lb. Thus, the energy required to heat the incoming water to 115 psig of saturated steam is about:

$$(347 \text{ F} - 224 \text{ F}) \times 1 \text{ Btu/lb-F} + 874 \text{ Btu/lb} = 997 \text{ Btu/lb}$$

According to operators, the temperature of exhaust gas from the boilers ranges from about 420 F at low steam demand to 610 F at high steam demand. At about 1/3 load, we measured the efficiency of the boiler to range from about 67% to 74%¹. Assuming the boiler operates

¹ The variation in efficiency readings was caused by incomplete mixing and turbulence in the exhaust gas

at the high end of this range at an efficiency of 75%, the natural gas required to generate the 115 psig of saturated steam is about:

$$997 \text{ Btu/lb} / 75\% = 1,329 \text{ Btu/lb}$$

This is about 35% higher than the average gas per pound of steam calculated using sub-metered gas and steam production data. Thus, we agree with the conclusion in the Process Heating Analysis section of the report that it is likely that either the boiler natural gas meter, the steam meter, and/or the steam pressure meter is out of calibration.

In the Boiler Energy and Mass Balance section that follows, it will be shown that measured water consumption by the boilers is less than steam output. Hence, we will assume that the steam meters are out of calibration and are overestimating steam production.

Combustion Efficiency

Combustion efficiency is the fraction of the energy in the natural gas that is imparted to the steam. Combustion efficiency can be calculated from the temperature of combustion air entering the burner, temperature of exhaust gas leaving the boiler and the quantity of unburned air in the exhaust gas. We measured the efficiency of Boiler #2 at about 3 PM on 8/5/2004 when the boiler was producing about 20,000 lb/hr of steam. The measurements fluctuated due to turbulence in the partially unmixed exhaust gasses; however, the average exhaust gas temperature was about 524 F and average percent excess air was about 216%. At these values, the boiler is about 63% efficient. Over another time interval, stack temperature varied from 580 F to 640 F, excess air varied from 80% to 190%, and the combustion efficiency varied from 67% to 74%.

As steam production increases, exhaust stack temperature increases and boiler efficiency decreases. According to operators, the temperature of exhaust gas from the boilers ranges from about 420 F during low steam demand to 610 F during high steam demand. Assuming the same quantity of excess air, boiler efficiencies at these exhaust air temperatures range from 69% to 57%. Well-tuned steam boilers are typically about 80% efficient. Thus, the boilers are operating far below peak efficiency. Assessment Recommendations B1 and B2 address cost-effective opportunities exist to increase boiler efficiency: decreasing the quantity of excess combustion air and increasing heat transfer effectiveness by cleaning the boiler tubes.

Energy and Mass Balance

The schematic of the steam delivery system is shown below. The mass flow rates represent our understanding of typical conditions, and were calculated from mass and energy balances using measured and estimated water and steam flow data. The primary input data used to create the mass flows were:

stream at the location of our combustion analysis probe

Date: 6/26/2004

Boiler operating time: 14 hours

City Water: 57,820 gal/day = 34,362 lb/hr

Total Water: 66,123 gal/day = 39,296 lb/hr

Total steam production: 609,600 lb/day = 43,453 lb/hr

Fraction blowdown = 10%

Fraction flash from blowdown = 10%

Steam to plate HX = 6,000 gal/day = 4,934 lb/hr

Steam System 1

Close inspection of Steam System 1 shows that 11,936 lb/hr of steam is returned to the deaerator tank to preheat the boiler feedwater. This represents 26% of all steam production, which seems unusually high. In addition, an energy balance on the deaerator tank using these inputs produces a deaerator water exit temperature of 424 F, which is much higher than the measured temperature of 225 F. These observations suggest that the total steam produced is too high.

The mass flow rates shown in Steam System 1 can be revised by reducing the total steam produced from 43,453 lb/day to 34,700 lb/day. Doing so produces the correct deaerator water exit temperature of 225 F, and a more reasonable estimate of the quantity of steam returned to the deaerator for preheating (7%). Thus, we believe that Steam System 2 is a better representation of typical operation. In addition, the reduced steam production is consistent with our finding in the Boiler Gas Use section that the steam meters may be out of calibration.

Steam System 2

Condition of Steam Leaving Hammers

Steam is supplied to the hammers at about 115 psig. Assuming the hammer cylinder is empty at the beginning of the filling process, an energy balance on a cylinder during the filling process yields:

$$m_2 h_s - W = m_2 u_2$$

where m_2 is the quantity of steam entering the cylinder, h_s is the enthalpy of steam entering the cylinder, W is the boundary work from raising the piston, and u_2 is the internal energy of the steam at the end of the filling period. The pressure during the filling process is

$$P = F/A + P_{\text{atm}}$$

Where F is the weight of the die, A is the area of the piston and P_{atm} is atmospheric air pressure. Thus, P is constant during the filling process and the energy balance simplifies to:

$$m_2 h_s - PV_2 = m_2 u_2$$

Specific volume, v , is volume divided by mass. Solving for mass gives:

$$m_2 = V_2 / v_2$$

Substituting the mass into the energy balance gives:

$$h_s - P v_2 - u_2 = 0$$

This equation can be solved by trial and error to determine u_2 and v_2 at a given pressure and enthalpy of the incoming steam. For example, if the pressure is 40 psia and $h_s = 1,193$ Btu/lb, then the steam leaves the cylinder in a super-heated state at about $u_2 = 1,113$ Btu/lb and $v_2 = 11.36$ ft³/lb. At this condition the steam temperature is about 320 F. As it is discharged into the steam return headers, which are open to the atmosphere, the temperature drops to about 300 F but the steam remains super heated (see figure below).

Thus, this result explains how the vast majority of steam leaving the hammers remains a vapor rather than condensing to a liquid. In Steam System 2 above, we estimate that 85% of the steam discharged from the hammers remains a vapor, even though it is returned in headers open to the atmosphere. For comparison, if the steam left the hammers as a saturated liquid at 115 psig, only about 14% would be flashed to a vapor as the pressure dropped to the atmospheric pressure.

VI. 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.

Boiler

- B1 Reduce Excess Combustion Air to Process Boilers
- B2 Mechanically Clean and Descale Tubes in Process Boilers
- B3 Install Economizers on Process Boilers
- B4 Redirect Condensate Return to Deaerator

Steam Systems

- S1 Install Additional Heat Exchanger to Reclaim Heat From Exhaust Steam
- S2 Insulate Steam Supply Piping

Furnaces and Ovens

- FO1 Preheat Combustion Air to Forging Furnaces
- FO2 Reduce Excess Air in Forging Furnaces
- FO3 Use Furnace Flue Gasses For Preheating Load and/or Dies

Compressed Air

- CA1 Reduce Compressed Air Demand
- CA2 Eliminate Compressor Cooling Water

Materials

- D1 Use High Temperature Die Materials and Lubricating Coating

Process

- P1 Convert Steam Hammers to Compressed Air Hammers

AR B1: Reduce Excess Combustion Air to Process Boilers

	Annual Savings			Project Cost	Simple Payback
	Resource	CO ₂ (lb)	Dollars		
Natural Gas	22,367 mmBtu	2,527,000	\$165,068	None	Immediate

Author: Kelly Kissock

Analysis

Boiler Gas Use

The plant employs three 60,000 pound per hour, water-tube boilers manufactured in 1968 to provide steam to the steam forging hammers. The plant also employs two small package boilers for space heating during the four winter months. This analysis focuses on the large water-tube boilers.

According to the boiler operator, total gas consumption in 2003 of the large 60,000 lb/hr boilers was 110,437 mcf and total steam production was 112,138,000 lb. From the Natural Gas Use Summary section of the report, annual natural gas use from July 2003 to June 2004 was about 218,768 mcf. Thus, the fraction of total gas use by the boilers is about:

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According to operators, the temperature of exhaust gas from the boilers ranges from about 420 F at low steam demand to 610 F at high steam demand. At about 1/3 load, we measured the efficiency of the boiler to range from about 67% to 74%². Assuming the boiler operates at an efficiency at the high end of this range of 75%, the natural gas required to generate the 115 psig of saturated steam is about:

² The variation in efficiency readings was caused by incomplete mixing and turbulence in the exhaust gas stream at the location of our combustion analysis probe

$$997 \text{ Btu/lb} / 75\% = 1,329 \text{ Btu/lb}$$

This is about 35% higher than the average gas per pound of steam calculated using sub-metered gas and steam production data. Thus, we agree with the conclusion in the Process Heating Analysis section of the report that it is likely that either the boiler natural gas meter, the steam meter, and/or the steam pressure meter is out of calibration.

Boiler Efficiency

Boiler efficiency is the fraction of the energy in the natural gas that is imparted to the steam. Boiler efficiency can be calculated from the temperature of combustion air entering the burner, temperature of exhaust gas leaving the boiler and the quantity of unburned air in the exhaust gas. We measured the efficiency of Boiler #2 at about 3 PM on 8/5/2004 when the boiler was producing about 20,000 lb/hr of steam. The measurements fluctuated due to turbulence in the partially unmixed exhaust gasses; however, the average exhaust gas temperature was about 524 F and average percent excess air was about 216%. At these values, the boiler is about 63% efficient.

As steam production increases, exhaust stack temperature increases and boiler efficiency decreases. According to operators, the temperature of exhaust gas from the boilers ranges from about 420 F during low steam demand to 610 F during high steam demand. Assuming the same quantity of excess air, boiler efficiencies at these exhaust air temperatures range from 69% to 57%. Well-tuned steam boilers are typically about 80% efficient. Thus, the boilers are operating far below peak efficiency. Two cost-effective opportunities exist to increase boiler efficiency: decreasing the quantity of excess combustion air and increasing heat transfer effectiveness by cleaning the boiler tubes. This recommendation considers decreasing the quantity of excess combustion air.

Recommendation

We recommend contacting your boiler service contractor to reduce the quantity of combustion air supplied with the natural gas, so that the fraction of excess air in the exhaust gasses is about 10%. Doing so entails a simple adjustment of the linkages that control the quantity of combustion air mixed with the natural gas. If the linkages cannot be adjusted to produce 10% excess air throughout the entire firing range, they should be adjusted to produce a minimum of 10% at either low or high fire, and then slightly more excess air at the other extreme.

If, for some reason, the linkages cannot be adequately adjusted, we recommend installing independent electronic controls on the gas supply and combustion air supply. The gas supply would be controlled by the boiler pressure. The supply of combustion air would be controlled to maintain excess air at about 10%. Independent controls are the best way to ensure peak combustion efficiency, and, because of the magnitude of savings from reducing excess air, are probably cost effective in this application.

Estimated Savings

In the analysis section of the report, we measured the combustion efficiency to be about 63% when the boiler was about 1/3 loaded. We believe this is fairly typically operation for three reasons.

First, according to operators, the temperature of exhaust gas from the boilers ranges from about 420 F during low steam demand to 610 F during high steam demand. Assuming the same quantity of excess air, boiler efficiencies at these exhaust air temperatures range from to 69% to 57%. Thus, our estimate of 63% appears to be mid range.

Second, according to boiler operators, 6/26/2004 was a typical day. During that day, two boilers generated about 609,600 lb while operating about 14 hours. Thus, the average rate of steam production by each boiler was about:

$$609,600 \text{ lb/hr} / (14 \text{ hr/day-boiler} \times 2 \text{ boilers}) = 21,771 \text{ lb/day}$$

Third, According to the boiler operator, the boilers operate about 12 hours per day. Assuming the plant operates five days per week for 49 weeks per year, the average rate of steam delivered to the steam hammers is about:

$$12 \text{ hours/day} \times 5 \text{ days/week} \times 49 \text{ weeks/year} = 2,940 \text{ hours/year}$$
$$112,138,000 \text{ lb/yr} / 2,940 \text{ hours/year} = 38,142 \text{ lb/hour}$$

If so, the boilers typically operate at about 1/3 load. For these reasons, we will estimate savings based on our measured efficiency of 63% at about 1/3 load.

Reducing the quantity of combustion air supplied to the boiler will increase the temperature of combustion and rate of heat transfer within the boiler, both of which influence the boiler efficiency. These effects have been modeled by Carpenter and Kissock (“Quantifying Energy Savings From Improved Boiler Operation”, Industrial Energy Technology Conference, New Orleans, Louisiana, May 11-12, 2005) and incorporated into the HeatSim software (Kissock and Carpenter, 2001). HeatSim is available free of charge at www.engr.dayton.edu/udiac. HeatSim output, which shows the current and target conditions, is shown below.

Based on this simulation model, we estimate that reducing excess air from about 216% to about 15% would increase boiler efficiency from about 63% to 79%, for a savings of 5,080 mmBtu/hr for each boiler.

To estimate annual gas savings, the current heat transferred to the steam can be estimated as the product of the current boiler natural gas use and the current efficiency:

$$Q_{\text{steam}} = 110,437 \text{ mcf/yr} \times 1 \text{ mmBtu/mcf} \times 63\% = 69,575 \text{ mmBtu/yr}$$

Proposed natural gas use can be estimated at the ratio of the current heat transferred to the steam and the proposed efficiency:

$$Q_{\text{ng,proposed}} = 69,575 \text{ mmBtu/yr} / (79\% \times 1 \text{ mmBtu/mcf}) = 88,070 \text{ mcf/yr}$$

Annual gas savings would be the difference between current gas use of 110,437 mcf/yr and proposed gas use:

$$110,437 \text{ mcf/yr} - 88,070 \text{ mcf/yr} = 22,367 \text{ mcf/yr}$$

$$22,367 \text{ mcf/yr} \times \$7.38 / \text{mcf} = \$165,068 / \text{yr}$$

Annual CO₂ savings would be about:

$$22,367 \text{ mcf/yr} \times 1 \text{ mmBtu/mcf} \times 113 \text{ lb CO}_2/\text{mmBtu} = 2,527,000 \text{ lb CO}_2/\text{yr}$$

Estimated Implementation Cost

The implementation cost for having boiler service contractors reduce the quantity of combustion air so that excess air is about 10% would be negligible. This should be part of their normal duties.

Estimated Simple Payback

Immediate

AR B2: Mechanically Clean and De-scale Tubes in Process Boilers

	Annual Savings			Project Cost	Simple Payback
	Resource	CO ₂ (lb)	Dollars		
Natural Gas	3,399 mmBtu	384,000	\$25,085	\$24,000	11 months

Author: Kelly Kissock

Analysis

In Assessment Recommendation B1, measurements and calculations indicate that the large 60,000 lb/hr steam boilers are currently operating at an efficiency of about 63%. Well-tuned steam boilers are typically about 80% efficient. Thus, the boilers are operating far below peak efficiency.

The primary reason for the low efficiency is excess combustion air. Assessment Recommendation B1 estimates that the efficiency of the boilers could be improved from about 63% to about 78% by simply reducing combustion air so that the exhaust gasses contain about 15% excess air. Thus, reducing the quantity of combustion air is certainly the most important mechanism for improving boiler efficiency. However, the estimated 78% efficiency would still be less than the 80% or higher efficiency that most boilers operate at. Because of the large quantity of natural gas consumed by the boilers, every incremental improvement in boiler efficiency will result in significant energy cost savings.

According to boiler operators, the water tubes are visibly scaled, despite the current practice of adding scale inhibitor to the makeup water. Moreover, the operators report that the tubes have never been mechanically cleaned in at least 15 years. Even small amounts of scale reduces the heat transfer from the combustion gasses to the water/steam, and, hence, reduces the efficiency of the boiler. Because most boilers operate at 80% or more, we estimate that mechanically de-scaling the boilers would increase the efficiency of the boilers by another 2%.

Recommendation

We recommend increasing heat transfer effectiveness by mechanically cleaning and de-scaling the boiler tubes during the two to three week summer shutdown.

Estimated Savings

In Assessment Recommendation B1, we estimated that the combustion efficiency of the boilers was typically about 63%. If so, the current heat transferred to the steam can be estimated as the product of the current boiler natural gas use and the current efficiency:

$$Q_{\text{steam}} = 110,437 \text{ mcf/yr} \times 1 \text{ mmBtu/mcf} \times 63\% = 69,575 \text{ mmBtu/yr}$$

Proposed natural gas use can be estimated at the ratio of the current heat transferred to the steam and the proposed efficiency. If the efficiency of the boilers were improved by 2%, boiler gas use would be about:

$$Q_{ng,proposed} = 69,575 \text{ mmBtu/yr} / (65\% \times 1 \text{ mmBtu/mcf}) = 107,038 \text{ mcf/yr}$$

Annual gas savings would be the difference between current gas use of 110,437 mcf/yr and proposed gas use:

$$110,437 \text{ mcf/yr} - 107,038 \text{ mcf/yr} = 3,399 \text{ mcf/yr}$$
$$3,399 \text{ mcf/yr} \times \$7.38 / \text{mcf} = \$25,085 / \text{yr}$$

Annual CO₂ savings would be about:

$$3,399 \text{ mcf/yr} \times 1 \text{ mmBtu/mcf} \times 113 \text{ lb CO}_2/\text{mmBtu} = 384,000 \text{ lb CO}_2/\text{yr}$$

Estimated Implementation Cost

According to boiler operators, mechanically de-scaling the boilers would cost about \$8,000 per boiler. If so, the total implementation cost would be about \$24,000.

Estimated Simple Payback

$$\$24,000 / \$25,085 / \text{yr} \times 12 \text{ months/year} = 11 \text{ months}$$

AR B3: Install Economizers on Process Boilers

	Annual Savings			Project Cost	Simple Payback (months)
	Resource	CO ₂ (lb)	Dollars		
Natural Gas	9,225 MM Btu	1,060,875	\$ 68,247	\$ 60,000	10.76
Electricity	-13,426 kWh	-42,359	\$ (1,315)		

Author: Arvind Thekdi

Analysis

The plant has three boilers each with a steam capacity of approximately 60,000 lbs. /hr. The steam is generated at 115 psig and temperature of 340 deg. F. The feed water is heated to 225 deg. F. It should be noted that boiling point for water at 115 psig is 347 deg. F. At this time, the exhaust gas temperature from the boiler is maintained at 550 deg. F. The boiler flue gas temperature is dependent on the steam production and correspondingly the burner firing rate. The boiler is operated by observing the exhaust gas temperature and hence the exhaust temperature becomes limiting factor for the steam production. With this operating procedure currently, each boiler can produce only about 35,000 lbs. /hr steam. Normally two boilers are used to meet the plant steam demand. The boiler steam is measured by using a steam flow meter.

A spot check on the boiler exhaust gases for process boiler #2, while boiler about 33% loaded generating 20,000 lb/hr steam, showed that the gas temperature was 524 F and oxygen content of the gases was 14.3%. This level of oxygen represents 216% excess Air and theoretical efficiency or available heat of 65.7%. At 550 deg. F. exhaust gas temperature (maximum allowed) with reduced oxygen of 5% in flue gases theoretical efficiency or available heat would be about 78%. For our calculations, we have assumed an average boiler available heat of 72.5%. When we allow for boiler radiation losses, heat used for blow-down water and other losses, it is expected that the actual boiler efficiency measured as (heat to steam delivered to header/fuel input in the burners) would be about 70%.

With steam temperature of 347 deg. F. at 115 psig pressure, total heat of steam is 1198 Btu/lb. and the feed water heat content at 225 deg. F. is 193 Btu/lb. Hence net heat required for steam generation would be $(1198 - 193 =) 1005$ Btu/lb. At assumed overall boiler efficiency of 70%, the burner heat input would be $= 1005/0.70 = 1435$ Btu/lb.

At 550 deg. F. flue gas temperature, more than 20% of the heat input is wasted in boiler exhaust gases. Two methods can be used to reduce heat lost in exhaust gases and hence increase the boiler efficiency. They are: (i) preheat combustion air and (ii) heat feed water by installing an economizer.

With 50 deg. F. exhaust gases the combustion air can be increased by about 250 to 300 deg. F. with final, air temperature reaching 300 deg. F. to 375 deg. F. This level of preheat would result in fuel savings of about 5%. However, this would require installation of a

combustion air preheater, hot sir duct and very likely a new burner and new combustion control system.

An economizer can be used to heat feed water from 225 deg. F. to about 300 deg. F. with heat content of 254 Btu/lb. This would reduce heat demand for steam by $(270 - 193) = 77$ Btu/lb. or $100 \times (77/1005)/(0.70) = 10.9\%$ of the gross heat input for steam production. Use of economizer would require minimal changes in other areas such as the piping, pumps, controls etc. The economizer is a water to gas heat exchanger and would be of much smaller size than the air preheater. Figure B2.1 shows a schematic of the system.

Figure B2.1 Schematic of the economizer and acid dew point values

Current heat input, based on 70% boiler efficiency, is
 $30,000 \text{ lbs./hr steam production} \times (1005 \text{ Btu/lb.}) \times (1.09 \text{ lbs. feed water/lb. of steam}) =$
 $33.56 \text{ Million (MM) Btu/hr.}$ Improvement of 10.9 % in heat input would be equal to $0.109 \times$
 $33.56 = 3.65 \text{ MM Btu/hr.}$

These calculations, based on overall heat balance were checked by using another approach of heat balance for the flue gases. These detail calculations are shown in Figure B2.2 below.

These two approaches indicate that use of an efficient economizer on the boilers would result in substantial savings. The notes given in Figure B2.3 give additional information related to use of economizer.

NOTE: This spreadsheet has been developed to arrive at approximate savings expected with the use of an economizer As all manufacturers arrive at a different way to arrive at expected savings please use this as a General idea of what to expect. E3M, Inc. can not be held responsible for the miss use of this spreadsheet or for the accuracy due to many variables might occur. Please fill in the areas in BLUE with your data.			
E3M, Inc. 15216 Gravenstein Way North Potomac, MD 20878 PH: 240.715.4333		DATE 25-May-05	
CUSTOMER		SIFCO Forge	
		ATTENTION: Doug Gatwood	
BOILER ECONOMIZER SAVINGS			
BOILER TOTAL SIZE		1750	BHP (approxiamtely 60,000 lbs./hr)
STEAM LOAD		50	%
STEAM LOAD		30187.5	LBS / HR
FLOW RATE OF FLUE GAS		34927	LBS/ HR (Assume 25% excess air)
BOILER PRESSURE		115	PSIG
TEMPERATURE AT OPERATING PRESSURE		347	F
EXPECTED 'STACK TEMP. WITHOUT ECONOMIZER		550	F (Measured)
REDUCE STACK TEMP FEASIBLE WITHOUT CONDENSING		300	F SAFE VALUE TO REDUCE TO = 300 F
DELTA T of Max Temp you can reduce to		250	F
FLUE GAS SPECIFIC HEAT		0.278	BTU / LB F
		GAS = .278, WATER 1.016	
% MAKE UP (or Process water. If Process water use		90	%
MAKE UP WATER TDS		330	MMHOS
FEED WATER TDS		367	MMHOS
BOILER WATER TDS		4000	MMHOS
BOILER CYCLES		10.90	CYCLES
% BLOWDOWN WATER		9.18	%
FEED WATER TO HEAT (Blowdown + Steam Load)		109	%
BTU AVAILABLE based on FEED WATER		2,650,138	BTU / HR
FACTOR FOR 1% LOSS from Heat Transfer		2,623,637	BTU / HR
FACTOR FOR 1% LOSS from Heat Transfer		661,157	Kcal / hr
FEED WATER TO HEAT		32,957	LBS / HR
FEED WATER TO HEAT		54.9	GPM IMP
FEED WATER TO HEAT		65.9	GPM US
FEED WATER TO HEAT		3,295.7	GPH IMP
SPECIFIC HEAT OF PRODUCT		1.016	BTU / LB F
FEED WATER TEMP or PROCESS WATER WITHOUT ECONOMIZER		225	F
PICK UP HEAT FROM ECONOMIZER		78	F
New Feed Water TEMP or Process Water OUT OF Economizer expected		303	F
FUEL COST		0.0074	/ CU ft. (\$7.38/MM Btu)
FUEL VALUE		1000	BTU / CU ft. - NATURAL GAS is 1000
BOILER FUEL TO STEAM EFFICIENCY (Combustion - Radiation Loss)		70	%
HEAT SAVINGS AT BOILER EFFICIENCY using Economizer (Input Gas)		3,689,028	BTU / HR
HEAT SAVINGS AT BOILER EFFICIENCY using Economizer (Input Gas)		929,636.0	Kcal / hr
FUEL SAVINGS USING ECONOMIZER		\$ 27.30	/ HR
HOURS OF OPERATION / DAY		10	HRS
SAVINGS / DAY USING ECONOMIZER		\$ 272.99	
DAYS / YEAR OPERATING		250	DAYS (5 days per week)
HRS / YEAR IN OPERATION		2,500	HRS / YEAR RUNNING
BTU / YEAR SAVED using economizer		9,222,570,155	BTU /YEAR SAVED
GAS COMPANY INCENTIVE PER CU M OF GAS SAVED		\$ -	If applicable. If not show ZERO "0"
GAS SAVINGS EXPECTED / YEAR with economizer		9,222,570	CU ft.
EXPECTED INCENTIVE FROM GAS COMPANY to add econom		\$ -	If applicable
SAVING / YEAR USING ECONOMIZER EXPECTED		\$ 68,247.02	
ECONOMIZER APPROX COST - Material Co		\$ 30,000.00	
ECONOMIZER APPROX INSTALLATION COST		\$ 15,000.00	
GAS COMPANY REBATE - If applicable		\$ -	
EQUIPMENT COST AFTER REBATE If applicable		\$ 45,000.00	
PAYBACK		0.66	Years 7.91 Months
Every manufacturer economizer will have different specifications. This calculation provides some guidelines to payback expected. When selecting the economizer it is important to determine selection of bundle material to overcome water related problems such as oxygen.			

Figure B2.2 Detail calculations for economizer performance and savings

NOTE:

Boiler/s normally operate with gas stack temperatures ranging from 300 F - 550 F. These temperatures are normally 50 F - 100F above the steam temperatures. The type of fuel, air to fuel ratio, burner characteristics, fouling of heat transfer area, atmospheric conditions and load will all have some impact on the exhaust gas temperatures. The lower the natural stack temperature the more efficient the boiler. It is normal to expect that boilers with larger heating surface area or more gas passes will have lower stack temperatures. It is important to note that environmental Government regulations for various areas restrict the limits that stack gases can exit the stack. These values depend on the fuel used.

Corrosion also can be a major factor that should be considered. The type of stack and how the condensed water is removed at the bottom of the stack should be reviewed.

Many situations have occurred where condensed stack water has destroyed the fire side of boilers due to poor design.

A rule of thumb is to maintain 250 - 300 F. in the stack temp. This is set by the type of fuel.

A major consideration on each economizer evaluation is the inlet water temperature verses the stack temperature. The smaller the delta T the less heat transfer is achievable.

Another consideration is that boilers with frequent start / stops with economizers will experience a high corrosive environment. This is due to constant cooling of condensed gases below 240 F.

Payback should always be reviewed based on all the above points.

If water to be heated is not treated corrosion will occur. Use stainless steel bundle on economizer if corrosion is possible.

Figure B2.3 Notes related to use of economizer for boilers

For the following calculations, it is assumed that two boilers are operating for 10 hours/day for 5 days/week and 50 weeks per year or 2500 hours/year. Natural gas cost is taken at \$7.50 per million Btu. Detail calculations show that annual fuel cost savings would be \$68,247/year. It should be noted that with the use of economizer the exhaust gas temperature from the economizer would drop to 300 deg. F. Since the current operations are controlled by observing the exhaust gas temperature to make sure that they do not exceed 550 deg. F., use of economizer offers a possibility of increasing steam production. Use of economizer also reduces net heat requirement in the boiler by about 11% hence even if the boiler furnace temperature was to be maintained at 550 deg. F., it is likely that the boiler steam production can be increased at least by 11% and perhaps considerably more. This is a definite benefit since in this case it is possible to operate the plant with one operating boiler only. In the following economic calculations, these benefits are not accounted for.

Recommendation

It is recommended that SIFCO consider installation of an economizer for at least two boilers. The general arrangement for the economizer installation is shown in Figure B2.1. The site inspection shows that there are severe space restrictions and it may not be necessary to use a special design for the economizer or install the economizer at a location away from the stack. There are two types of economizers that can be used. One is conventional design where the flue gases are maintained above the condensation temperature or acid dew point. As shown in Figure B2.1, for natural gas fired boilers the acid dew point temperature is 150 deg. F. and minimum allowable stack temperature is 250 deg. F. while the minimum allowable feed water temperature is 210 deg. F. The other type actually condenses boiler flue gas water content and allows much higher percentage of heat recovery. However in this case the heat exchanger has to be made from stainless steel and could be considerably more expensive than the one used for the first type. In view of relatively low number of operating hours, we recommend use of conventional noncondensing economizer that can use steel tubing.

Implementation Cost

The cost is calculated by using current installation as an example. As shown in Figure B2.4 similar installation, with higher values of steam production water inlet temperature, flue gas temperature, and most importantly very similar heat transfer rates has a total cost of \$45,000. Presumably, this is total installed cost since it is used for payback calculations.

Data without a Cain System

- Combustion source: 55,000 PPH Steam Boiler
- Heat Sink: Boiler Feedwater
- Waste Exhaust Temp: 509 °F
- Water Temp Inlet: 227°F
- Btu/hr Burner Input: 70,000 Mbtu

Performance with a Cain System

- Boiler Feedwater Flow : 118 GPM
- Final Exhaust Temp: 343°F
- Water Temp. Outlet: 276 °F
- Water Pressure Drop: 0.87 PSIG
- Exhaust Pressure Drop: 0.68 WC"
- **Btu/hr Recovered: 2,781,000**
- **Total Cost: \$45,009**

Figure B2.4 Cost of a typical economizer installation

Based on this cost and past experience we have used the economizer hardware cost of \$30,000 with installed cost of 15,000.

It may be necessary to install the economizer “inside” the stack – replace existing stack and use extension from the economizer outlet or at a location other than the stack itself. This may require additional cost for installation of an exhaust fan. The fan capacity would be 35,000 lbs./hr or 7,625 scfm or 11,150 cfm at 300 deg. F. Cost and installation of exhaust fan is estimated to be additional \$15,000. This is considered optional however, we have included this in the payback calculations.

The fan horse power, based on 3 inch w.c. pressure rise is

Gas Flow – 7,625 scfm

Pressure rise – $3/(27.7/16)$ psi = 1.73 psi

Efficiency of fan and motor system ~ 50%

Motor HP required = $7,625 \times 1.73 / (3,670 \times 0.5) = 7.2$ HP

Electricity used for 2500 hours operation = $(7.2 \times 0.746) \times 2500$ kWh = 13,426 kWh

Electricity cost – 9.8 cents/kWh

Annual electricity cost = $0.098 \times 13,426 = \$1,315$

Cost of basic economizer – installed: \$45,000

Cost of I.D. fan – installed: \$15,000

Total installed cost : \$60,000

Annual savings from use of economizer: \$68,247

Cost of electricity for blower – ID fan = \$1,315

Net cost savings = $(68,247 - 1,315) = \$66,932$ per year

No credit is taken for increased steam production due to installation of the economizer.

Simple payback period = Total installed cost / Annual savings
= \$60,000/\$66,932 = ~ 0.9 years or 11 months

Summary of Savings

Natural gas savings = 3.69 Million Btu/hr.

Number of hours/year = 2,500

Total natural gas savings = 3.69 x 2,500 = 9,225 MM Btu/year

Carbon Dioxide savings = 115 x 9,225 = 1,060,875 lbs/year

Dollar savings = \$68,247

Electricity savings: - 13,426 kWh/year

Carbon Dioxide savings = 3.155 x (-13,426) = - 42,359 lbs/year

Dollars saved = -1,315

Net cost savings = \$66,932

Cost of installation = \$60,000

Payback period = 11 months.

AR B4: Redirect Condensate Return to Deaerator

	Annual Savings			Project	Simple
	Resource	CO ₂ (lb)	Dollars	Cost	Payback (months)
Natural Gas	1,334 MM Btu	153,410	\$ 9,921	\$ 13,600	15.71
Electricity	-3,371 kWh	-10,636	\$ (362)		
Water	1,652,285 Gallons	0	\$ 826		

Author: Arvind Thekdi

Analysis

Condensate from a small percentage of the steam used in the plant is returned to the boiler house and is collected in a tank. The same tank collects makeup water from the city and cooling water from the air compressors. The mixed water is treated and used as boiler feed water for the boiler. At this time it is not possible to get exact value of the returned condensate. We see a pipe of 1 ½ inch diameter that carries condensate return to the tank. In this case, we estimated condensate water flow using conventional methods. The method of estimating the condensate is shown in the following table (Figure B4.1). Based on these assumptions, it is estimated that about 5,500 lbs./hr. of condensate is returned. This represents about 15% of the total steam production.

Pipe size	inch	1.5
Water velocity	ft/sec	2
C/S area of pipe	ft^2	0.0123
Volume flow rate	cu.ft./hr	88.36
Flow rate	gph	660.91
	lbs/hr	5,513.51
No. of hours operation	hrs/yr.	2500
Total gallons/year	gpy	1,652,285
Temperature change	deg. F.	80
Heat for raising temperature	MM Btu/year	1,103

Figure B4.1 Estimate of returned condensate flow

The plant steam is distributed at 115 psig with condensation temperature of about 330 deg. F. At this time, the condensate return line is not properly insulated and there could be substantial drop in condensate temperature. Measurement of the return water temperature shows that it is at about 160 deg. F. When this water is mixed with other water in the tank, its temperature drops and by the time it reaches deaerator, it is practically at ambient temperature. Hence we are losing (150-70) = 80 deg. F. temperature for the condensate water. The condensate returns at least at 150 deg. F. If this water is piped directly to the deaerator tank, then it would be unnecessary to treat this water and also we can save a small amount of heat used for heating water in the deaerator. The heat used in the deaerator is supplied from the steam. Estimated heat savings is 1,103 million (MM) Btu/year.

It would be necessary to install a small pump to raise the condensate pressure to approximately 150 psig before it enters the deaerator. It is assumed (as an extreme case) that 150 psig pressure may be required. The total volume of water is 660 gph or 11 gpm. Theoretical horse power for pump = $(11 \times 150)/1747 = 0.95$ HP. Assuming pump efficiency of 50%, actual motor HP would be 2 HP.

Recommendation

It is recommended that the condensate return line insulation be repaired and redone as necessary to retain as much heat as possible for the condensate and the return condensate be directly taken to the deaerator.

Savings.

The savings are calculated in two areas. One is savings in chemicals used for water treatment and other in steam (and its heat) savings by reducing heat demand in the deaerator.

Following table (Figure B4.2) shows the calculations and results.

Savings - Water Treatment		
Cost of gas	\$/MM Btu	\$ 7.38
Condensate flow rate	gph	660.91
	lbs/hr	5,513.51
No. of hours operation	hrs/yr.	2500
Total gallons/year	gpy	1,652,285
Cost of water treatment	\$/1000 gallons	\$ 0.50
Cost savings/year	\$/year	\$ 826.14
Savings - Cost of water reheating		
Temperature change	deg. F.	80
Heat for raising temperature	MM Btu/year	1,103
Cost of steam	\$/MM Btu	\$ 9.00
Cost of heating water	\$/year	\$ 9,921
Cost of Electricity for Pump		
Pump HP required	HP	2.00
Electricity cost	\$/kWh	\$ 0.097
Cost of Electricity for Pump	\$/year	\$ 362
Total savings	\$/year	\$ 10,385

Figure B4.2 Calculations for savings for repiping of return condensate

Implementation Cost

Cost of implementing this recommendation includes (i) cost of repiping the condensate; (ii) cost of pump and controls and (iii) cost of insulating the condensate return pipes.

Total cost estimate is shown in the following table (Figure B4.3) and it is \$13,600.

Cost		
Pipes and pump hardware		\$ 6,000.00
Insulation		\$ 2,000.00
Installation labor		
Man-hours	hrs	80
Rate	\$/man-hour	\$ 70.00
Total labor	\$	\$ 5,600.00
Total cost	\$	\$ 13,600.00

Figure B4. Cost estimate for implementation of the recommendation

Please note that this includes small (2 HP) pump for the condensate. It may not be necessary to have this pump.

Payback Calculations

The cost of implementation is \$13,600.

Net savings for the project is \$10,385

Simple payback period = $\$13,600 / \$10,385 = 1.31$ years or 15.7 months.

Savings

Natural gas savings = 1,334 MM Btu/year

Carbon Dioxide reduction = 153,410 lbs./year

Dollar savings = \$9,921

Electricity use = - 3,371 kWh

Carbon Dioxide reduction = -10,636 lbs./year

Dollar savings = -362 per year

AR S1: Install Additional Heat Exchanger to Reclaim Heat From Exhaust Steam

	Annual Savings			Project Cost	Simple Payback
	Resource	CO ₂ (lb)	Dollars		
Natural Gas	9,722 mmBtu	1,099,000	\$71,748	\$10,000	2 months

Author: Kelly Kissock

Analysis

The forging plant employs nine active steam-powered forging hammers. The 115 psig steam is supplied by three 60,000 pound per hour, water-tube boilers. The cylinders in the steam hammers are lubricated with graphite oil. After raising the pistons, the contaminated steam is exhausted into steam headers with diameters from about 6 to 14 inches. The steam headers are attached to six vertical vent pipes, which are open to the atmosphere; thus, much steam is flashed to atmospheric pressure in the headers and exhausted to the atmosphere as a gas through the vent pipes. Clouds of steam can be observed leaving the exhaust vents when the hammers are operating.

In the Steam System Analysis section of the report we estimated that during typical operation, about 23,211 lb/hr of steam leaves through the vents. This represents about 72% of the steam supplied to the hammers. About 4,160 lb/hr of steam is passed through a plate heat exchanger that preheats boiler feed water from about 90 F to about 150 F, before it enters the deaerator.

The deaerator tank operates at about 6 psig (~20 psia) at a temperature of about 225 F. The saturation temperature at 20 psia is about 228 F; thus it is not possible to raise the temperature of the water in the deaerator tank much above the current temperatures.

Recommendation

We recommend adding a second plate heat exchanger after the current plate heat exchanger to heat the boiler feed water from about 150 F to about 220 F. The feed water would be heated with steam from the return steam headers. To ensure sufficient pressure in the return headers to force steam through the plate heat exchanger, it may be necessary to close one or more vertical vent pipes. However, we don't think that this would raise the pressure in the steam headers to a high enough back pressure to slow down the operation of the hammers.

Estimated Savings

In Steam System 2, we estimate that during typical operation about 35,428 lb/hr of water to the deaerator tank leaves the current plate heat exchanger at about 150 F. The proposed plate heat exchanger would heat the water from 150 F up to about 220 F. If so, the total heat added to water would be about:

$$35,428 \text{ lb/hr} \times 1 \text{ Btu/lb-F} \times (220 \text{ F} - 150 \text{ F}) = 2,480,000 \text{ Btu/hr}$$

This would dramatically reduce the need for steam from the boilers to the deaerator tank. We assume that steam from the steam headers would enter the plate heat exchanger as a

saturated vapor at 5 psig (~20 psia). If so, the temperature would be about 228 F and enthalpy of evaporation would be about 960 Btu/lb-F. We believe that this is a conservative assumption since the steam could actually enter the heat exchanger as a super heated vapor at about 300 F, as was shown in the Steam System Analysis section of the report.

If so, the quantity of steam condensed in the heat exchanger would be about:

$$2,480,000 \text{ Btu/hr} / 960 \text{ Btu/lb} = 2,583 \text{ lb/yr}$$

In Steam System 2, we estimate that during typical operation, about 23,211 lb/hr of steam is currently exhausted to atmosphere. Thus, this is well within the quantity of steam available.

During 2003/2004, the cost of natural gas was about \$7.38 /mcf. Conservatively assuming the boilers are 75% efficient, the annual savings would be about:

$$2,480,000 \text{ Btu/hr} / 75\% \times 2,940 \text{ hours/year} = 9,722 \text{ mmBtu/yr}$$
$$9,722 \text{ mmBtu/yr} \times 1 \text{ mmBtu/mcf} \times \$7.38 / \text{mcf} = \$71,748 / \text{yr}$$

Annual CO₂ savings would be about:

$$9,722 \text{ mcf/yr} \times 1 \text{ mmBtu/mcf} \times 113 \text{ lb CO}_2/\text{mmBtu} = 1,099,000 \text{ lb CO}_2/\text{yr}$$

Estimated Implementation Cost

At the conservative temperatures assumed above, the log mean temperature difference, DTLM, would be about:

$$DT1 = 228 \text{ F} - 220 \text{ F} = 8 \text{ F}$$

$$DT2 = 228 \text{ F} - 150 \text{ F} = 78 \text{ F}$$

$$DTLM = (DT1 - DT2) / \ln(DT1 / DT2) = (8 \text{ F} - 78 \text{ F}) / \ln(8 \text{ F} / 78 \text{ F}) = 30.7 \text{ F}$$

The overall heat transfer coefficient, U, for condensers is about 666 Btu/hr-ft²-F. If so, the required heat transfer area would be about:

$$A = Q / (U \text{ DTLM}) = 2,480,000 \text{ Btu/hr} / (666 \text{ Btu/hr-ft}^2\text{-F} \times 30.7 \text{ F}) = 121 \text{ ft}^2$$

According to a heat exchanger vendor, the cost of a plate heat exchanger designed for a 10 psig with a 120 ft² of surface area would be about \$5,000. We estimate that the cost of installation would be about equal to the cost of the heat exchanger. If so, the total implementation cost would be about \$10,000.

Estimated Simple Payback

$$(\$10,000 / \$71,748 / \text{year}) \times 12 \text{ months/year} = 2 \text{ months}$$

AR S2: Insulate Steam Supply Piping

	Annual Savings			Project Cost	Simple Payback
	Resource	CO ₂ (lb)	Dollars		
Natural Gas	602 mmBtu/yr	68,000	\$5,923	\$2,880	6 months

Author: Kelly Kissock

Analysis

We identified about 170 ft of uninsulated steam header pipe. Adding insulation would reduce heat loss and the natural gas consumption of the boilers.

Recommendation

We recommend insulating all steam header and condensate return piping with two-inches of professionally installed xxx insulation.

Estimated Savings

We identified 65 ft of 4-inch header, 85 ft of 6-inch header, and 20 ft of 8-inch header which is uninsulated. The header pipes carry 115 psig steam. The cost of natural gas in 2003/2004 was about \$7.38 /mcf. In Assessment Recommendation B1, we estimated that the efficiency of the boilers was 75% of less. Thus, the cost of thermal energy from the boilers is about:

$$\$7.38 / \text{mcf} \times 1 \text{ mmBtu/mcf} / 75\% = \$9.84 / \text{mmBtu}$$

According to boiler operators, the boilers run about 12 hours per day. Assuming five day per week operation for 49 weeks per year, annual operating hours are about:

$$12 \text{ hours/day} \times 5 \text{ days/week} \times 49 \text{ weeks/yr} = 2,940 \text{ hours/year}$$

According to insulation contractors, the conductivity of insulation is about 0.3 Btu-in/hr-ft²-F. We recommend adding two-inch thick insulation.

Heat loss through insulated and uninsulated heat pipes is calculated by the HeatSim (Kissock and Carpenter, 2001) software which is available free of charge at www.engr.udayton/udiac. HeatSim solves energy balances to estimate heat loss from insulated and uninsulated pipes from convection and radiation. HeatSim output screens showing current heat loss, heat loss if the pipes were insulated, and the savings from insulating the pipes are shown below.

The HeatSim results indicate the total savings would be about:

$$256 \text{ mmBtu/yr} + 264 \text{ mmBtu/yr} + 82 \text{ mmBtu/yr} = 602 \text{ mmBtu/yr}$$

$$\$2,519 \text{ /yr} + \$2,598 \text{ /yr} + \$806 \text{ /yr} = \$ 5,923 \text{ /yr}$$

The reduction in CO₂ emissions would be about:

$$602 \text{ mmBtu/yr} \times 113 \text{ lb CO}_2\text{/mmBtu} = 68,000 \text{ lb CO}_2\text{/yr}$$

Estimated Implementation Cost

According to insulation contractors, the installed cost of insulation is about \$10 per square foot. If so, the total implementation cost would be about:

$$\$851 + \$1,558 + \$471 = \$2,880$$

Estimated Simple Payback

$$(\$2,880 / \$5,923 \text{ /year}) \times 12 \text{ months/year} = 6 \text{ months}$$

AR FO1: Preheat Combustion Air to Forging Furnaces

	Annual Savings			Project	Simple
	Resource	CO ₂ (lb)	Dollars	Cost	Payback (months)
Natural Gas	41,527 MM Btu	4,775,605	\$ 306,469	\$ 664,725	26.03
Electricity	0 kWh	0	\$ -		

Author: Arvind Thekdi

Analysis

SIFCO forging plant has a number of furnaces for reheating steel prior to forging. They consist of rotary furnaces, box furnaces, pusher furnaces and induction furnaces. As discussed in previous section these furnaces vary in size, capacity (lbs. of steel heated) and heat input. The following table (Figure FO1.1) gives list of major furnaces, their burner Btu rating and operating parameters during the production when the data was collected.

FURNACE #	O ₂ %	COMB %	CO ppm	EXH TEMP	OPER TEMP	OUTPUT %	BTU RATING
3552 ROTARY	3.6	0	0	1700	2100	48	2,700,000
2175 PUSHER	8.9	0	0	2020	2250	8	14,640,000
2174 PUSHER	4	0	0	1900	2250	35	14,640,000
125 SMITH	8.9	0	0	1700	2200	37	Unknown
2154 PUSHER	8.8	0	6	1890	2275	59	14,640,000
5546 ROTARY	4.7	0	47	2110	2275	15	13,500,000
3556 ROTARY	4	0	0	1900	2250	35	7,680,000
3700 ROTARY	7.8	0	0	2070	2150	30	9,000,000
3553 ROTARY	6.5	0	15	1960	2200	23	12,000,000
3554 ROTARY	4	0	0	1900	2250	35	4,800,000

NOTE: Readings were taken with furnaces in production.

Figure FO1.1 List of major forging furnaces and their ratings

Flue gases from these furnaces are discharged at very high temperature, in the range of 1900 deg. F. to 2400 deg. F. and represent anywhere from 50% to 75% of the total heat input in the furnace. It is possible to recover a large percentage of the heat being lost in flue gases to preheat combustion air going to the burners used on these furnaces. A general arrangement for most commonly used combustion air preheating system is shown in Figure FO1.2. In this system a recuperator is installed in the flue gas passages. Combustion air for the burners flows through the recuperator and picks up part of the flue gas heat so that the combustion air temperature rises from the inlet temperature. In most cases, the recuperator is designed to recover 30% to 60% of the flue gas heat. Trade off between capital cost of the recuperator system and savings in fuel cost determine the exact design and percentage of flue gas heat recovery.

Recently another type of combustion air preheating system known as regenerative burners, shown in Figure FO1.3, is highly efficient and offers up to 75% heat recovery. Detail description of this system is given in Reference 1. This system is relatively expensive and could present maintenance problems when used on very high temperature exhaust gases with

particulates such as iron oxide or scale. For this recommendation, we have not considered this system.

Figure FO1.2 Commonly used recuperative air preheating system

Figure FO1.3 Regenerative burners for combustion air reheating

For this analysis, we have used furnace performance data given in Figure FO1.1. In each case, it is assumed that the combustion air preheater would be able to transfer 45% to 50% of the flue gas heat to combustion air. For example, furnace temperature for furnace 3552 rotary furnace the flue gas temperature is 2100 deg. F. The combustion air temperature is estimated to be at 45% of the furnace temperature or $0.45 \times 2100 = 945$ deg. F. This is strictly arbitrary for preliminary calculations and it is based on the author's experience in design and application of recuperators to high temperature furnaces. The combustion air preheat temperature and proper air-fuel ratio (2% oxygen in flue gases) is used to calculate percentage fuel savings and corresponding dollar savings for an annual 2500 hours operation at the set conditions,. Once again, we have used the data collected at one operating point hence the results could differ if other operating conditions are used. The results are shown in Figure FO1.4.

FURNACE #	Combustion air temperature (F)	Savings - comb. Air preheating	Savings \$/yr	Estimated recup. Cost \$	Installation cost incl. Regen Burners etc.	Installation cost incl. Burners etc.	Payback - months
3552 ROTARY	945	39.3%	\$ 11,268.62	\$ 15,238	\$ 26,666	\$ 41,904	45
2175 PUSHER	909	69.0%	\$ 17,886.90	\$ 15,057	\$ 26,350	\$ 41,408	28
2174 PUSHER	855	34.8%	\$ 39,440.40	\$ 33,765	\$ 59,089	\$ 92,854	28
125 SMITH				\$ -	\$ -	\$ -	
2154 PUSHER	851	61.0%	\$ 116,721.00	\$ 72,765	\$ 127,339	\$ 200,104	21
5546 ROTARY	950	45.0%	\$ 20,189.96	\$ 18,131	\$ 31,730	\$ 49,861	30
3556 ROTARY	855	34.8%	\$ 20,690.05	\$ 21,495	\$ 37,617	\$ 59,112	34
3700 ROTARY	932	62.6%	\$ 37,431.01	\$ 25,478	\$ 44,587	\$ 70,065	22
3553 ROTARY	882	48.9%	\$ 29,908.72	\$ 24,319	\$ 42,558	\$ 66,877	27
3554 ROTARY	855	34.8%	\$ 12,931.28	\$ 15,469	\$ 27,070	\$ 42,539	39
			\$ 306,468	\$ 241,718	\$ 423,007	\$ 664,725	26
NOTE: Readings were taken with furnaces in production.							

Figure FO1.4 Analysis for use of preheated air – savings and cost

It should be noted that use of preheated air would require change of preheated air burners, new piping for hot air and associated control system for the furnace.

Recommendation

The plant may consider use of preheated air by installing a recuperator. Further analysis shown below shows at the current operating condition of the furnace and 3000 hours annual operation, economics of use of preheated air are not as attractive as some other options. However, we have included this recommendation to give information on this recommendation so, if the conditions change, the plant can use this analysis for further considerations.

Implementation Cost

The implantation cost includes cost of the recuperator, required hardware and cost of installation. The cost data is based on recent quotation from a recuperator supplier for similar application. Installation cost is assumed 175 % of the hardware cost. For example in case of # 3552 rotary furnace, flue gas volume is 17,000 scfh or 283 scfm. The quotation for 1333 scfm (80,000 scfh) is \$30,000. It is assumed that the cost of higher temperature recuperator and cost increases during the last year would increase the cost by 50% and the cost would be affected by 0.7 power of the flue gas volume ratio.

Hence cost for the current application $= 1.5 \times \$30,000 \times (281/1333)^{0.7}$
 $= \$15,238$

Installation cost including burners etc. $= 1.75 \times \$15,238 = \$26,667$

Total project cost $= \$15,238 + \$26,666 = \$41,904$

Please note that this cost is on the low side. Actual cost may be higher due to cost associated with modifications to the furnace, removal of burners etc.

Similar calculations are carried out for other furnaces. Figure FO1.4 shows cost for all furnaces.

Payback

The payback period is calculated using the above mentioned savings in fuel cost and estimated implementation cost for each of the furnaces.

For 3552 rotary furnace

Fuel savings = \$11,268

Implementation cost = \$41,904

Payback period = \$41,094/\$11,268 = 3.7 year or 45 months.

Similar calculations are carried out for other furnaces. The payback period varies from 21 months to 45 months.

Summary of Savings

Natural Gas

A list of energy savings in terms of MM Btu/hr. for natural gas and dollars per year for the forging furnaces are given in Figure FO1.5. The total energy saving fro all furnaces accounted for here is 41,527 MM Btu/year.

FURNACE #	Energy savings MM Btu/year	Savings \$/yr
3552 ROTARY	1,527	\$ 11,269
2175 PUSHER	2,424	\$ 17,887
2174 PUSHER	5,344	\$ 39,440
2154 PUSHER	15,816	\$ 116,721
5546 ROTARY	2,736	\$ 20,190
3556 ROTARY	2,804	\$ 20,690
3700 ROTARY	5,072	\$ 37,431
3553 ROTARY	4,053	\$ 29,909
3554 ROTARY	1,752	\$ 12,931
	41,527	\$ 306,468

Figure FO1.5 Summary of energy and Dollar savings with use of preheated combustion air

Cost savings for all furnaces considered.

Natural gas energy savings – 41,527 MM Btu/year

Annual savings of \$306,468.

Carbon Dioxide savings – 4,775,605 lbs./year

Electricity.

Reduction or increase in electric cost is negligible due to several pluses and minuses for use of electricity.

AR FO2: Reduce Excess Air in Forging Furnaces

	Annual Savings			Project	Simple
	Resource	CO ₂ (lb)	Dollars	Cost	Payback (months)
Natural Gas	10,194 MM Btu	1,172,310	\$ 75,232	\$ 28,000	4.47
Electricity	0 kWh	0	\$ -		

Author: Arvind Thekdi

Analysis

SIFCO forging plant has a number of furnaces for reheating steel prior to forging. They consist of rotary furnaces, box furnaces, pusher furnaces and induction furnaces. As discussed in previous section these furnaces vary in size, capacity (lbs. of steel heated) and heat input. The following table (Figure FO2.1) gives list of major furnaces, their burner Btu rating and operating parameters during the production when the data was collected.

FURNACE #	O2 %	COMB %	CO ppm	EXH TEMP	OPER TEMP	OUTPUT %	BTU RATING
3552 ROTARY	3.6	0	0	1700	2100	48	2,700,000
2175 PUSHER	8.9	0	0	2020	2250	8	14,640,000
2174 PUSHER	4	0	0	1900	2250	35	14,640,000
125 SMITH	8.9	0	0	1700	2200	37	Unknown
2154 PUSHER	8.8	0	6	1890	2275	59	14,640,000
5546 ROTARY	4.7	0	47	2110	2275	15	13,500,000
3556 ROTARY	4	0	0	1900	2250	35	7,680,000
3700 ROTARY	7.8	0	0	2070	2150	30	9,000,000
3553 ROTARY	6.5	0	15	1960	2200	23	12,000,000
3554 ROTARY	4	0	0	1900	2250	35	4,800,000

NOTE: Readings were taken with furnaces in production.

Figure FO2.1 List of major forging furnaces and their ratings

Flue gases from these furnaces are discharged at very high temperature, in the range of 1900 deg. F. to 2400 deg. F. and represent anywhere from 50% to 75% of the total heat input in the furnace under the optimized operating conditions with minimum amount of excess air for the burners. A survey of oxygen (O₂) content of flue gases from these furnaces was carried out by Conrad Kacsik. The results, shown in Figure FO2.1 indicate that flue gases from these furnaces contain 3.6% to 8.9 % O₂ indicating the gases include large amount of excess air. The level of excess air is a function of oxygen in flue gases. Figure FO2.2 shows relationship between percentage O₂ in flue gases vs. excess air contained in the flue gases. For example at 4% O₂ in flue gases the gases contain about 20% more air than necessary for combustion of natural gas used in the burners. The excess air could be result of use of excess air in the burners or air entering the furnace through various openings in the furnace due to negative pressure inside the furnace. In either case, the excess air represents significant loss of heat from the furnace and costs money. Cold air entering the furnace could result in poor temperature distribution in the furnace and may require longer heating time to achieve the required temperature uniformity for the load. This may represent reduction in productivity of the furnace.

Figure FO2.2 Relationship between oxygen vs. excess air in flue gases

Presence of this excess air at high temperature represents significant energy loss from the furnace, particularly at higher flue gas temperatures. Figure FO2.3 shows loss of heat in flue gases at various temperatures and excess air in flue gases for natural gas. For example, at 2000 deg. F. flue gas temperature and 4% 20% excess air (4% O₂), 64% of the total heat input is the lost.. If the excess air is reduced to 10% (2% O₂) in flue gases, the heat loss drops to 58%, resulting in fuel saving of 13% from the current fuel use.

Figure FO2.3 Exhaust (flue) gas heat loss vs. exhaust gas temperature and excess air

Following steps are necessary to operate the furnace at “optimum” condition where the flue gases contain about 2% O₂.

- (i) Adjust air-fuel ratio for the burners, commonly known as “tuning the burner” step.
- (ii) Seal the furnace by closing unnecessary openings and minimize the time when furnace door is open for loading or unloading of the furnace

- (iii) Control furnace pressure by using proper damper and/or pressure control system to maintain slightly (0.01” to 0.025” water column) pressure in the furnace.

Measurement of furnace pressure at a few locations within the furnace indicates that the furnaces are operating at negative pressure and there are a number of cracks where ambient air leaks into the furnace. All of this results in presence of high excess air in flue gases.

We have measured oxygen level in flue gases and fuel input rating by using percentage output for the valve motor during “spot-check” of the furnace operations. This information and the burner ratings (Btu/hr. burner firing capacity) are used to calculate possible savings if the flue gas oxygen is adjusted to 2% and the furnace is operated at the “spot-check” condition for 2500 hours/year. We recognize that the furnace could be operating at much higher fuel input or lower input compared to values used here however, we have attempted to illustrate economic benefits under these conditions to indicate benefits of O₂ control in flue gases. .

These savings, given in Figure FO2.4 below indicate that it is possible to save as much as 20,388 MM Btu/year. At \$7.38 per MM Btu, this represents annual savings of \$150,462 per year.

FURNACE #	O ₂ %	EXH TEMP	OPER TEMP	OUTPUT %	BTU RATING	% xs air	savings with 2% O ₂	Hrs./year	Gas cost \$/MM Btu	Energy savings MM Btu/year
3552 ROTARY	3.6	1700	2100	48	2,700,000	19.44	10%	2500	\$7.38	330
2175 PUSHER	8.9	2020	2250	8	14,640,000	68.59	56%	2500	\$ 7.38	1,625
2174 PUSHER	4	1900	2250	35	14,640,000	22.15	10%	2500	\$ 7.38	1,313
125 SMITH	8.9	1700	2200	37	Unknown					
2154 PUSHER	8.8	1890	2275	59	14,640,000	67.25	47%	2500	\$ 7.38	10,056
5546 ROTARY	4.7	2110	2275	15	13,500,000	27.15	18%	2500	\$ 7.38	930
3556 ROTARY	4	1900	2250	35	7,680,000	22.15	10%	2500	\$ 7.38	689
3700 ROTARY	7.8	2070	2150	30	9,000,000	55.03	45%	2500	\$ 7.38	3,061
3553 ROTARY	6.5	1960	2200	23	12,000,000	41.90	28%	2500	\$ 7.38	1,954
3554 ROTARY	4	1900	2250	35	4,800,000	22.15	10%	2500	\$ 7.38	431
										20,388

Figure FO2.4 Calculations for savings resulting from control of excess air

It is recognized that it may not be possible to eliminate all the air entering the furnace due to need for opening doors or difficulty in plugging openings or cracks. Hence, it is assumed that it is possible to save 50% of the calculated energy savings.

Recommendation

The plant should initiate a program to adjust air-fuel ratio for burners on all forging furnaces, inspect furnace casing, doors and other areas to find openings, cracks etc. that can be sealed. It is also recommended that the furnace stack be properly blocked to maintain slightly positive pressure in the furnace at normal operating conditions. It is recognized that firing rate and hence flue gas volume going through the stack changes resulting in changes in the draft (negative pressure) in the furnace however the plant should try to adjust this pressure at the average firing rate or at the lower firing rate. The furnace pressure would increase slightly as the firing rate is increased and the adjustment should be carefully watched in the beginning to reach a compromise between negative pressure in the furnace at low firing rate vs. excessive pressure in the furnace at high firing. Ideally, the furnaces should be equipped

with a pressure controller but considering the number of operating hours and cost of installation of such a system, we suggest using the above mentioned approach.

Savings

As mentioned above potential energy savings is 20,388 MM Btu/year or \$150,463 at current gas price of \$7.38 per MM Btu. There would not be any change in use of electricity or any other utility.

Implementation Cost

Implementation of this recommendation requires following actions.

- (i) Measure furnace pressure at normal operating condition or high-low fire conditions
- (ii) Inspect openings, cracks, seals etc. to find areas where air can leak into the furnace
- (iii) Measure furnace flue gas oxygen at normal operating condition or high-low fire conditions
- (iv) Seal openings, cracks, repair seals and train operators to keep the furnace door opening time to the minimum required
- (v) Adjust furnace pressure to achieve slightly positive (0.01 to 0.025 inch w.c.) under low firing condition. This may require installation of a damper on flue gas stack or simply installing a restriction that can be adjusted as necessary.
- (vi) Adjust air-fuel ratio to allow 10% excess air in the burners at normal operating condition. This may require taking flow readings for air and gas if flow meters are installed or use of burner calibration graphs. Simply observing flame color is not a proper indication of air-fuel ratio for the combustion.

Estimated cost and payback period for this work is given in a table (Figure) FO2.5

It is assumed that two days (16 hours) would be required for each furnace for performing above mentioned work. A rate of \$100 per hour is assumed for in-house or contract labor for the work. Material cost is assumed to be \$12,000. These numbers are conservative and on the “high” side.

Payback period for excess air reduction - Forge furnaces		
Annual estimated savings	\$/year	\$ 150,463
Practical savings due to difficulty in fixing leaks		
Assume only 50% of calculated		\$ 75,231
Annual cost (adjustment, tuning, fixing openings etc.)		
No. of man-hours (16 hours/furnace)		160
Cost/man-hour (average contractor, in-house)	\$	\$ 100
Total labor cost		\$ 16,000
Other cost (materials etc.)	\$	\$ 12,000
Total cost	\$/year	\$ 28,000
Simple payback	Months	\$ 4.47

Figure FO2.5 Estimated cost and payback period for the recommendation FO2

Payback Period

As mentioned in Figure FO2.5, that payback period is about 4 ½ months.

Summary of Savings

Natural gas savings = 10,194 MM Btu/year

Dollar savings = \$75,232 per year.

Carbon Dioxide savings = 1,172,310 lbs/year

Electricity savings = None

AR FO3: Use Furnace Flue Gases for Preheating Load and/or Dies

	Annual Savings			Project	Simple
	Resource	CO ₂ (lb)	Dollars	Cost	Payback (months)
Natural Gas	41,240 MM Btu	4,742,600	\$ 304,351	\$ 265,106	10.71
Electricity	-74,592 kWh	-235,338	\$ (7,310)		

Author: Arvind Thekdi

Analysis

SIFCO forging plant uses a large number of forging and heat treating furnaces. The forging furnaces are the largest user of energy after the boilers. They use more than 30% of the total natural gas used by the plant. The rotary furnaces, box furnaces and pusher furnaces are operated at very high temperature and as mentioned in other recommendations (FO1 and FO2) discharge flue gases that contain, in most cases, more than 50% of the heat input for the furnace.

Several possible heat recovery systems can be used to recover and use heat from the flue gases. The methods include:

- (i) Combustion air preheating
- (ii) Load or charge preheating
- (iii) Steam generation or hot water heating
- (iv) Use of gases in lower temperature process furnaces

Methods (i) and (ii) use the flue gas heat within the furnace or process itself and are preferred method of heat recovery if possible. Methods (iii) and (iv) require matching of schedule for the furnaces and the “host” processes and is not as preferable as the other two methods. Economics of air preheating are discussed in recommendation FO1. This recommendation deals with the second method in which furnace flue gases are used to preheat load charged in the furnace or die used for the presses/hammers.

As mentioned in other recommendations (FO1 and FO2), the furnace operating parameters were measured and used for calculating heat input, excess air and flue gas temperature for all large forging furnaces. These calculations are further expanded to estimate total flue gas volume and amount of heat that can be recovered from these gases when all furnaces are operating. Figure FO3.1 shows these calculations.

Total volume of flue gases from these furnaces is estimated to be 422,638 scfh. Total heat content, calculated by two different methods varies from 18.23 MM Btu/hr. to 19.57 MM Btu/hr. For these calculations, we have used the lower figure of 18.23 MM Btu/hr. Average temperature of these gases is 1800 deg. F.

SIFCO - FORGE										
FURNACE #	EXH TEMP	OPER TEMP	OUTPUT %	BTU RATING	% xs air	Av. Heat	net heat in furnace	Heat in ex. Gases (MM Btu/hr)	Flue gas volume scfh	Heat in flue gases on vol basis MM Btu/hr
3552 ROTARY	1700	2100	48	2,700,000	19.44	34.72	0.45	0.85	17,027.21	0.61
2175 PUSHER	2020	2250	8	14,640,000	68.59	18.16	0.21	0.96	21,720.12	0.94
2174 PUSHER	1900	2250	35	14,640,000	22.15	39.53	2.03	3.10	68,847.64	2.79
125 SMITH	1700	2200	37	Unknown						-
2154 PUSHER	1890	2275	59	14,640,000	67.25	23.68	2.05	6.59	158,905.70	6.40
5546 ROTARY	2110	2275	15	13,500,000	27.15	31.34	0.63	1.39	28,322.14	1.28
3556 ROTARY	1900	2250	35	7,680,000	22.15	39.53	1.06	1.63	36,116.80	1.46
3700 ROTARY	2070	2150	30	9,000,000	55.03	21.57	0.58	2.12	46,044.56	2.04
3553 ROTARY	1960	2200	23	12,000,000	41.90	30.41	0.84	1.92	43,081.34	1.80
3554 ROTARY	1900	2250	35	4,800,000	22.15	39.53	0.66	1.02	22,573.00	0.91
							8.52	19.57	442,638.52	18.23

Figure FO3.1 Calculations for heat in flue gases form major forging furnaces

Heat from the flue gases can be used to preheat load that is heated in the forging furnaces. For example forging furnaces are used to heat material from ambient temperature to as high as 2000 deg. F. in the furnaces. It is possible to preheat the material to a reasonable temperature (from 400 deg. F. to 700 deg. F.) prior to charging it into the furnace. This would recover a substantial percentage of the waste heat of flue gases. Two possible systems often used by the industry are shown and described below. In each case the higher temperature flue gases are mixed with ambient air to drop temperature to manageable temperature of 1200 deg. F.

The exhaust gases can be handled in two ways for heat recovery. In one way, shown in Figure FO3.2, a high temperature (1200 deg. F.) fan is used to pressurize gases and send to the heater box. In this case the entire system including the heater box is operating under pressure and it is likely that hot gases would leak out of the box through openings. This system requires high temperature fan and higher volume of gases making the fan considerably more expensive than the lower temperature “standard” fan.

Figure FO3.2 Arrangement for handling of furnace exhaust gases for load preheating

Another way is to maintain the heater box and the system under slight suction and uses a lower temperature fan. Figure FO3.3 shows an arrangement where an exhaust (I.D.) fan is used for transport of flue gases through the heater box.

Figure FO3.3 Alternate arrangement for handling of furnace exhaust gases for load preheating

The following analysis is carried out for the first system in which a two forced draft fans are used to transfer tempered flue gases at 1200 deg. F. to the heater box.

Here flue gases from all furnaces are tempered to about 1200 deg. F. temperature for ease of handling and are collected in two main headers. These gases are then taken to a heater box that contain parts to be reheated for forging and or dies used for presses and hammers. In this recommendation we have illustrated and used one heater box however it is possible to install several heater boxes to accommodate several parts. The heater box may include an internal recirculating fan for distribution of gases and heat transfer improvement. In this case we have not included such a fan. The heater box temperature is controlled at relatively low (600 deg. F. to 1000 deg. F.) value.

As shown below, when the flue gases at an average temperature of 1800 deg. F. are mixed with ambient air to drop the mixture temperature to 1200 deg. F., total volume of flue gases increases to 675,606 scfh.

Avg. heating gas temp	deg. F.	1200
Mixing air required to drop temp to 1200 deg. F.	scfh	232,967.64
Total vol of gases	scfh	675,606.16

Figure FO3.4 shows calculations for sizing the fan used for handling of flue gases at 1200 deg. F.

Possible savings through preheating of load using furnace flue gases		
Heat available in flue gases	MM Btu/hr.	18.23
Flue gas volume	scfh	442,638.52
Average flue gas temperature from forge furnace	deg. F.	1,800.00
Average available heat for forge furnaces	%	30.94
Assume % of the heat is used for preheating		35%
Heat used for preheating	MM Btu/hr	6.38
Gross heat equivalent reduced in furnaces	MM Btu/hr	20.62
Gas cost	\$/MM Btu	\$7.38
No. of hours/year		2000
Savings/year	\$/year	304,309.04

Figure FO3.4 Calculations for flue gas volumes at 1200 deg. F. and savings

Calculations for possible amount of heat recovery and weight of steel that can be heated to using these flue gases are shown in Figure. FO3.5. We have used steel as primary material used in the plant for forgings. It is assumed that steel can be heated to about 600 deg. F. without major problems in its handling. Heat content of the mixed flue gases is 18.23 MM Btu/hr. at 1200 deg. F. For heating steel to 600 deg. F. average temperature with 700 deg. F. surface temperature, average exhaust gas temperature from the heater box is assumed at 600 deg. F. (during starting from cold to the final temperature). This means that , after allowing for heat losses, about 45 % of the total heat from gases can be used for heating steel. We have used 40% of the heat going to steel.

Based on this heat going to steel = $(0.35 \times 18.23) = 6.38$ MM Btu/hr.

Steel sp. Heat	Btu/(lb. deg. F)	0.135
Average final temp.	deg. F.	600
Initial temp	deg. F.	60
Heat used/lb	Btu/lb	72.9
Heat used in preheating	MM Btu/hr	6.38
Wt. of the steel that can be heated	lbs./hr	87,502

Figure FO3.5 Calculations for heat recovery and steel preheating

Due to preheating of steel prior to charging, actual heat to be transferred to steel in the forging furnace would be reduced by this amount. Since the furnace operates at a low efficiency (assumed at maximum possible value equal to average available heat for all furnaces) of 30.94% (calculated from the furnace data), gross heat input for furnaces would reduce by $6.38/0.3094 = 20.62$ MM Btu/hr. It is assumed that the heater box would be used effectively for 2000 hours (as opposed to 2500 hours for furnaces) and using gas cost of \$7.38 per MM Btu, estimated savings with use of the heater box is \$304,309 per year.

As shown in the following calculations this arrangement would allow the plant to preheat about 87,000 lbs steel per hour if the system is used regularly.

Two major components are required for use of this system. They are flue gas fan and the heater box. Their sizes were calculated to estimate cost of the system. The fan size calculations are shown in Figure FO3.6.

I. D. Fan Sizing and Cost		
Total vol of gases	scfh	675,606
Fan volume at avg. temp	cfm	35,946
No. of fans used		2
Fan volume per fan	cfm	17,973
Fan wheel size	inch	45
Pressure drop required	w.c.	4
Motor HP (each with ~35% efficiency)	HP	20

Figure FO3.6 Calculations for fan sizing to handle flue gases at 1200 deg. F.

We would require two fans, each with 45 inch wheel size, and 20 HP motor. Assuming that these fans operate for 2500 hours (for 200 hours heater box operation), and electricity cost of 9.8 cents/kWh,

Total power cost for two fans = $2 \times (20 \times 0.746) \times 2500 \times 0.098 = \$7,310$ per year

Electricity use = $2 \times (20 \times 0.746) \times 2500 = 74,600$ kWh per year

The heater box size is based on volume of steel to be accommodated and a factor of five by which this volume has to be expanded for accommodating gas distribution etc.

Wt./ft. for 6 inch round billet	lbs./ft	94.25
No. of ft for possible weight to be heated		928.42
Cu.ft of steel	ft ³	182.30
Design volume for heater	times	5.00
Required volume	cu.ft.	911.48
Dimensions for a cube	ft.	9.70
Approx box size	ft	10 x 10 x 10

Recommendation

We recommend that the plant consider installation of a heating station at an appropriate location for preheating material to be heated in forging furnaces. This station can be used for preheating dies also.

Implementation Cost

Cost of the proposed system includes items such as cost of fans, heater box, hot duct work and insulation, controls and other miscellaneous hardware together with installation cost.

Cost is estimated based on standard tools and past experience with design and installation of such low to medium temperature heating systems. A summary of estimated costs is given below (Figure FO3.7).

Fan Cost Estimate - An interactive JavaScript equipment capital cost estimating aid (order-of-magnitude). These costs are helpful during a project's early development and budgeting. The actual cost of a piece of equipment depend upon many factors. You should exercise caution in use of this educational content.

Bin Cost
 Blender Cost
 Blower Cost
 Boiler Cost
 Centrifuge Cost
 Classifier Cost

Fan Type:
 Diameter should be between 12 and 36 inch
 Fan Diameter: inches
 Material:
 Pressure: Atmospheric

Cost 2003 US \$: **13200** F.O.B. Gulf Coast U.S.A.

Implementation Cost		
Cost of each fan for 2005 (5% price escalation)	\$	14,553
Cost for two fans	\$	29,106
Installation of fans	\$	6,000
Duct and insulation	\$	25,000
Controls etc.	\$	15,000
Heater box	\$	150,000
Installation + other parts	\$	40,000
 Total cost	 \$	 265,106

Figure FO3.7 Cost estimate for implementation of recommendation FO3.

Payback period

As mentioned above, savings are from reduced use of natural gas in forging furnaces (or die preheating). The system would use additional electricity for two fans.

Savings in gas use = 20.62 MM Btu/hr x 2000 hours/year = 41,240 MM Btu/year

Cost savings at \$7.38/MM Btu = 41,240 MM Btu/year x \$7.38/MM Btu = \$304,351 per year

(Note: These numbers are slightly different from the previous spreadsheet calculations due to rounding error)

Cost of electricity use for fans (as estimated above) = \$7,310 per year

Net possible savings = (\$304,351 - \$7,310) = \$297,041

Cost of implementation as discussed above = \$265,106

Simple payback period = (\$265,106/\$297,041) = 0.89 years or 10.7 months.

Summary of Savings

Natural gas

Reduction in gas use = 41,240 MM Btu/year

Carbon dioxide savings = 4,742,600 lbs./year

Dollar savings = \$304,351

Electricity

Additional use of electricity = 74,600 kWh per year

Additional carbon dioxide generation = 235,338 lbs./year

Dollar expenses for electricity = \$7,310 per year

AR CA1: Reduce Compressed Air Demand

	Annual Savings			Project Cost	Simple Payback
	Resource	CO ₂ (lb)	Dollars		
Electricity	142,957 kWh	328,800	\$13,438	\$13,500	12 months

Author: Len Bishop

Analysis

Analysis of a compressed air system requires review of the supply, distribution, and demand. Reduction in demand reduces the loaded hours and the energy costs on the compressed air system supply. The Audit determined that the average compressed air demand was 600-625 CFM. Based on average utility costs of \$0.094/kWh, the facility pays \$41,987 annually for electricity to generate compressed air.

If over a years time the 76 operating hours weekly are split evenly between the two compressors the electrical costs would be as calculated using the following formula

$$\begin{aligned} & (\text{Amps} \times \text{Volts} \times 1.73 \times \text{PF}/1000) \times \text{Hours} \times \$/\text{kW} = \text{Annual Energy Cost} \\ & (\text{PF of 2300 volt synchronous motors} = 0.8) \end{aligned}$$

Compressor #1

$$\begin{aligned} & (38 \text{ Amps} \times 2300 \text{ Volts} \times 1.73 \times 0.8)/1000 = 121 \text{ kW} \\ & 121 \text{ kW} \times (76 \times 52)/2 \times \$0.094 = \$22,475 \end{aligned}$$

Compressor #2

$$\begin{aligned} & (33 \text{ Amps} \times 2300 \text{ volts} \times 1.73 \times 0.8)/1000 = 105.1 \text{ kW} \\ & 105.1 \text{ kW} \times (76 \times 52)/2 \times \$0.094 = \$19,521 \end{aligned}$$

This equates to an annual energy cost of \$67.19 (\$41,999/625) per CFM or 714.8 kWh per CFM per year (assuming average demand of 625 CFM). The existing multi-step compressor controls provide very efficient control. A 200 CFM reduction in average demand will result in an annual reduction of 142,957 kWh or \$13,438 in compressed air energy costs.

A review of the compressed air system demands identified two specific opportunities to reduce the demand by at least 100 CFM. The first is Leaks in the compressed air system. The second is Open Blowing to clear condensate to avoid staining a final product.

Leaks

A sampling leak survey was completed with an ultra sonic leak detector on August 31, 2004. The 20 leaks identified accounted for an estimated 200 CFM. Based on experience in similar facilities SIFCO should assume the current leak load is at least 300 CFM. All of the leaks identified were in locations where it would have been nearly impossible to find without aid

of an ultrasonic leak detector. The leaks fell into three broad maintenance related categories; regulator and valve seals, hose and pipe fittings, and old red hose.

Condensate Purge Air

Compressed air is purged to eliminate the possibility of moisture staining the final product during blow off. Compressed air is continuously purged in the Pickling Area even though the need for air is intermittent. A 1/4 to 3/8 inch line is open to control moisture that can stain certain parts. The purge losses will range between 113-254 CFM assuming a line pressure of 110 PSIG. The addition of a properly sized point of use desiccant dryer and filters would reduce the purge to < 10 CFM and establish a constant compressed air quality.

Recommendation

We recommend SIFCO establish a Leak Management program and install a point of use contaminant removal equipment in the Pickling Area.

A basic Leak Management Program would include the purchase of an Ultra-Sonic Leak Detector and committing one day per quarter to finding and fixing Leaks. Leak Management is an ongoing program.

An engineered blow nozzle (air amplifier) should be selected for the Pickling Area point of use application. A desiccant dryer with appropriate filtration should be matched to the engineered blow off nozzle. This solution will establish a source of Clean Dry Air and reduced the existing purge losses by as much as 90%.

Estimated Savings

The reduction of average compressed demand by 200 CFM will have the following effects:

- ❑ The average CFM demand will be reduced from 625 to 425.
- ❑ Annual energy consumption will be reduced by 142,957 kWh.
- ❑ Annual energy costs will be reduced by \$13,428
- ❑ Enhanced product quality in the Pickling packaging area
- ❑ Enhanced pressure stability.

Leak Management

Establishing a Leak Management Program will conservatively reduce the average CFM demand in the facility by 100 CFM. To accomplish this SIFCO would need to purchase an Ultra-Sonic Leak Detector and commit 4 man days per year to finding and fixing Leaks. Quality Ultra-Sonic Leak Detection equipment can be purchased for \$2-4,000. The assumption has been made that internal labor cost would not exceed \$750 per day. Leak Management can be outsourced at \$1000-\$1500 per day.

Point-of-Use Contaminant Removal

Replacing the uncontrolled open blowing with an engineered solution will conservatively reduce the average compressed air demand 100 CFM. The combination of an engineered nozzle and contaminant removal package rated at 50-60 CFM will provide more than enough Clean Dry for the Pickling Area. The package shown on the left below is rated for 60 SCFM and has a purge loss of <10 CFM. The nozzle below is designed specifically for drying and is similarly rated. SIFCO should be able to purchase and install these products for <\$7500

Estimated Implementation Cost

The first year costs associated with a Leak Management program are the cost of a leak detector and 4 man-days to find and repair leaks. The ongoing costs of \$3000 per year for four man-days of leak detection and repair are factored into the IRR below. The Pickling area contaminant removal equipment will cost roughly \$7,500 to acquire and install. The total cost to implement will be \$13,500.

Estimated Simple Payback

1 year

AR CA2: Eliminate Compressor Cooling Water

	Annual Savings			Project Cost	Simple Payback
	Resource	CO ₂ (lb)	Dollars		
Water	11,856,000 gallons		\$47,424	\$100,000	25 months

Author: Len Bishop

Analysis

The plant has two water-cooled oil-lubricated reciprocating air compressors. The process of compressing air generates significant amounts of excess heat that must be managed. Both of the compressors use city water for cooling. The compressor cooling water is passed through the after-coolers and discharged to the sanitary sewer.

Only one compressor is required to support general plant air requirements. The compressors are manually controlled. One or the other compressor is on 76 hours per week. Compressor #1, a 200 HP, requires 45 GPM of cooling water when on. Compressor #2, a 250 HP, requires 55 GPM when on. Neither compressor has a water regulating valve to reduce cooling water usage.

Current annual cooling water requirements can be reduced by 11,856,000 gallons. Based on a combined water and sewer cost of \$4.00 per 1,000 gallons, a savings of \$47,424 is possible.

Analysis of the compressed air system demand established the current compressed air demand of 600-625 CFM.

Recommendation

We recommended elimination of the compressor cooling water with a change to air-cooled compressors. Two cost effective options exist. The first allows utilization of a pair of 100 HP air-cooled compressors currently in storage. The second option would require the purchase of two new 125 HP air-cooled compressors. With the existing storage fully utilized, the compressor with load no-load or variable speed drive controls will have similar energy costs to the existing compressors. A new compressor with variable speed drive controls would provide very stable air pressure and serve as an ideal 'trim' compressor as the compressed air system is expanded to replace steam as the primary energy source for the hammers.

Option #1 Use 100 HP Air-Cooled Compressors in Inventory

The first option available would be to use the 100 HP air-cooled compressors currently in storage. These compressors would be piped into the existing receivers. The internal check valves on the new compressors would eliminate the need for manual isolation of the back up compressor. Figure 1 shows the new piping layout. This storage configuration will allow the compressor to run most efficiently in a load no-load control mode.

This option is only available if the compressed demand reductions of at least 200 CFM, as recommended in AR #1, are completed. The 100 HP compressors are rated @ ~450 CFM. The electronic controls on the 100 HP compressors provides for automatic start up of the back up compressor in the event of a compressor failure or abnormal demand. The addition of a

Master Control would allow for automatic start up to match the current or future production scheduling.

Figure 1. Recommended location of 100 HP compressors in storage.

In the future, the system could be enhanced with the addition of a Pressure Flow Controller on the header piping leaving the compressor room. Pressure Flow Controllers in combination with Engineered Storage allow for the separation of the Compressed Air Supply and Compressed Air Demand. As the plant compressed air system expands to include all forge applications, this technology has the potential to reduce the online compressor horsepower required. The existing plant piping does not lend itself to the installation of a Pressure Flow Controller at this time.

Estimated Savings

The elimination of cooling water will have following effects:

- ❑ Up to 11,856,000 gallons of water will not be purchased from the local water authority
- ❑ Up to 11,856,000 gallons of water will not be discharged into the sanitary sewer.

Conservative costs for cooling water/sewer range from \$4-\$8 per 1000 gallons. Compressor #2, the 250 HP, requires an estimated 55 GPM of 80 F cooling water. Compressor #1, the 200 HP compressor requires an estimated 45 GPM.

$$\frac{52 \text{ Weeks}}{\text{Year}} \times \frac{76 \text{ Hours}}{\text{Week}} \times \frac{60 \text{ Minutes}}{\text{Hour}} \times \frac{50 \text{ Gallons}}{\text{Minute}} \times \frac{\$4.00}{1000 \text{ Gallons}} = \$47,424/\text{yr}$$

With the compressors running in a load no-load mode and a nominal pressure band of 6-10 psig, the new compressors will have roughly the same energy costs of the existing compressors. Annual maintenance costs should be equal to or lower than the existing compressors.

Estimated Implementation Cost

SIFCO already owns the compressors discussed. Installation costs are assumed not to exceed \$20,000.

Estimated Simple Payback

$\$20,000 / 47,424 \text{ /yr} \times 12 \text{ months/year} = 5 \text{ months}$

Option #2 Purchase and install 125 HP Air-Cooled Compressors

The second option available would be to purchase and install a pair of new 125 HP air-cooled compressors. These compressors would be piped into the existing receivers. The internal check valves on the new compressors would eliminate the need for manual isolation of the back up compressor. Figure 2 shows the new piping layout. This storage configuration will allow the compressor to run most efficiently in a load no-load control mode.

This option can be implemented prior to any compressed demand reductions recommended in AR #1 are completed. The 125 HP compressors are rated @ >630 CFM @ 100 PSIG. The electronic controls on new compressors provides for automatic start up of the back up compressor in the event of a compressor failure or abnormal demand. Most new microprocessor controls simplify the integration of multiple compressor rooms and remote monitoring service packages.

In the future, the system could be enhanced with the addition of a Pressure Flow Controller on the header piping leaving the compressor room. Pressure Flow Controllers in combination with Engineered Storage allow for the separation of the Compressed Air Supply and Compressed Air Demand. As the plant air system expands to include all forge applications this technology has the potential to reduce the online compressor horsepower required. The existing plant piping does not lend itself to the installation of a Pressure Flow Controller at this time.

Estimated Savings

The elimination of cooling water will have following effects:

- ❑ Up to 11,856,000 gallons of water will not be purchased from the local water authority
- ❑ Up to 11,856,000 gallons of water will not be discharged into the sanitary sewer.

Conservative costs for cooling water/sewer range from \$4-8 per 1000 gallons. Compressor #2, the 250 HP, requires an estimated 55 GPM of 80 F cooling water. Compressor #1, the 200 HP compressor requires and estimated 45 GPM.

$$\frac{52 \text{ Weeks}}{\text{Year}} \times \frac{76 \text{ Hours}}{\text{Week}} \times \frac{60 \text{ Minutes}}{\text{Hour}} \times \frac{50 \text{ Gallons}}{\text{Minute}} \times \frac{\$4.00}{1000 \text{ Gallons}} = \$47,424/\text{yr}$$

With the compressors running in a load no-load mode and a nominal pressure band of 6-10 psig the new compressors will have roughly the same energy costs of the existing

compressors. Annual maintenance costs should be equal to or lower than the existing compressors.

Estimated Implementation Cost

The cost of a 125 HP air-cooled compressor with micro-processor based load no-load controls producing at least 630 CFM @ 100 PSIG should be less than \$40,000 each. Installation costs are assumed to not to exceed \$20,000.

Estimated Simple Payback

$\$100,000 / \$47,424 \text{ /yr} \times 12 \text{ months/year} = 25 \text{ months}$

AR M1: Use High Temperature Die Materials and Lubricating Coatings

	Annual Savings			Project Cost	Simple Payback
	Resource	CO ₂ (lb)	Dollars		
Tooling			\$275,000		
Productivity			\$150,000		
Coatings			-\$180,000		
Net			\$245,000	None	Immediate

Authors: Neil Chaudhry and Kelly Kissock

Analysis

Data for the analysis and recommendation were obtained during the project kick-off meeting on April 14, 2004, and from Ms. Linda Slack, Comptroller, and from Mr. Gary Danielson, Die Shop Supervisor, on October 5, 2004.

SIFCO purchases several different types of die materials including D2, INCO-718, 4140, 4340, 2714, WFT-1, FX 1 & 2, H13, Rene 41, and few other alloys. Finkle-Steel Temper 2, which is a modified 4340 tool steel, makes up about 70% of all dies. Twenty percent of all die materials are the 2714 Elwood alloy, which are used for the inserts. Ten percent of the dies consist of INCO-718, H-13, and other materials like D2, Rene 41.

In fiscal year 2004, SIFCO spent \$44,000 for Die Steel and \$191,000 for Die Block Material. In addition, die material and processing costs were about \$357,000 to outside vendors and \$497,000 for in-house costs. Thus, the total material and labor cost for dies in 2004 was about \$1,100,000. SIFCO uses about 20 die sets per year. Thus, the die sets cost about \$55,000 per set.

During a typical day, the forging presses are down about 30%-40% of the time. Of this, about 20%-25% of the time is for tooling changes and 10%-15% of time is for non-tooling changes such as tool adjustment, part issues, etc. The cost of labor and overhead in 2004 was about \$15,000,000. The cost of energy was about 8% of sales or about \$2,400,000 and material costs were about 40% of sales or \$12,000,000.

Recommendation

We recommend using high-temperature, high-performances die materials and applying a hard, wear-resistant, lubricating coating on the dies and the inserts to increase die life. For example, IN-100 is a high-temperature, high-performance Ni-Cr-Co cast alloy with excellent thermal fatigue and crack resistant properties. Similarly, customized die inserts made by RSP Tooling in Solon, Ohio use rapid solidification that provide excellent wear resistance. The bi-metallic tool increases wear resistance on the surface and low thermal conductivity material on the back keeps dies hotter. The existing dies and inserts could also be precision coated with the LaserCut964 coating by SAC Coatings in Dayton, Ohio. The use of these advanced die sets would result in greater die life, reduced downtime, and lower part defects.

Estimated Savings

We recommend identifying “tough jobs” that have a history of high tool wear. For example, according to Mr. Gary Danielson, the “toughest job” as of Oct. 5, 2004 is Job # 30247. The die sets are approx. 8-10 inches in diameter by 8 inches long. Mr. Danielson mentioned that they were looking to try an “EMT – Deep Nitriding Treatment”, but were not sure it would be beneficial.

Next, we recommend using high-temperature, high-performances die materials with wear-resistant, lubricating coating for these dies. This could be done by asking an expert from a vendor company to 'baby-sit' trial runs and optimize for dies for the specific operating conditions at SIFCO.

Based on previous experience with these dies and coatings, we estimate that the tooling would last from 50% to 500% longer than uncoated or conventionally-coated tools. If tool life were improved by an average of 50% for 50% of the tooling, then tooling savings would be about:

$$\text{\$1,100,000 /yr} \times 50\% \times 50\% = \text{\$275,000 /yr}$$

In addition, productivity would increase since harder tools can be run at a faster rate. Again, customer histories with the SAC coatings have shown 10% to 100% increases in throughput. Assuming 10% of the total \\$15,000,000 per year spent on labor and overhead is directly devoted to the forge presses and that increased productivity could reduce these costs by 10%, the productivity savings would be about:

$$\text{\$15,000,000 /yr} \times 10\% \times 10\% = \text{\$150,000 /yr}$$

Based on previous experience, we estimate that the cost of coating all forging tools used by SIFCO would be about \\$130,000 per year. There would be an additional cost for coating optimization and testing of about \\$50,000 per year. Thus, the total cost would be about \\$180,000 per year.

The net annual savings would be about:

$$\text{\$275,000 /yr} + \text{\$150,000 /yr} - \text{\$180,000 /yr} = \text{\$245,000 /yr}$$

Estimated Implementation Cost

There are no one-time implementation costs, since all costs are ongoing.

Estimated Simple Payback

Immediate

AR P1: Convert Steam Hammers to Compressed Air Hammers

	Annual Savings			Project	Simple
	Resource	CO ₂ (lb)	Dollars	Cost	Payback (months)
Natural Gas	165,200	18,998,000	\$ 1,219,176	\$ 600,000	8.66
	MM Btu				
Electricity	-4,051,093	-12,781,198	\$ (392,956)		
	kWh				
Water	16,177,500	0	\$ 72,799		
	Gallons				
Persoonel + Maintenance			\$ 314,468		
Compressor Lease + Maint.			\$ (382,389)		

Author: Arvind Thekdi and Len Bishop

Analysis

SIFCO forging plant uses a large number of hammers and presses fro forging of steel and titanium parts. All of these forging equipment are powered by steam obtained from one or more of the three boilers. Each boiler is designed to produce approximately 60,000 lbs./hr. The steam is generated at 115 psig and temperature of 340 deg. F. Steam is distributed through out the plant by stem headers. The steam distribution system includes a number of steam traps. Steam used in the forging equipment is contaminated by lubricating oil and cannot be reused as condensate. More than 80% of the steam is vented through stacks. Since neither the latent heat of steam nor condensate from steam is recovered, use of steam represents inefficient use of energy for the plant operations.

In addition to expenses for energy use, there may another looming of impending state EPA regulations threat that would require a large investments to ‘clean” vented steam. The current schedule for State Implementation Plan (SPI) related to non attainment indicates that it may be necessary to clean the vented steam or stop discharging lubricant laden steam by as early as the year 2008. Obviously, this presents a great economic and business challenge for the plant.

For these energy and environmental reasons, the forging industry has been trending toward the conversion of steam driven hammers to compressed air. Although compressed air system uses electricity as a source of energy, it utilizes energy in a much more efficient way and does not present the environmental challenges. Additionally, as discussed later, use of automated compressors eliminates dedicated manpower required for the boiler operation resulting in substantial economic advantage and savings.

The plant currently has nine single-blow forging hammers. The forging hammers range in size from 2000 to 22000 lbs. These forging hammers are currently powered by steam generated by natural gas fired boilers. A counter-blow hammer rated @ 45 MkG (calculated to be the equivalent of a ~33,000 lbs) was in the process of installation during the system analysis. The documentation available on the forging hammers does not include actual steam or compressed air requirements.

A review of forging industry documents and calculations using compressed air industry standards resulted in similar compressed air requirements for each of the hammers. The table given in Figure P1.1 below quantifies the maximum compressed air requirements for each hammer as presented by forging industry leaders and compressed air industry calculations. Differences in the CFM requirements are based on the methodology. The forging industry uses a flat 35 CFM per 100 lbs of rated force rule-of thumb. The compressor industry calculations take into consideration cylinder size, compression ratio, and cycles per minute.

Hammer Size (lbs.)	Forging Industry (CFM)	Blows per Minute	Compressor Industry (CFM)
2,000	700	60	1,173
5,000	1,750	50	2,443
6,000	2,100	50	2,593
8,000	2,800	50	2,616
12,000 (2)	4,250	45	3,908
15,000	5,250	45	4,528
20,000	7,000	40	5,143
22,000	8,750	40	7,494
35,000	12,250	20	4,690
Totals	49,000		38,497

Figure P1.1 Compressed air requirement for forge hammers

The nature of the forging process places an intermittent demand on the compressed air system. The primary intermittent demand is refilling the cylinder that resets the hammer weight for the next blow. Secondary intermittent demands associated with forging include Key Knockers, Rams, and Scale Blow Off. The current compressed air system supports all the Key Klocker, Ram, and Scale Blow Off demands in the Forging Shop. This demand averages < 600 CFM in total.

This intermittent nature of the forging operation encourages the use of certain storage and controls strategies that reduce the required on-line horsepower. As noted above, the 'peak' demand possible in the SIFCIO foundry using Forging Industry guidelines would be almost 50,000 CFM if all 10 hammers were operating at the same time. This would require as much as 10,000 HP of compressed air equipment.

According to Ed Rogers at SIFCO the maximum number of hammers in operation at any one time would be 5. This reduces the peak demand to <38,000 using the forging industry guidelines and <26,000 using the compressor industry calculations.

Both the forging and compressor industries require storage to support large intermittent demands. Again, the forging industry uses the rule-of-thumb of 20% capacity of free air volume. The compressor industry calculations take into consideration pressure drop when calculating storage. The table below (Figure P1.2) documents the storage requirements as recommend by the forging industry and those of the compressor industry limiting the pressure drop to 10 psig.

Hammer Size	Forging Industry		Blows per min.	Compressor Industry	
Lbs.	ft ³	gallons		ft ³	gallons
2,000	140	1,047	60	227	1,700
5,000	350	2,618	50	331	3,542
6,000	420	3,142	50	503	3,760
8,000	560	4,789	50	564	3,794
12,000 (2)	840	6,283	45	842	5,667
15,000	1,050	7,854	45	975	6,566
20,000	1,400	10,472	40	1,371	7,458
22,000	1,750	13,090	40	1,634	10,866
35,000	2,450	18,326	20	909	6,801
Totals	9,800	73,304		8,198	61,623

Figure P1.2 Compressed air storage requirement for forge hammers

With proper storage and distribution piping, demand diversity will also factor into required compressor capacity and the true operating costs. The demand diversity equation will be unique to SIFCO. The chart below (Figure P1.3) shows the cubic feet of compressed air @ 100 PSIG required for a single stoke of the individual hammers. Which hammers are in operation and the pace of the five crews that may be working at any given time will dramatically affect the required compressor capacity on-line?

Hammer Size (lbs)	Cubic feet of 100 PSIG
	Compressed Air per Blow
2,000	20
5,000	49
6,000	52
8,000	58
12,000 (2)	87 each
15,000	101
20,000	129
22,000	187
35,000	235

Figure P1.3 Per blow compressed air requirement for forge hammers

SIFCO needs the compressed air capacity available to meet the maximum demand. The compressors and their control system need to supply as much or as little compressed air as might be required. If the 5 largest hammers are operating at 20 blows per minute the peak compressed air demand would be 14,780 CFM $((87+101+129+187+235)*20)$. If only two hammers are operating at an average of 10 blows per minutes the demand drops to a peak 4,220 CFM $((187+235)*10)$. If the five smallest hammers were working at 60 blows per minute the peak compressed air demand would be 11,970 CFM $((20+49+52+58+87)*45)$. If the same five hammers are running at 20 blows per minute the peak compressed air demand drops to 6,384 CFM $(15960 * (20/50))$.

The peaks are only reached if all five hammers are synchronized. In reality the hammers are all operated independently. The four examples show the effect of demand diversity on the selection of compressed air equipment.

Recommendation

It is recommended that the steam driven forging hammers be converted to compressed air for both energy and environmental reasons. This can be accomplished by following industry-accepted recommendations for storage sizing and locations, distribution piping, and compressor sizing.

SIFCO should plan to add compressed air storage equal to or greater than the recommendations above. The more stored air in the system the less compressed air capacity needs to be on-line. SIFCO should plan to put roughly 50% of the storage requirements close to the individual hammer and the other 50% on the supply side of a pressure-flow-controller that will effectively separate the supply from the demand.

SIFCO should plan to install a 12-inch loop in the forging area. Using CAGI Table 13.23 the pressure drop on a 500 ft loop would be < 1 psig with a flow of 30,000 CFM. Drops to individual hammers and their individual regulators should be sized for maximum demand.

SIFCO should start with 5 compressors rated 3250 CFM @ 100-110 PSIG controlled by a common master control. The individual compressors should incorporate load no-load capacity controls. The addition of remote monitoring would reduce the internal maintenance manpower required. The compressors should be air-cooled or utilize an engineered closed loop cooling system to eliminate the need for an external source of cooling water. Note: Centrifugal compressors do not have the turn down capabilities to efficiently match the demand dynamics typical of a foundry operation.

These recommendations are based on information available at the time of this report. It is recommended that a more thorough analysis of potential hammer rates (blows per minute) be completed prior to moving forward with a total conversion to compressed air.

Estimated Savings

Steam is currently produced in natural gas fired boilers. The total cost of steam production includes several components such as utility (water, electricity, natural gas etc.) cost, maintenance cost and labor (boiler operator) cost. In case of boilers, due to age of the boilers, no consideration is given to amortization of capital cost. A detail cost analysis for steam production and comparable compressed air system is carried out and reported in the following.

Figure P1.4 shows cost of water and natural gas used for steam production. The bottom line shows that total cost of fuel is about \$1.22 million dollars per year and cost of water is about \$56,600 per year. Power cost for pumps is based on 300 HP power use for the pumps. This is considered “conservative” and actual cost may be more than reported here. Maintenance cost is assumed as 2.5% of the steam cost.

Total cost savings for steam system \$/year \$ 1,661,181		
Power cost for pumps etc.		
Electricity cost	\$/kWh	\$0.098
Total HP used for power house approx.	HP	300
Cost of power/hr	\$/hr	\$21.93
Power cost	\$/year	\$54,831
Maint cost etc.		
% of steam cost	%	2.5%
Maint cost etc.	\$/year	\$33,668
Personnel cost		
No. of boiler operators		3
Hrs/year per person	\$	30.00
No of man hours for operators per year (total) (52wks, 5 days, 8hrs/day)		2080
Labor cost	\$/year-person	\$ 62,400
Overhead, benefits etc.	%	50%
Total cost related to boiler operators	\$/year	\$ 280,800
Cost of steam - Gas cost for the boilers		
Fuel cost	\$/MM Btu	\$ 7.38
Production	lbs/hr	60000
No. of hours/year		2500
Boiler efficiency		80%
Heat supply to steam	Btu/lb	1,101.25
Heat input	MM Btu/hr	66.08
Fuel cost	\$/hr	\$ 488
Fuel cost	\$/year	\$ 1,219,084
Cost of water		
Feed Water to boiler	gph	7,190
Condensate return	% of steam	10%
Actual water used	gph	6,471
Cost of water incl. Treating	\$/1000 gallons	\$ 4.50
Cost of water	\$/year	\$ 72,798

Figure P1.4 Estimate of cost and possible savings for current steam supply

We have been informed that it is SIFCO's intension to outsource the supply of compressed air. The outsourced services and hardware would include compressors, controls, cooling system, and primary storage with pressure flow control. Maintenance, remote monitoring, and regular system status reports should be included in the responsibility of the compressed air vendor. It is estimated the monthly cost should be ~\$30,000.

Instrumentation to monitor power consumption and the flow of compressed air delivered are items that will need to be negotiated. We recommend any contract also include quarterly leak management services and an annual assessment of the compressed air demands.

Experience and preliminary estimates suggest the average load on the compressed air system after conversion to steam will be <10,000 CFM. If we assume the same running hours as boilers and forging equipment (2500 hours/year) 3 compressors each rated 3250 CFM will support the average demand and a 4th unit comes on to satisfy peaks. This assumes leaks are managed and the system is shut down when the forge shop is inactive. Cost related to compressed air system lease, power cost and maintenance cost are given in Figure P1.5.

Based on the assumptions discussed above, potential annual savings will be \$1,661,181 - \$830,176 = 831,005.

Total Cost of Compressed Air System			
Leased system			
Monthly cost for lease	\$/month	\$	30,000
Annual cost for compressor lease	\$/year	\$	360,000
Power cost/month			
No. of compressors running (average)			3.5
HP for compressors - each			700
No of hours/year	hrs/year		2500
Power cost/year	\$/year		\$447,787
Maint. Cost			
As % of power cost			5.00%
Maint. Cost per year	\$/year		\$22,389
Total annual cost - compressed air related	\$/year		\$830,176

Figure P1.5 Estimate of operating cost for the proposed compressed air system

Implementation Cost

As mentioned earlier it is SIFCO's intention to outsource the supply of compressed air. With this in mind there will not be any investment required for purchase of compressors, installation and associated hardware cost.

However SIFCO would be responsible for the upgrades to the compressed air system infrastructure. This would include the 12-inch header and individual storage, pressure controls associated with each hammer. It is estimated the initial cost will be \$500,000.

Instrumentation to monitor power consumption and the flow of compressed air delivered are items that will need to be negotiated. We recommend any contract also include quarterly leak management services and an annual assessment of the compressed air demands.

Additional cost related to site preparation etc. is conservatively (high side) assumed to be \$100,000.

Total implementation up-front cost for SIFCO would be \$600,000.

Simple Payback

Simple payback on SIFCO's investment = $600,000 / 831,005 = 0.73$ years or 8.7 months.

Savings

Natural gas savings = 162,200 MM Btu/year

Dollar savings = 1,219,176

Carbon Dioxide savings = 18,998,000

Electricity Use = - 4,051,093

Dollars Savings = \$392,956

Carbon Dioxide savings = 12,781,198

Personnel and maintenance cost savings for steam system = \$314,468

Lease and maintenance cost for compressed air system = - 382,389

VII. Implementation and Replication Plan

This section summarizes Sifco'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.

The forging industry uses more than 500 Trillion Btu (fuel and electricity) while the heat treating industry uses more than 600 Trillion Btu per year. These figures include energy use in all categories (heating, steam, compressed air, electric motors etc.). Replication in the forging industry by the Forging Industry Association (FIA) could result in 40 Trillion Btu per year. This is based on a conservative estimate of 15% average savings and replication in the 50% of all forging companies. This does not include additional energy savings related to replication in the heat treating industry and all other industries that use fuel fired and induction furnaces, small to medium capacity steam systems, compressed air systems, pumping systems etc. It is not possible to make meaningful projections for these energy savings.

Results of this assessment are applicable to two major industries (forging and heat treating) and several other areas of energy consuming operations in industrial plants. The forging industry sales for the year 2001 were approximately \$5.82 Billion. Based on this, SIFCO Forge Group represents approximately 1% of the industry sales and the benefits and savings identified at SIFCO can have as many as 100 times more potential applications within the forging industry. Practically, even if 25% of the industry adapts energy saving measures identified at SIFCO, the total savings for the industry could amount to more than \$40 million per year. This does not include savings related to the heat treating industry replication.

Replication in forging industry will be carried out through close cooperation of the forging industry trade and technical organization FORGING INDUSTRY ASSOCIATION, with headquarters in Cleveland, Ohio. It comprises of over 140 North American producers of forged metal components, accounting for about 54% of the custom forging volume produced annually. The membership companies supply forged metal for critical applications in the aerospace, automotive, construction, materials handling, ordnance, hand tool, agricultural and industrial equipment industries – to name a few! Some 75 firms supplying equipment, materials or technical services to the forging industry are also members of FIA. The Association, with its predecessor organizations, has served the metal forging industry since 1913. More information on FIA and its member companies is available on the forging industry website at www.forging.org. Mr. George Mochnal, Technical Director of FIA is a team member for the assessment and will support activities in preparation of material that would be transmitted to the forging industry.

In addition to this two other organizations will contribute to the replication efforts. Dominion East Ohio Energy Company, located in Cleveland serves more than 1000 major industrial customers who could benefit from use of results of this assessment. Dominion will collect and disseminate appropriate information to these industries through their marketing staff, new letters, seminars etc. throughout their market territory that encompasses Ohio, Pennsylvania, West Virginia and Virginia. Another team member and project supporter, The

Ohio Department of Development through its energy efficiency office will disseminate energy efficiency improvement related information to the Ohio industries through various means such as statewide seminars and publications.

VIII. Project Participants

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