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Installation Wide Conservation Demonstration

**US Army Corps of Engineers**
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# Installation-Wide Energy Conservation Demonstration at Fort McClellan, Alabama

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**Five major areas of energy conservation were identified and investigated: (1) pressure reduction in district steam heating systems; (2) reduction of outdoor air in heating, ventilation, and air-conditioning (HVAC) systems; (3) replacement of oversized and inefficient motors in HVAC systems; (4) reduction of outdoor air infiltration in family housing; and (5) combustion optimization of gas-fired heating equipment. Other areas of investigation included radio controlled exterior lighting, and temperature reduction in the high temperature hot water system. Each conservation project was evaluated on a small scale to verify energy savings before it was implemented.**

**An energy information management system was developed to maintain annual consumption data for each building. The system provides immediate feedback on energy use so managers can make correct decisions on conservation measures.**

**The energy conservation programs that were implemented at Fort McClellan contributed to the 14 percent reduction in baseline (weather independent) energy consumption from FY84 to FY86. These programs have wide applicability to other U.S. Army installations. This research has also shown the importance of preliminary, small-scale testing of energy conservation programs before implementation.**

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The objective of the installation-wide energy conservation demonstration at Fort McClellan, AL, was to evaluate the effectiveness of applying available energy conservation technologies and techniques on an installation-wide basis to produce significant and predictable reductions in energy use and cost.

Five major areas of energy conservation were identified and investigated: (1) pressure reduction in district steam heating systems; (2) reduction of outdoor air in heating, ventilation, and air-conditioning (HVAC) systems; (3) replacement of oversized and inefficient motors in HVAC systems; (4) reduction of outdoor air infiltration in family housing; and (5) combustion optimization of gas-fired heating equipment. Other areas of investigation included radio controlled exterior lighting, and temperature reduction in the high temperature hot water system. Each conservation project was evaluated on a small scale to verify energy savings before it was implemented. (Cont'd)


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An energy information management system was developed to maintain annual consumption data for each building. The system provides immediate feedback on energy use so managers can make correct decisions on conservation measures.

The energy conservation programs that were implemented at Fort McClellan contributed to the 14 percent reduction in baseline (weather independent) energy consumption from FY84 to FY86. These programs have wide applicability to other U.S. Army installations. This research has also shown the importance of preliminary, small-scale testing of energy conservation programs before implementation.

## FOREWORD

This research was performed by the Energy Systems (ES) Division, U.S. Army Construction Engineering Research Laboratory (USA-CERL) for the Engineering and Housing Support Center (USAEHSC). The work was completed under Project 4A263734DT09, "Energy and Energy Conservation"; Task 02, "Installation Energy Conservation Strategy"; Work Unit 001, "Installation Wide Conservation Demonstration." Mr. B. Wasserman, CEHSC-FU, served as the USAEHSC Technical Monitor. Mr. Larry Windingland and Mr. William Dolan were USA-CERL's Principal Investigators. Dr. G. R. Williamson is Chief of ES. The Technical Editor was Gloria J. Wienke, Information Management Office.

COL Carl O. Magnell is Commander and Director of USA-CERL and Dr. L. R. Shaffer is Technical Director.

## EXECUTIVE SUMMARY

At many Army installations, energy consumption and the associated costs are increasing due to greater demands on heating, ventilation, air-conditioning (HVAC), and lighting. Between Fiscal Year (FY) 83 and FY87, the U.S. Army Construction Engineering Research Laboratory (USA-CERL) developed and implemented a research demonstration program at Fort McClellan, Alabama that evaluated the effectiveness of applying energy management techniques and energy conservation measures on an installation-wide basis to produce significant and predictable reductions in installation energy use and cost.

Fort McClellan was selected as the demonstration site for this research because it has a history of increased energy consumption, and has sufficient heating and cooling requirements to provide an appropriate sample to test technologies that affect these requirements. The installation is small enough that the energy reduction can be readily seen in the total utility bill and the energy cost per British thermal unit (Btu) is close to the national average.

Five major areas of energy conservation were identified and investigated: (1) pressure reduction in district steam heating systems; (2) reduction of outdoor air in HVAC systems; (3) replacement of oversized and inefficient motors in HVAC systems; (4) reduction of outdoor air infiltration in family housing; and (5) combustion optimization of gas-fired heating equipment. Other areas of investigation included radio controlled exterior lighting, and summer shutdown of the district high temperature hot water (HTHW) system. Projects representing each of the five major areas are discussed below.

Prior to implementation, each proposed conservation project was evaluated considering factors such as the initial cost, payback period, and Army-wide applicability to determine potential energy savings and economic feasibility. Some projects with a modest return on investment were rejected. The conservation programs that were implemented contributed to the 14.4 percent reduction in baseline (weather independent) energy consumption since 1984. These programs have wide applicability to other Army installations.

To identify energy conservation opportunities in buildings served by district steam heating, condensate meters were installed to measure the amount of steam energy used by each building and to verify the actual savings of implemented projects. By using meters, management personnel can forecast energy use, identify buildings that are using heat inefficiently, and assess the overall efficiency of the district heating system. Although the Army does not have a policy that mandates metering of energy use in buildings served by central plant heating systems, it is recommended that installation of metering equipment in future construction as well as in existing buildings be considered.

One measure to improve fuel-use efficiency and also reduce operating costs is to reduce boiler operating pressure where possible in district heating systems. At many Army facilities, the boiler operating pressure is maintained at a constant rate throughout the year regardless of demand. The objective of this Fort McClellan project was to reduce system conduction and live steam losses by implementing a reduction in steam pressure at the central boiler plants while maintaining the original heating capacity of the system.

Predicted values for building steam demand and information gathered through condensate metering enabled USA-CERL researchers to develop an accurate means of computing energy savings. The savings were calculated by attributing 75 percent of the losses to conduction and the remainder to leaks and faulty valves. The project cost was $100,000; a simple payback of 1.5 years was achieved. It is recommended that other agencies with district heating systems operating at much higher than end-use pressures consider this type of retrofit. In any case a detailed engineering analysis must be performed before implementation.

To reduce the amount of outdoor air infiltration in HVAC systems, measurements of the outdoor air intake were taken in 249 air-handling systems. The dampers were adjusted on those systems that were not in compliance with the most recent American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) standards on outdoor air requirements.

Operating efficiency measurements were taken on 201 motors used to power fans in HVAC systems. Sixty-eight of the motors that were operating inefficiently were replaced with properly sized motors, resulting in a 17 percent decrease in electrical power demand.

A study was conducted to determine the potential energy savings of reducing infiltration of outdoor air in family housing units. The results show that the average reduction in leakage area was 37 percent for the four units retrofitted--mostly the result of sealing leaks in the building envelope. By investing 3 hours of labor and $60 in materials per unit, an average of $38 annual savings per unit can be recognized. This represents an approximate simple payback period of 3 years. Because this did not meet the requirement of rapid payback, an installation-wide infiltration reduction program was not implemented as part of the demonstration. However, such a program may apply to installations with larger heating requirements.

To demonstrate the effectiveness of combustion adjustment of gas-fired heating equipment, researchers evaluated 204 furnaces and hot water heaters with a combined heating capacity of 88,952,000 Btu/hr. The combustion efficiency of more than half the units was improved, with efficiency increases as high as 25 percent in some cases and an average improvement for all the units of 2.5 percent. Based on the average heating season and fuel costs for Fort McClellan, this project is estimated to have saved $16,000 with an investment of $17,000 in labor and equipment costs; hence, a simple payback of 1.1 years was achieved. Combustion optimization not only improves heating efficiency and cost-effectiveness, it also minimizes dangers to the environment and to occupants of the heated space presented by incomplete combustion.

USA-CERL developed an energy information management system that maintains annual consumption data for each building located at the installation. The program uses input from the DEIS II report, post meter readings, boiler logs, and utility bills. The management system provides immediate feedback on how energy is distributed to and used by the various functions at an Army installation. Target areas for energy reduction can be readily identified and the extent of the potential application of an energy conservation opportunity and associated savings can be computed. By using this system, energy managers can perform consistent analyses on energy consumption and make correct decisions on the course to take in energy conservation.

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# INSTALLATION-WIDE ENERGY CONSERVATION DEMONSTRATION AT FORT McCLELLAN, ALABAMA

## 1 INTRODUCTION

### Background

Energy consumption and the associated costs are increasing at many Army installations. These increases are often due to greater demands on heating, ventilation, and air-conditioning (HVAC), and lighting. In a continuing effort to curb rising consumption/costs, the Office of the Chief of Engineers (OCE) asked the U.S. Army Construction Engineering Research Laboratory (USA-CERL) to demonstrate energy conservation measures on an installation-wide basis. Between Fiscal Year (FY) 83 and FY87, USA-CERL developed and implemented a research demonstration program to investigate energy problems on an installation. The program has potential for application at other installations.

### Objective

The objective of this research was to demonstrate and evaluate the effectiveness of applying energy management techniques and energy conservation measures on an installation-wide basis to produce significant and predictable reductions in installation energy use and cost.

### Approach

Fort McClellan was selected as the demonstration site for this research because it has a history of increased energy consumption, and has sufficient heating and cooling requirements to provide an appropriate sample to test technologies that affect these requirements. An Energy Engineering Applications Program (EEAP) study conducted at Fort McClellan identified areas with significant savings potential. Also, the installation is small enough that the energy reduction can be readily seen in the total utility bill and the energy cost per British thermal unit (Btu) is close to the national average.

Five major areas of energy conservation were identified and investigated: (1) pressure reduction in district steam heating systems; (2) reduction of outdoor air in HVAC systems; (3) replacement of oversized and inefficient motors in HVAC systems; (4) reduction of outdoor air infiltration in family housing; and (5) combustion optimization of gas-fired heating equipment. Other areas of investigation included radio controlled exterior lighting, and summer shutdown of the district high temperature hot water (HTHW) system.

Data collected through metering and monitoring were evaluated before deciding whether or not to implement a project. This approach provided an accurate assessment of project viability based on current energy use in comparison to the retrofit designs.

The projects are listed and briefly discussed in Chapter 2, and are discussed in detail in Chapters 3 through 6.

**Mode of Technology Transfer**

Results of various projects in this demonstration will be transferred through the issuance of Engineering Technical Notes (ETNs) describing individual energy conservation technologies verified by the demonstration.[1] It is also recommended that the information be used in formulating energy conservation study/summary scopes of work for installation-wide energy conservation opportunities.

[1]Engineer Technical Note (ETN), *Methods of Identifyting Oversized and Inefficient HVAC Motors in Army Installations*, Draft (August 1984); ETN, *Metering of Steam Heating Energy*, Draft (August 1984); ETN, *Reduction of Outdoor Air Infiltration in Air Handling Systems*, Draft (August, 1984); ETN, *Combustion Optimization of Residential Heating Equipment*, Draft (December 1985); ETN, *Reduction of Outdoor Air Infiltration in Family Housing*, Draft (January 1986); ETN, *Radio Controlled Street Lighting*, Draft (January 1986).

## 2 PROJECT SUMMARIES

Over the past 10 years, Fort McClellan has experienced considerable growth in both facility square footage and population served. With this growth came an increase in energy use. In FY76 Fort McClellan had a reported* thermal energy use of 99,526 Btu/sq ft and an electrical energy use of 30,623 Btu/sq ft. By FY79 energy use had risen to 99,899 Btu/sq ft and 39,091 Btu/sq ft for thermal and electrical use, respectively. It can be seen that a substantial energy conservation effort was already underway. The energy use for FY84 was 71,453 Btu/sq ft thermal and 34,904 Btu/sq ft electrical. Data for FY86 showed a thermal usage of 61,491 Btu/sq ft and 31,495 Btu/sq ft for electrical. This is a 14.4 percent reduction since 1984. The projects listed below and described in detail throughout the report contributed to this reduction.

### Energy Metering

Condensate meters were installed at all buildings that receive steam from the central plants. While this project did not cause any direct energy savings, knowledge of the steam demand of individual buildings provides management personnel with valuable information about the way heat is used at the installation. This information can be used to forecast energy use or to identify buildings that are using heat inefficiently. Also, overall efficiency of the district heating system was assessed using this information.

Electrical meters were installed on a small scale to evaluate potential energy conservation measures and to verify the actual savings of implemented projects. In particular, a large star-shaped barracks and the central chiller in boiler plant No. 4 were metered to assess the feasibility and confirm the energy savings from the project titled Boiler Plant No. 4 Summer Shutdown.

Short term, portable metering was used for project evaluation in areas such as electrical motor energy, air flow rates, and combustion efficiency measurements.

### Steam Pressure Reduction in District Heating System

Steam pressure reducing stations were replaced to allow boiler plants 1 through 3 to provide steam for district heating at reduced pressures. Reducing steam pressure also reduces the steam temperature, thereby reducing energy losses that occur through conduction of heat from the distribution system to the ground or elsewhere (e.g., in the boiler plant).

### Steam Vault Drainage

Sump pumps were installed in 16 steam vaults that contain valves, tees, traps, etc., for the steam distribution system of boiler plants 2 and 3. This reduced the heat transfer losses resulting from vault flooding.

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**Facilities Engineering and Housing, Annual Survey of Operations*, Vol III--Installation Performance (U.S. Army Corps of Engineers Fiscal Year [FY] 76, FY79, FY84, and FY86.)

### Oxygen Trim Systems for Central Plant Boilers

An oxygen trim system was installed on each of nine central plant boilers. The trim system was designed to control the amount of combustion air introduced into the boiler by measuring flue gas oxygen concentration and automatically adjusting air dampers to achieve optimum combustion.

### Boiler Plant No. 4 Summer Shutdown

Plant No. 4 produces HTHW for heating, and chilled water for cooling. Although demand for HTHW is relatively low during the summer months, the system is still operated at the same temperature. Because piping for both networks is run through the same system of concrete trenches, heat transfer to the chilled water piping causes a significant increase in the load on the central chiller. To alleviate this problem, separate self-contained boilers were installed at each of the barracks complexes that require hot water during the summer months. This allows the central boiler to be turned off for a significant portion of the year, and eliminates the problem of heat transfer from one pipe network to the other.

### Reduction in Outdoor Air Ventilation

Measurements of the rate of outdoor air flow were taken in 249 air-handling systems and the dampers were adjusted on those systems that were not in compliance with the most recent American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) standards on outdoor air requirements.

### Replacement of Inefficient Electrical Motors

Operating efficiency measurements were taken on 201 motors used to power fans in HVAC systems. Motors that were operating inefficiently were replaced with properly sized high-efficiency motors.

### Dispensary Chiller Installation

The dispensary was provided with chilled water from central plant No. 4. Because of the nature and use of the building, the air-conditioning system operates earlier in the spring and later in autumn than other buildings served by the plant. As a result, for 30 to 90 days each year, the central plant may operate solely to accommodate the requirements of the dispensary. During these periods, the plant (which is sized for 1,700 tons of cooling) is operating for two 12.3-ton air-conditioning units. Plant efficiency at such light loads is very low. A 30-ton packaged chiller was installed at the dispensary to alleviate this inefficiency.

### Combustion Optimization of Gas-fired Heating Equipment

Combustion air intake was measured and adjusted on 204 furnaces and hot water heaters ranging in size from 50,000 to 250,000 Btu/hr.

## Infiltration Study of Family Housing

A study was conducted to determine the energy savings that could be realized by reducing infiltration of outdoor air in family housing units. Thirty housing units of different construction types were analyzed; six were selected for infiltration reduction. Retrofits such as window caulking and door weatherstripping were performed. Based on estimates of energy savings, it was determined that the energy savings did not justify the cost of performing such retrofits on an installation-wide basis.

## Computer Decision Support for Energy Management

USA-CERL developed an energy information management system that maintains annual consumption data for each building located at the installation. The program uses input from the DEIS II report, post meter readings, boiler logs, and utility bills. Actual consumption is used for known buildings; linear regression equations are used to estimate consumption for buildings that are not metered.

## Radio Control of Exterior Lighting

Radio controlled switches were purchased for use with the existing FM radio control system at Fort McClellan. As opposed to common photosensitive switches that turn streetlamps on at dusk, the 250 radio controlled switches allow the lamps to be turned on and off as needed.

Note: The following projects were investigated but not implemented due to only modest gains in performance.

## Central Plant No. 4 Cooling Towers

Facility personnel at Fort McClellan have experienced difficulty in maintaining sufficiently cool chilled water to satisfy the air-conditioning loads on central plant No. 4. An investigation of a suspected cooling tower problem was performed. A solution was proposed but not implemented due to the modest gains in performance.

## Hospital Plant Investigation

An investigation similar to the project listed above was performed for the hospital plant with the same outcome.

## Central Plant No. 4 HTHW Reset

The objective of this measure was to reduce the temperature of HTHW leaving central plant No. 4 while still satisfying all the loads on the system. Recent modifications to the circulating pumps, and the modest savings due to the summer shutdown rendered the project uneconomical.

### Central Plant No. 2 Pressure Reset

A means to maintain the hospital steam pressure at 50 psig by controlling boiler pressure settings at central plant No. 2 was investigated. It was determined that to satisfy all loads on the system, pressure reducing stations required modification (see the project titled Steam Pressure Reduction in the District Heating System).

### Central Plant No. 1 Automatic Blowdown

An investigation into minimizing the amount of boiler water lost during blowdown was performed. It was determined that the small energy savings would not justify the cost to implement.

# 3 DISTRICT HEATING

## Energy Metering

To identify energy conservation opportunities in buildings served by district steam heating, condensate meters were installed to measure the amount of steam, and hence the energy, used by each building. The meters provide valuable information on building steam consumption. Of primary concern was correlating energy use before reducing the steam pressure with energy use after pressure reduction.

At Fort McClellan, condensed steam is collected from converters, air heating coils, and cooking and sterilization equipment in a receiving vessel at each building before being pumped back to the central plant. The pump is operated by a float valve within the receiving vessel. A positive displacement, nonresettable, totalizing flowmeter was installed between the receiving vessel and the pump at each building served by district steam. Through the use of flowmeters, condensate (and correspondingly pounds of steam) used by individual buildings was accounted for before being returned to the central plant.

If meter readings are taken on a regular basis, correlations of steam demand and such variables as occupancy and weather conditions can be developed. These correlations can be used to predict energy demand for future periods. Comparisons of predicted and actual steam consumption can also indicate when a building is using more steam than it has in the past, which may indicate problems such as failed traps, steam leaks, or poor energy management procedures. Tables 1 through 3 contain spreadsheets of the sampled data and compare the actual steam use with a predicted value (calculated in million British thermal units [MBtu]). The predicted values were calculated using regression formulas from *USA-CERL Technical Report* (TR) E-143.[2] The steam use for building 2220, for example, is considerably higher than the predicted value. Hence, by simply reading the meters on a regular basis, a flag is raised when consumption is excessive.

Perhaps the most important information that can be gained through the use of metering is the overall efficiency of the district heating system. Records of fuel consumption at the plant can be compared to the heating energy consumed by all the buildings supplied by the system to determine the cost of supplying energy. This information is useful when making decisions on maintenance and replacement of heating equipment.

The Army does not currently have a firm policy regarding metering of energy use in buildings served by central plant heating systems. However, due to the number of such systems in existence throughout the Army, and the amount of information which can be gained, installing metering equipment in future construction as well as in existing buildings should be considered.

[2]Benjamin J. Sliwinski, D. Leverenz, L. Windingland, and A. R. Mech, *Fixed Facilities Energy Consumption Investigation: Data Analysis*, Technical Report E-143/ADA060-513 (U.S. Army Construction Engineering Research Laboratory [USA-CERL], February 1979).

## Table 1

**Metered Data for Actual Steam Use vs Predicted Use, 1 March Through 28 March**

| BLDG | DESCRIPTION | SQ FT | DATE 1 | READING 1 | DATE 2 | READING 2 | TIME DIFF. (DAYS) | MBTU USED | MBTU CALC |
|---|---|---|---|---|---|---|---|---|---|
| 292 | Hospital | - | 01 MAR | 0 | 28 MAR | 65687 | 27 | 541.26 | - |
| 294 | Hosp. Plant | - | 01 MAR | 0 | 28 MAR | 89040 | 27 | 733.69 | - |
| 295 | Barracks | 11964 | 01 MAR | 1 | 28 MAR | 15604 | 27 | 128.57 | 98.59 |
| 1020 | Barracks | 36416 | 01 MAR | 0 | 28 MAR | 21658 | 27 | 178.46 | 300.09 |
| 1021 | Barracks | 36416 | 01 MAR | 0 | 28 MAR | 46337 | 27 | 381.82 | 300.09 |
| 1022 | Barracks | 36416 | 01 MAR | 0 | 28 MAR | 42806 | 27 | 352.72 | 300.09 |
| 1023 | Barracks | 36416 | 01 MAR | 0 | 28 MAR | 36315 | 27 | 299.24 | 300.09 |
| 1081 | ACAD (N-S) | 40164 | 01 MAR | 1066 | 28 MAR | 18280 | 27 | 141.84 | 307.95 |
| 1081 | ACAD (E-W) | 40165 | 01 MAR | 1958 | 28 MAR | 10947 | 27 | 74.07 | 307.96 |
| 2203 | Clothing Store | 11817 | 01 MAR | 0 | 28 MAR | 19784 | 27 | 163.02 | 117.25 |
| 2213 | Hilltop Club | - | 01 MAR | 2464 | 28 MAR | 27775 | 27 | 208.56 | - |
| 2220 | Barracks | 36416 | 01 MAR | 885 | 27 MAR | 74725 | 26 | 608.44 | 295.34 |
| 2221 | Barracks | 36416 | 01 MAR | 0 | 27 MAR | 0 | 26 | 0.00 | 295.34 |
| 2223 | Barracks | 36416 | 01 MAR | 12 | 27 MAR | 33642 | 26 | 277.11 | 295.34 |
| 2224 | Barracks | 36416 | 01 MAR | 458 | 27 MAR | 42331 | 26 | 345.03 | 295.34 |
| 2225 | Barracks | 36416 | 01 MAR | 0 | 27 MAR | 11639 | 26 | 95.91 | 295.34 |
| 2227 | Barracks | 36416 | 01 MAR | 0 | 27 MAR | 0 | 26 | 0.00 | 295.34 |
| 2275 | Barracks | 25147 | 01 MAR | 8924 | 27 MAR | 29803 | 26 | 172.04 | 203.94 |
| 2276 | Barracks | 25147 | 01 MAR | 15 | 27 MAR | 21206 | 26 | 174.61 | 203.94 |
| 2277 | Barracks | 25147 | 01 MAR | 0 | 27 MAR | 35311 | 26 | 290.96 | 203.94 |
| 2281 | Barracks | 46334 | 01 MAR | 27 | 28 MAR | 28232 | 27 | 232.41 | 355.26 |
| 2290 | Dispensary | 8876 | 01 MAR | 4546 | 28 MAR | 14735 | 27 | 83.96 | 124.62 |
| 2293 | Chapel | 8072 | 01 MAR | 188 | 28 MAR | 18774 | 27 | 153.15 | 66.52 |
| 2299 | Chem. Museum | 23739 | 01 MAR | 17 | 28 MAR | 36392 | 27 | 299.73 | 182.01 |
| 3130 | Barracks | - | 01 MAR | 0 | 28 MAR | 39517 | 27 | 325.62 | - |
| 3133 | Barracks | - | 01 MAR | 0 | 28 MAR | 0 | 27 | 0.00 | - |
| 3134 | Barracks | - | 01 MAR | 0 | 28 MAR | 0 | 27 | 0.00 | - |
| 3135 | Barracks | - | 01 MAR | 122 | 28 MAR | 9001 | 27 | 73.16 | - |
| 3136 | Hirise barracks | - | 01 MAR | 5118 | 28 MAR | 27530 | 27 | 184.67 | - |
| 3191 | Training | - | 01 MAR | 10 | 28 MAR | 3691 | 27 | 30.33 | - |

HDD FOR THIS PERIOD: 295

MBTU USED IS THE BTU'S RELEASED FROM SATURATED STEAM AT 350 °F TO CONDENSATE AT 200 °F

MBTU CALC IS THE PREDICTED FUEL INPUT CALCULATED USING REGRESSION FORMULAE FROM USA-CERL TR E-143/ADA060-513

## Table 2

**Metered Data for Actual Steam Use vs Predicted Use, 1 March Through 26 April**

| BLDG | DESCRIPTION | SQ FT | DATE 1 | READING 1 | DATE 2 | READING 2 | TIME DIFF. (DAYS) | MBTU USED | MBTU CALC |
|---|---|---|---|---|---|---|---|---|---|
| 292 | Hospital | - | 01 MAR | 0 | 26 APR | 132120 | 56 | 1088.67 | - |
| 294 | Hosp. Plant | - | 01 MAR | 0 | 26 APR | 164710 | 56 | 1357.21 | - |
| 295 | Barracks | 11964 | 01 MAR | 1 | 26 APR | 33256 | 56 | 274.02 | 192.65 |
| 1020 | Barracks | 36416 | 01 MAR | 0 | 26 APR | 50366 | 56 | 415.02 | 586.39 |
| 1021 | Barracks | 36416 | 01 MAR | 0 | 26 APR | 84241 | 56 | 694.15 | 586.39 |
| 1022 | Barracks | 36416 | 01 MAR | 0 | 26 APR | 87867 | 56 | 724.02 | 586.39 |
| 1023 | Barracks | 36416 | 01 MAR | 0 | 26 APR | 60844 | 56 | 501.35 | 586.39 |
| 1081 | ACAD (N-S) | 40164 | 01 MAR | 1066 | 26 APR | 33292 | 56 | 265.54 | 591.59 |
| 1081 | ACAD (E-W) | 40165 | 01 MAR | 1958 | 26 APR | 35245 | 56 | 274.28 | 591.60 |
| 2203 | Clothing Store | 11817 | 01 MAR | 0 | 26 APR | 31420 | 56 | 258.90 | 234.12 |
| 2213 | Hilltop Club | - | 01 MAR | 2464 | 26 APR | 38357 | 56 | 295.76 | - |
| 2220 | Barracks | 36416 | 01 MAR | 885 | 26 APR | 102968 | 56 | 841.16 | 586.39 |
| 2221 | Barracks | 36416 | 01 MAR | 0 | 26 APR | 98407 | 56 | 810.87 | 586.39 |
| 2223 | Barracks | 36416 | 01 MAR | 12 | 26 APR | 66711 | 56 | 549.60 | 586.39 |
| 2224 | Barracks | 36416 | 01 MAR | 458 | 26 APR | 68895 | 56 | 563.92 | 586.39 |
| 2225 | Barracks | 36416 | 01 MAR | 0 | 26 APR | 46240 | 56 | 381.02 | 586.39 |
| 2227 | Barracks | 36416 | 01 MAR | 0 | 26 APR | - | 56 | - | 586.39 |
| 2275 | Barracks | 25147 | 01 MAR | 8924 | 26 APR | 50252 | 56 | 340.54 | 404.93 |
| 2276 | Barracks | 25147 | 01 MAR | 15 | 26 APR | 36497 | 56 | 300.61 | 404.93 |
| 2277 | Barracks | 25147 | 01 MAR | 0 | 26 APR | 73493 | 56 | 605.58 | 404.93 |
| 2281 | Barracks | 46334 | 01 MAR | 27 | 26 APR | 58442 | 56 | 481.34 | 682.47 |
| 2290 | Dispensary | 8876 | 01 MAR | 4546 | 26 APR | 25468 | 56 | 172.40 | 245.13 |
| 2293 | Chapel | 8072 | 01 MAR | 188 | 26 APR | 35113 | 56 | 287.78 | 129.98 |
| 2299 | Chem. Museum | 23739 | 01 MAR | 17 | 26 APR | 55755 | 56 | 459.28 | 349.66 |
| 3130 | Barracks | - | 01 MAR | 0 | 26 APR | 74110 | 56 | 610.67 | - |
| 3133 | Barracks | - | 01 MAR | 0 | 26 APR | 5105 | 56 | 42.07 | - |
| 3134 | Barracks | - | 01 MAR | 0 | 26 APR | 8920 | 56 | 73.50 | - |
| 3135 | Barracks | - | 01 MAR | 122 | 26 APR | 18358 | 56 | 150.26 | - |
| 3136 | Hirise barracks | - | 01 MAR | 5118 | 26 APR | 49132 | 56 | 362.68 | - |
| 3191 | Training | - | 01 MAR | 10 | 26 APR | 10811 | 56 | 89.00 | - |

HDD FOR THIS PERIOD: 550

MBTU USED IS THE BTU'S RELEASED FROM SATURATED STEAM AT 350 °F TO CONDENSATE AT 200 °F

MBTU CALC IS THE PREDICTED FUEL INPUT CALCULATED USING REGRESSION FORMULAE FROM USA-CERL TR E-143/ADA060-513

## Table 3

**Metered Data for Actual Steam Use vs Predicted Use, 1 March Through 16 May**

| BLDG | DESCRIPTION | SQ FT | DATE 1 | READING 1 | DATE 2 | READING 2 | TIME DIFF. (DAYS) | MBTU USED | MBTU CALC |
|---|---|---|---|---|---|---|---|---|---|
| 292 | Hospital | - | 01 MAR | 0 | 16 MAY | 169890 | 77 | 1399.89 | - |
| 294 | Hosp. Plant | - | 01 MAR | 0 | 16 MAY | 203160 | 77 | 1674.04 | - |
| 295 | Barracks | 11964 | 01 MAR | 1 | 16 MAY | 34224 | 77 | 282.00 | 225.44 |
| 1020 | Barracks | 36416 | 01 MAR | 0 | 16 MAY | 59497 | 77 | 490.26 | 686.19 |
| 1021 | Barracks | 36416 | 01 MAR | 0 | 16 MAY | 88694 | 77 | 730.84 | 686.19 |
| 1022 | Barracks | 36416 | 01 MAR | 0 | 16 MAY | 103510 | 77 | 852.92 | 686.19 |
| 1023 | Barracks | 36416 | 01 MAR | 0 | 16 MAY | 61089 | 77 | 503.37 | 686.19 |
| 1081 | ACAD (N-S) | 40164 | 01 MAR | 1066 | 16 MAY | 36005 | 77 | 287.90 | 656.29 |
| 1081 | ACAD (E-W) | 40165 | 01 MAR | 1958 | 16 MAY | 47322 | 77 | 373.80 | 656.30 |
| 2203 | Clothing Store | 11817 | 01 MAR | 0 | 16 MAY | 38516 | 77 | 317.37 | 291.64 |
| 2213 | Hilltop Club | - | 01 MAR | 2464 | 16 MAY | 40481 | 77 | 313.26 | - |
| 2220 | Barracks | 36416 | 01 MAR | 885 | 16 MAY | 111461 | 77 | 911.15 | 686.19 |
| 2221 | Barracks | 36416 | 01 MAR | 0 | 16 MAY | 100279 | 77 | 826.30 | 686.19 |
| 2223 | Barracks | 36416 | 01 MAR | 12 | 16 MAY | 71137 | 77 | 586.07 | 686.19 |
| 2224 | Barracks | 36416 | 01 MAR | 458 | 16 MAY | 78660 | 77 | 644.38 | 686.19 |
| 2225 | Barracks | 36416 | 01 MAR | 0 | 16 MAY | 66225 | 77 | 545.69 | 686.19 |
| 2227 | Barracks | 36416 | 01 MAR | 0 | 16 MAY | - | 77 | 0.00 | 686.19 |
| 2275 | Barracks | 25147 | 01 MAR | 8924 | 16 MAY | 54625 | 77 | 376.58 | 473.84 |
| 2276 | Barracks | 25147 | 01 MAR | 15 | 16 MAY | 40909 | 77 | 336.97 | 473.84 |
| 2277 | Barracks | 25147 | 01 MAR | 0 | 16 MAY | 93788 | 77 | 772.81 | 473.84 |
| 2281 | Barracks | 46334 | 01 MAR | 27 | 16 MAY | 75316 | 77 | 620.38 | 757.11 |
| 2290 | Dispensary | 8876 | 01 MAR | 4546 | 16 MAY | 39291 | 77 | 286.30 | 292.55 |
| 2293 | Chapel | 8072 | 01 MAR | 188 | 16 MAY | 35816 | 77 | 293.57 | 152.10 |
| 2299 | Chem. Museum | 23739 | 01 MAR | 17 | 16 MAY | 60683 | 77 | 499.89 | 387.90 |
| 3130 | Barracks | - | 01 MAR | 0 | 16 MAY | 91440 | 77 | 753.47 | - |
| 3133 | Barracks | - | 01 MAR | 0 | 16 MAY | 5971 | 77 | 49.20 | - |
| 3134 | Barracks | - | 01 MAR | 0 | 16 MAY | 10220 | 77 | 84.21 | - |
| 3135 | Barracks | - | 01 MAR | 122 | 16 MAY | 22364 | 77 | 183.27 | - |
| 3136 | Hirise barracks | - | 01 MAR | 5118 | 16 MAY | 52717 | 77 | 392.22 | - |
| 3191 | Training | - | 01 MAR | 10 | 16 MAY | 11660 | 77 | 96.00 | - |

HDD FOR THIS PERIOD: 550

MBTU USED IS THE BTU'S RELEASED FROM SATURATED STEAM AT 350 °F TO CONDENSATE AT 200 °F

MBTU CALC IS THE PREDICTED FUEL INPUT CALCULATED USING REGRESSION FORMULAE FROM USA-CERL TR E-143/ADA060-513

**Steam Pressure Reduction in the District Heating System**

Fort McClellan has four natural gas operated boiler plants that provide for the heating needs of various facilities. Three plants produce steam at 100 pounds per square inch gauge (psig). Plant No. 4 is a HTHW facility operating separately from the others. An analysis was performed to identify the major causes for heat loss in the steam district heating system.[3] On the basis of this analysis, the annual efficiency of the system was determined to be 53.4 percent. The objective of this project is to reduce system conduction and live steam losses by implementing a reduction in steam pressure at the central boiler plants while maintaining the original heating capacity of the system.

One measure to improve fuel-use efficiency and also reduce operating costs is to reduce the boiler operating pressure where possible. At many Army facilities, the boiler operating pressure is maintained at a constant rate throughout the year regardless of demand. Ideally, the steam temperature/pressure is optimized so that reliability and customer satisfaction are maintained while heat transfer and live steam losses are minimized. The intent is to supply facilities and steam equipment with steam at the lowest pressure that will adequately handle the demand and equipment specifications. Typically, high pressure steam exiting the plant loses pressure in transit due to heat conduction, steam leaks, customer demand, and friction. When steam enters a building, it is throttled to the operating pressure of the equipment. For many applications, including domestic hot water production and building heat, this pressure is usually between 15 to 20 psig. Because this operating pressure is low, it is reasonable to consider supplying steam at a low pressure by operating the boiler at a reduced temperature. The following technical considerations were used to evaluate this change:

- Determine the reduction in energy consumption
- Ensure that facility and steam equipment demands are met
- Ensure that steam velocities are well within specified standards
- Calculate the pressure drops at the reduced sendout pressure such that item 2 is satisfied (this includes the hospital which has a demand for 50 psig steam)
- Survey all steam traps and evaluate their performance at the reduced pressure
- Maintain boiler operation within the manufacturers specifications
- Conduct a cost analysis and determine the payback period.

*Energy Savings*

USA-CERL researchers determined that the pressure in the steam distribution system at Fort McClellan could be safely reduced without loss of steam availability. A reduction in steam pressure from 100 to 70 psig corresponds to a drop in temperature from 337 to 316 °F. (See Table 4. Note: pressure is absolute pressure.) This results in a 9 percent reduction in heat transfer losses. Similarly, a reduction in pressure from 100 to 50 psig yields a 14 percent reduction in heat transfer losses. Moreover, a considerable

[3]ETN, *Reduction of Steam Pressure in District Heating Systems*; G. D. Pine, *Analysis of a Small Steam District Heating System at Fort McClellan, Alabama* (Oak Ridge National Laboratory, July 1984).

Table 4

Saturated Steam Temperature Table

| | | Specific Volume | | Internal Energy | | | Enthalpy | | |
|---|---|---|---|---|---|---|---|---|---|
| Temp (°F) $T$ | Pressure Lbf/Sq In $P$ | Sat. Liquid $v_f$ | Sat. Vapor $v_y$ | Sat. Liquid $u_f$ | Evap. $u_{fy}$ | Sat. Vapor $u_y$ | Sat. Liquid $h_f$ | Evap. $h_{fy}$ | Sat. Vapor $h_y$ |
| 290 | 57.53 | 0.017352 | 7.467 | 259.25 | 838.5 | 1097.7 | 259.44 | 917.8 | 1177.2 |
| 300 | 66.98 | 0.017448 | 6.472 | 269.52 | 830.5 | 1100.0 | 269.73 | 910.4 | 1180.2 |
| 310 | 77.64 | 0.017548 | 5.632 | 279.81 | 822.3 | 1102.1 | 280.06 | 903.0 | 1183.0 |
| 320 | 89.60 | 0.017652 | 4.919 | 290.14 | 814.1 | 1104.2 | 290.43 | 895.3 | 1185.8 |
| 330 | 103.00 | 0.017760 | 4.312 | 300.51 | 805.7 | 1106.2 | 300.84 | 887.5 | 1188.4 |
| 340 | 117.93 | 0.017872 | 3.792 | 310.91 | 797.1 | 1108.0 | 311.30 | 879.5 | 1190.8 |
| 350 | 134.53 | 0.017988 | 3.346 | 321.35 | 788.4 | 1109.8 | 321.80 | 871.3 | 1193.1 |

savings is expected where steam is being lost via leaks in valves, pipes, and faulty steam traps. Although these losses are difficult to quantify, the magnitude may be quite significant as many leaks were observed in mechanical rooms and along the main piping. When leaks occur, live steam losses are proportional to the square root of the pressure difference.

$$Q = AC\sqrt{2gh} \qquad \text{[Eq 1]}$$

where Q = steam, loss (lbs/hr)
A = area of opening
C = discharge coefficient
g = gravity constant
h = pressure drop

Hence, a reduction in pressure from 100 to 50 psig decreases live steam losses by 29 percent.

*Steam Demand*

Each building on the district heating system is equipped with a pressure reducing station which has a known capacity at a given inlet pressure. For a reduced inlet pressure, the steam capacity will also decrease. Maintaining existing maximum steam capacities for each building when the pressure is reduced involves replacing pressure reducing valves (PRVs) with valves which have a larger flow coefficient (Cv). Consider the case where steam pressure is being reduced from 100 to 50 psig. Referring to the saturated steam table (Table 4), the enthalpy (heat content) of dry saturated steam is about 1,190 Btu/lb and 1,180 Btu/lb for 100 psig and 50 psig, respectively. Hence, the maximum heat delivered to the equipment is 99 percent of the original maximum value although the temperature is approximately 40 °F lower.

To determine the new appropriate Cv, the Cv of the existing valve must be obtained (available from the valve manufacturer). Use Equation 2 to determine the existing steam capacity.[4]

$$Q = 2.1(Cv)\sqrt{P1 - P2}\sqrt{P1 + P2} \quad \text{[Eq 2]}$$

where Q = steam, flow (lbs/hr, capacity)
P1 = inlet pressure (pounds per square inch area [psia])
P2 = outlet pressure (psia)

Once the maximum steam capacity is determined for existing conditions, the new Cv is found by using Equation 2 with the new operating pressures. As a precautionary measure, the maximum capacities calculated in this manner were compared with condensate measurements at each building. (See the discussion of the project titled Metering of Steam Heating Energy for more details.) The calculated maximum capacities using the valve Cv proved most conservative and were therefore employed.

*Steam Velocities*

When pressure is reduced in a system where demand is unchanged, an associated increase in steam velocities must occur to balance the decrease in steam density. The calculation is straightforward:

$$V = 3.05(Q)(v)/(d^2) \quad \text{[Eq 3]}$$

where V = velocity (fpm)
Q = steam, flow (lb/hr)
v = specific volume ($ft^3$/lb)
d = internal pipe diameter (in.)

The current maximum steam velocities are approximately 4,000 ft/min. For a reduced sendout pressure of 50 psig and a pipe diameter of 6 in., the new velocities are approximately 6,000 to 7,000 feet per minute (fpm) which is well below the standard suggested steam velocity of 8,000 to 12,000 fpm with a maximum of 15,000 fpm as given by ASHRAE Fundamentals.

*Pressure Drop*

A consequence of increased steam velocities throughout the piping is increased pressure drops due to friction. An analysis of the pressure-flow relationships for the piping network connected to boiler plants No. 2 and No. 3 was performed to determine whether the capacity of the system is adequate at reduced pressures.[5] At the system peak loads (outdoor temperature of 13 °F) the steam pressure at the hospital was calculated to be 28 psig for a sendout pressure of 50 psig and 40 psig for a sendout pressure of 60 psig. Because the hospital is equipped with a sterilizer which requires 50 psig, this drop is too great. At a boiler operating pressure of 70 psig, the hospital will maintain 50 to 55 psig throughout the peak demand. For boiler plant No. 1, there is no demand for high pressure steam and there are no links with the other boiler plants. Hence, an operating pressure below 75 psig can meet the steam demands. As a precautionary measure, a

[4] *Flow of Fluids Through Valves, Fittings, and Pipe*, Technical Paper No. 410M (Crane Co., 1977).
[5] G. D. Pine.

separate analysis was performed where the desired facility pressure was set and the boiler pressure was back calculated to determine the setting. Using Equation 4, the Reynold's number is calculated as follows:

$$Re = 354(Q)/(d)(u) \qquad [Eq\ 4]$$

where
Re = Reynolds number
Q = steam, flow (kg/hr)
d = diameter (mm)
u = absolute viscosity (centipoise)

Equation 5 is the Darcy formula[6] which is used to determine the pressure drop per 100 meters of pipeline.

$$\Delta P = 62530(f)(Q^2)(v)/(d^5) \qquad [Eq\ 5]$$

where
$\Delta P$ = pressure drop per 100 meters
Q = flow (kg/hr)
v = specific volume ($m^3$/kg)
f = friction factor

The friction factor for clean steel pipe can be obtained from Figure 1.

Of course the pressure drop due to friction is only part of the total drop. So to predict the overall drop, heat transfer and live steam losses need to be estimated. For this project, the area of major concern was the line that transports steam to the hospital. For this line, the overall drop for the boiler set at 100 psig has been measured. Hence, the frictional loss may be subtracted to determine the other contributions. For example, the hospital is located 2,000 ft from the boiler. Using Equation 4 and Figure 1, the friction factor is found to be .016. Hence, from Equation 5 the drop due to friction is 7 psig. In the winter, the overall drop is measured at 20 psig. Therefore, 13 psi is due to conduction, leaks, and other customers. The conservative approach is to assume this 13 psi unchanged at the reduced pressure/temperature. The problem then becomes one of finding the initial pressure so that the hospital receives 63 psig. Using a boiler operating pressure of 75 psig, the drop due to friction is 10.2 psi which adds to the 13 psi to yield 52.8 psig at the hospital. The boilers in plants No. 2 and No. 3 should be operated at 80 psig during the peak season to ensure a margin of safety. A sendout pressure of 80 psig will maintain the hospital pressure at 58.3 psig during the peak loads.

*Steam Traps*

Steam trap capacities are decreased by the system changes. A reduced inlet pressure results in a reduced capacity. A survey of the steam traps throughout the installation was performed to determine whether the traps had sufficient capacity at reduced operating pressure. Along the steam main lines, the hardest working trap (with the most frequent cycles) is in the vault nearest the hospital. During the summer, the trap operates at approximately 25 percent of maximum capacity. If the trap proves to be undersized at the lower pressure, a larger trap should be used. Of the approximately 125 traps located in Fort McClellan's mechanical rooms, 3 high pressure traps were blowing steam when inspected. None of the these traps appear to be undersized for the reduced inlet pressure.

[6] *Flow of Fluids Through Valves, Fittings, and Pipe.*

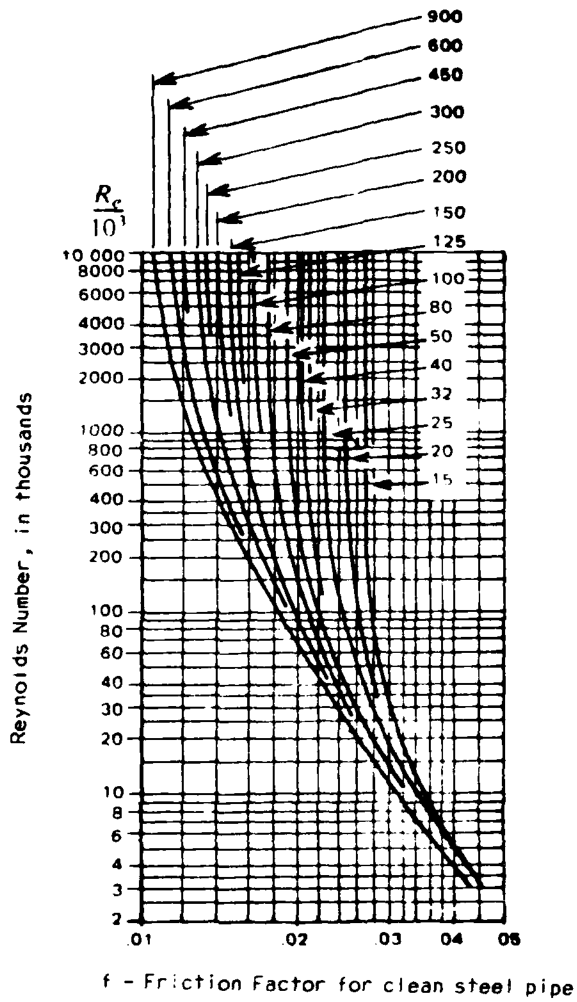


**Figure 1. Reynolds number vs friction factor.**

*Boiler Operation*

A project which involves reducing the boiler pressure depends on the boiler's f' ·xibility. For example, many coal-fired boilers cannot function at reduced pressures without reducing tube life expectancy. At Fort McClellan, the boilers are natural gas fired. The boiler manufacturers were consulted on this project and they approved the reduced operating pressure. Reducing operating pressure does reduce boiler capacity. However, this proves advantageous because the boilers do not currently operate above 70 percent full load as they should for optimum efficiency. Boiler pressure should be reduced very gradually (1 psi every 10 minutes). The pressure reducing valves should then be adjusted such that the downstream pressure is at the desired point (an adjustment screw is located on the pilot regulator).

*Savings*

Information gathered through condensate metering, and the predicted values for building steam demand, provided USA-CERL researchers with an accurate means of computing energy savings. The efficiency of the district heating system (delivered steam/steam entering the pipes at the boiler plants) is approximately 53.0 percent. In 1982, the fuel use in boiler plants No. 1, No. 2, and No. 3 was approximately 180,000 MBtu. Assuming a boiler efficiency of 75 percent, an estimated amount of 135,000 MBtu enters the steam distribution system in the form of steam. Of this, 47 percent (63,450 MBtu) is lost mainly due to conduction between the pipes and the ground and live steam losses from leaks and faulty equipment. Natural gas has a unit cost of $10.86 per MBtu. This results in losses of $689,067 annually.

Table 5 indicates the savings possible from operating the boilers at the recommended settings. The savings of $66,104 were calculated as explained in the previous section. The project cost was $100,000, hence, a simple payback of 1.5 years was achieved. Other agencies with district heating systems operating at much higher than end-use pressures should consider performing a detailed engineering analysis to determine the feasibility of this type of retrofit.

## Steam Vault Drainage

Central boiler plants No. 2 and No. 3 serve 24 buildings. The steam system includes 17 vaults that contain service valves, traps, tees, and other steam service equipment. Typically, the vaults are equipped with drainage pumps that keep the vault dry when it rains or equipment leaks. Random inspection of these vaults revealed that they were often filled with water and many times the piping was in contact with the water. In some cases the vaults were not equipped with pumps and in other cases the pumps (steam powered) were inoperative. Draining this water can save approximately $4,600.00 per year. This estimate is based on the decreased heat loss from the pipes. Water in the vaults was assumed to be heated to 150 °F. The estimated heat loss from a bare, 6-in. steam pipe carrying 338 °F steam and covered by 150 °F water is 50,000 Btu/hr-ft (this could be as high as 150,000 Btu/hr-ft depending on the assumed heat transfer mechanism). The heat loss is nearly 60 times that of dry bare pipe, and is equivalent to the heat transfer in 1,000 feet of dry pipe with 4 in. of calcium silicate insulation. A general recommendation to keep the vaults dry is applicable to any Army installation.

**Table 5**

**Savings Due to Proper Boiler Operation**

| Season | Annual Savings | Plant No. 1 | Plants No. 2 and No. 3 |
|---|---|---|---|
| Winter | $66,104. | 80 psig | 50 psig |
| Summer | | 75 psig | 50 psig |

## Oxygen Trim Systems for Central Boiler Plants

An oxygen trim system allows minor adjustments to the amount of air flowing into the boiler combustion chamber to provide near-optimum air flow at all firing rates. The trim system should function only to adjust for the random variations in the combustion process that take place because of changes in air temperature, humidity, or fuel composition.

The items discussed below should help avoid the problems researchers encountered when applying oxygen trim controls to the boilers at Fort McClellan. The controls must be carefully planned, set up, and calibrated to result in significant energy savings.

### *Check Damper Travel*

Damper travel and linkage adjustments may vary from one trim system to the next. Some systems replace part of the damper linkage with a variable-length member; others impose an angular rotation directly to the dampers. Whatever method is used, it is important to check that the travel imposed by the trim controller on the dampers is not excessive. The trim controller should cause the dampers to move only about 5 percent of their total sweep at low fire, and about 10 percent at high fire. Any more travel can cause flame failure in certain boilers due to too much or too little combustion air.

### *Interface With Existing Controls*

Take care when interfacing the trim system with existing boiler controls. Many boilers have limit switches at the maximum or minimum damper position, or at both. During the purge cycle, these switches make contact to signal to the master controller that the dampers have opened or closed. If the limit switches cannot be met, the controller will not allow the boiler to start. It is important to ensure that the trim system does not impose any offset to the dampers which would interfere with the control sequence. This precaution is necessary even on boilers which do not have limit switches, because the dampers may need to be fully closed to light the flame.

### *Protect Against Failure*

If the trim system fails, it may freeze in a certain position which would not allow the limit switches to be met. If this happens, the boiler may have to be taken offline. Therefore, although it may be difficult to achieve, set up the trim system in such a way that the failure of one component will at worst cause the boiler to run as it did before the trim system was installed.

On one particular boiler, the oxygen trim system had an installed linkage that allowed the lower limit switch to be met regardless of the trim position. In other words, when the oxygen trim system moves from maximum to minimum, there is no corresponding damper travel. However, the upper limit switch could only make contact when the trim controller was at its maximum position. This was not a problem when the system was functioning correctly, but it was clear that the boiler would not start in the event the trim system froze in some other position. To solve this problem, the high limit switch was replaced by a spring-loaded switch that allowed enough travel so that the upper limit switch could also be met regardless of the trim position.

*Control Chamber Pressure*

Pressure in the combustion chamber should always be kept slightly negative to avoid releasing toxic combustion products into the boiler room. Many old boilers (often the best candidates for oxygen trim) are leaky, and due to the slight vacuum, will draw a certain amount of air from the boiler room into the combustion chamber. This air will eventually affect the oxygen concentration seen by the sensor in the stack and will cause slightly higher readings.

The excess air decreases the efficiency of the boiler, because it is heated and sent out the stack. However, the excess air will not affect the performance of the trim system so long as the oxygen concentration remains the same as it was during calibration of the trim controller. But if the boiler pressure controller is not accurate enough, and if the flue gas oxygen concentration changes significantly, the trim controller will attempt to decrease oxygen concentrations by closing the dampers. Since the excess air does not take part in combustion, closing the dampers at this point causes a decrease in efficiency, and may also cause increased carbon monoxide production. Therefore, unless pressure in the combustion chamber is accurately maintained, an oxygen trim system may not be effective.

*Simulate Controller Operation*

A major concern with any oxygen trim system is the operation of the controller itself. It is difficult to assess the performance of a trim system simply by watching it operate at various loads. If the system is stable, there is very little outward difference between a trim controller that is set up properly and one that is not. Ideally, one would like to see what happens when oxygen concentrations in the flue gas go above or below the setpoint. Do the dampers open or close by a reasonable amount? Is there instability? Is the deviation from the setpoint acceptable?

Because such conditions are difficult to create except in the laboratory, researchers used an accurate calibration/setup procedure which involved using simulated inputs to the controller. On one controller, oxygen content was sensed by a 4 to 20 milliampere (mA) signal from the oxygen sensor. During calibration, the sensor was disconnected and replaced with a variable current source. Likewise, the controller sensed the fuel valve position by way of a variable resistor. This resistor was replaced by another variable resistor placed in the vicinity of the controller so that it could be adjusted easily during calibration. This allowed the boiler operation to be simulated over a wide range of operating conditions without affecting the boiler itself. These simulated inputs were also useful for assessing calibration drift.

Oxygen trim systems require a greater investment of time and effort than one might expect. The correct setup procedure includes installing the appropriate linkages, proper interfacing with existing controls, accurate setup of the controller, and periodic calibration of the oxygen sensor and the controller itself. And as always, verification is necessary to ensure the success of the investment. If all equipment is functioning correctly, however, an oxygen trim system can improve combustion control. Before implementing a new oxygen trim system, installation personnel should be thoroughly familiar with the system.

**Boiler Plant No. 4 Summer Shutdown**

This investigation showed the technical feasibility and implementation of shutting down the HTHW system in boiler plant No. 4 for up to 8 months a year. The shutdown coincides with a seasonal shift in heating equipment operation whereby local hot water or steam generators will supply the summer hot water needs of each building on the system. The reason for shutting down plant No. 4 is twofold. First, on-site heating can save money and second, the same system of concrete trenches which contain the hot water piping also contain the chilled water pipes. The heat transfer to the chilled water lines is causing a significant increase in the load on the central chiller.

Currently, the system is operated continuously all year. During periods when space heating is not required in the buildings being served, domestic hot water and kitchen steam loads are met with the central system. The proposed alternative operating scheme would entail using new heat generating equipment and laying up the central system equipment for these periods. This layup will impose conditions on the system which had not been experienced on a routine basis.

To identify and evaluate these new service conditions and their impact on facility maintenance and operating requirements, the layup was conducted in the following three stages which are discussed below.

1. Cooldown

2. Idle Period, and

3. Startup.

*Cooldown*

The boilers should be brought to low fire and shut off according to manufacturer's recommended procedures. It will be necessary to keep the boiler and system circulation pumps operating until the system water temperature is stabilized at ambient conditions. During the cooling process, makeup water must be provided as necessary to maintain water level in the expansion tank to keep surfaces from dry corrosion and leaking.

Water chemistry must be monitored and controlled within recommended limits during and after the cooldown. Once the system temperature has stabilized at ambient levels, normal equipment shutdown procedures can be continued. A positive pressure should be maintained on the system, although the magnitude of that pressure can be reduced from 240 psig at the expansion tank to (approximately) 25 psig.

As the system cools, the piping system components also contract. To what extent will this contraction cause leaks to develop? It is anticipated that some new leaks will be created. However, they are not expected to be a significant problem.

As the system cools, what will be the effect on leaks already present in the system? The current flashing of HTHW into steam upon exiting from the piping will be eliminated. When the system pressure is reduced, however, the net effect will be a decrease in leakage rate.

As the system water cools, the solubility of air in the water increases. Will this cause corrosion problems during the shutdown? Water chemical treatment can be adjusted during the shutdown period to offset the increased solubility of air in the water.

With proper treatment (probably two to three times the current concentration of sulfites), corrosion problems can be prevented.

After the system has been cooled, what pressure should be maintained? Lowering the pressure of system water would decrease the rate of flow through existing leaks and diminish the tendency to create new leaks. Two factors, however, limit the extent to which the pressure may be reduced:

1. The system must be kept at a pressure greater than atmospheric to ensure against leakage of air into the system, and

2. The system must be kept at a sufficiently high pressure to prevent cavitation in the circulating pumps. This requirement can be satisfied for low temperature water (less than 100 °F) with any pressure above atmospheric.

*Idle Period*

Will prolonged idleness have detrimental effects on system components, including boilers, controls, pumps, and piping? Problems can be minimized by using some or all of the preventive measures listed below.

- Follow the boiler manufacturers' published standard wet-layup procedures for temporary storage of their equipment.

- Keep the circuits energized even while the equipment is off to minimize condensation that could cause relays in control cabinets to stick.

- Operate the pumps for short periods at intervals throughout the layup season.

- Inhibit corrosion in the piping system by properly modifying normal treatment procedures according to water treatment specialist recommendations.

- Keep the fireside dry by using lime or another desiccant.

Will water treatment lose homogeneity if the system pumps are not operated, and will this lead to localized corrosion problems? During the layup, it is necessary to periodically sample and maintain system water chemistry according to the recommendations of the water treatment specialist. The pumps should be operated for short periods of time (several minutes) every 2 weeks throughout the layup period, unless the water treatment specialist requires more frequent operation to maintain treatment specifications.

Central plant boilers are currently used to keep No. 4 fuel oil stores heated so they will be available in the event of an emergency curtailment of natural gas. If the central plant is shut down, will there be adequate fuel heating capacity to maintain proper emergency backup conditions? The plant is equipped with electric heater sets to backup the hot water supplied units. These can be used in the event of an emergency when the central plant is shut down.

While the system is idle, locate and repair existing leaks. It may be desirable to temporarily increase system pressure to normal operating levels so leaks can be easily located.

### *Startup*

As the system is brought up to operating temperature, the piping system components will expand. Although some new leaks will be created, they are not expected to be significant. The system should be started before the buildings need space heat to allow time to repair any leaks.

Start up may introduce air bubbles in the system. Vent valves should be operated throughout the system to relieve any air released from solution during heating.

After the system pressure is back to normal (240 psig at the expansion tank) and the system pumps are back in operation, the boilers should be restarted according to manufacturer's guidelines. The rate at which system water is brought up to temperature should not exceed guidelines provided by the boiler manufacturer. Once the system is up to temperature, the local hot water (HW) generators at each building can be secured after shifting over to central system supply.

## Conclusions and Cost Analysis

The investigation identified no significant technical constraint which would limit the ability to shutdown the HTHW system at Fort McClellan annually. Typically, district heating systems operate with an efficiency between 50 and 75 percent (Btu used by the customer/Btu entering the pipes less the return Btu). This energy represents the savings associated with use of the on-site boilers. Hence, for the months of interest, approximately 37,500 MBtu enter the system and assuming a district efficiency of 75 percent results in losses of 9,375 MBtu annually. A cost of $10.86 per MBtu results in savings of $101,812.50 annually.

In addition to significant energy and operating cost savings, a regular system shutdown will afford the opportunity for preventive and corrective maintenance. It is noted, however, that a potential increase in the number of system leaks may occur as a result of the thermal cycles on joints. These are expected to be relatively minor, and the additional maintenance required for repair will be more than offset by the savings of not operating the system during the extended shutdown period. Finally, the heat transfer from the HTHW piping to the chilled water piping will be alleviated.

## 4 HVAC SYSTEMS

### Reduction in Outdoor Air Ventilation

The purpose of this project is to conserve energy by measuring and reducing the rate of forced outdoor air ventilation in large air-handling systems. This information applies to all Army facilities with air-handling systems that introduce outdoor air into the building environment for ventilation and/or free cooling.

Most air handlers are designed to introduce some outdoor air into the building for ventilation. While this air is required for health and comfort reasons, it can also cause significant energy expense during the heating or cooling season by contributing to the load. Since minimum ventilation requirements have decreased over the years, it is likely that many air handlers are introducing more outdoor air than necessary. Therefore, it is imperative from the standpoint of energy conservation that air be introduced at the minimum requirement established by the American Society of Heating, Refrigeration, and Air-conditioning Engineers (ASHRAE) Standard 62-1981.[7]

The purpose of the ASHRAE Standard is "to provide healthful and comfortable indoor environments by using materials and methods that optimize efficiency of energy utilization." The publication emphasizes the effect of ventilation on energy consumption by listing building types and ventilation rates per area and number of occupants. The Corps of Engineers has used this standard to formulate minimum ventilation requirements for Army buildings.

The researchers performed a survey on all air-handling equipment at the installation to determine if outdoor air reduction was feasible. The survey identified the size and type of system (e.g., draw-through, multi-zone, variable air volume [VAV]) and suitable locations for temperature flow measurements. On the basis of this survey, 249 air handlers were selected for additional analysis.

For each unit, the number of occupants served by the system; temperature and flow rate of supply, return, and outdoor air; temperature rise across the fan; pressure rise across the fan; and the electrical characteristics of the fan including power, power factor, voltage, and current were determined. Measurements were cross-checked and the data were organized for additional conservation work. Of the 249 identified air handlers, 28 were introducing more outdoor air than required. Damper adjustments resulted in a reduction of 17,835 cubic feet per minute (cfm), or about 11 percent of the total outdoor air intake of the air handlers.

The annual savings for the heating season can be estimated by the following expression:

$$\text{Heating Savings} = k(R)(HDD)(C) \qquad \text{[Eq 6]}$$

where R = reduction in outdoor air ventilation rate (17,835 cfm)
HDD = heating degree days (2551)
C = delivered fuel cost per MBtu ($10.86)
k = a constant (2.592 E-05)

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[7]ASHRAE Standard 62-1981, *Ventilation for Acceptable Indoor Air Quality* (ASHRAE, 1981).

On this basis, the project will save an estimated $12,807 in annual heating costs. Note that this does not account for efficiency of the steam or HTHW distribution system or efficiency of local boilers if used.

For the cooling season, the calculations are less straightforward. However, using a method similar to Equation 6, the potential savings can be estimated by the following expression:

$$\text{Cooling Savings} = k(R)(CDD)(C)/(COP) \qquad \text{[Eq 7]}$$

where R = reduction in outdoor air ventilation rate (17,835 cfm)
CDD = cooling degree days to a base of 80 °F (600)
C = electrical cost per kWh ($0.05)
COP = coefficient of performance (1.5)
k = a constant (7.60 E-03)

Using Equation 7, the potential savings for the cooling season are estimated to be $2,711. The total annual heating and cooling savings are $15,235.

Since the total project cost was $29,000, the simple payback period is less than 2 years. Outdoor air reduction to conserve energy at this installation was first suggested in a 1979 EEAP study. Based on their estimates of outdoor air flow rates, and adjustment in accordance with ASHRAE Standard 62-1973 (an earlier edition of the standard used in this study), the firm suggested a simple payback period of about 1 month. Since the current standard allows less outdoor air, thereby saving additional energy at virtually the same cost, one would expect a shorter payback period using the more recent standards. While the actual 2 year payback is reasonable, it does point out the value of accurate field measurements as opposed to blanket estimates.

### Replacement of Inefficient Electric Motors

The electric motors that power fans in HVAC systems are a significant source of electrical power consumption. Proper matching of load and motor power is important, because an oversized motor consumes significantly more power than a motor which is properly sized for its application.

Of 201 motors surveyed at Fort McClellan, 68 were sufficiently oversized to warrant replacement. Some of the motors had operating efficiencies as low as 50 percent. Properly sized integral horsepower motors manufactured today operate at efficiencies of 75 to 95 percent, depending on the design and size.

IEEE Standard 112-1984[8] specifies several methods for determining motor efficiency. These methods however, are intended for laboratory testing. They are much too involved, if not impossible, for field implementation. To facilitate the measurements, USA-CERL developed a method of approximating motor efficiency using an optical tachometer. A small strip of reflective tape is placed on the motor shaft, and the motor is placed in operation. The tachometer is aimed at the reflective tape, and the rotative

[8]IEEE Standard 112-1984, *Standard Test Procedures for Polyphase Induction Motors and Generators* (Institute of Electronics and Electrical Engineers, 1984).

speed is read on the instrument. The formulas developed below approximate the load and efficiency of the motor using this simple speed measurement and the motor nameplate data.

Because the majority of motors used in HVAC applications are of the NEMA type B design, equations were developed for that design only. Similar equations for other motor designs could be developed using the same principles.

Development of an equation for motor efficiency first requires a discussion of fractional load (FL) and its relationship to motor speed. This relationship, which is nearly linear except during startup, involves three quantities: the revolutions per minute synchronous ($rpm_{sync}$), rpm operating speed ($rpm_{meas}$), and rpm nameplate ($rpm_{np}$). $Rpm_{sync}$ is a function of line frequency and motor design (number of poles); $rpm_{meas}$ is the operating speed of the motor, which should be measured when the motor is under the maximum system load; $rpm_{np}$ is the rpm stamped on the motor's nameplate and indicates the speed at which it operates when delivering its rated horsepower ($hp_{np}$). Fractional load (in percent) can be calculated using these parameters as follows:

$$FL(\%) = 100(rpm_{sync} - rpm_{meas})/(rpm_{sync} - rpm_{np}) \qquad \text{[Eq 8]}$$

OR

$$FL = \frac{Bhp}{hp_{np}}$$

where Bhp = Brake horsepower (load delivered by motor).

As an example, consider a 20 hp motor with a nameplate rpm of 1,740, a synchronous speed of 1,800 rpm, and an operating speed measured at 1,780 rpm. The fractional load is calculated as 100(1800-1780)/(1800-1740) = 33.

Replacement motor size should be the smallest motor which will not incur a load greater than its nameplate rating (FL <= 100%). Motors are typically designed to withstand loads equal to 115 percent of their nameplate horsepower to provide for a margin of error.

It is important to confirm the Bhp when considering motor replacement. All motors in this study contained adjustable belt drives to spin centrifugal fans. This configuration allows simple adjustment to ensure that the replacement motor delivers the same Bhp as the original. Without this adjustment, the more efficient motor could actually consume more power than the original if it is delivering a greater Bhp. In the case of centrifugal fans, this would result in a larger airflow. Although true Bhp cannot be practically measured in the field, the objective is to rotate the fan at the same rpm as before. This can be checked with a tachometer and the current draw of the new motor can be checked to detect overloading.

Assuming that adjustments can be made after replacement of the motor to ensure equivalent Bhp, the following relationship gives an approximation for replacement motor size:

$$Bhp = (FL_r)\left(hp_{np_r}\right) = (FL_o)\left(hp_{np_o}\right) \qquad \text{[Eq 9]}$$

The subscripts r and o refer to replacement and original, respectively.

Equation 9 can be rearranged to give a relation for the required horsepower of the replacement motor ($HPN_r$) in terms of the nameplate horsepower and the fractional load of the original motor:

$$hp_{np_r} = (FL_o\%)\left(hp_{np_o}\right)/100\% \qquad \text{[Eq 10]}$$

Continuing the example above, the required horsepower of the replacement motor is (33%)(20 hp)/100% = 6.6 hp. The next largest motor size commonly available is 7.5 hp. Rearranging Equation 9 and substituting the known variables yields the fractional load of the replacement motor: 88%(33 X 20/7.5).

With the fractional loads determined, the efficiencies can be approximated from performance curves. The relationship of efficiency versus fractional load for a typical NEMA design B motor can be approximated by the following equation (derived from a curve fit to the NEMA design B parameters):

$$E\ motor = 16.1 + 170(FL) - 100(FL)^2 \qquad \text{[Eq 11]}$$

In the example, the fractional loads of the original and replacement motors are 33 and 88 percent, respectively. Substituting these values into Equation 11 yields efficiencies of 61 percent for the original motor and 88 percent for the replacement motor. Calculation of the energy savings is performed as follows:

Change in hp demand = $Bhp[1/E_o - 1/E_r] = FL_o \times HP_{No}[1/E_o - 1/E_r]$

Change in kW demand = $(0.746\ kW/hp) \times FL_o \times HP_{npo}[1/E_o - 1/E_r]$

Assuming 0.05$/kWh and 8,760 hours of operation per year:

$saved/year = $0.05/kWh X 8,760hr/yr X .746kW/hp X .33% X 20hp[1/.61% - 1/.88%]

= $1084/year

Assuming the cost of a 7.5 hp motor to be approximately $300 and installation costs to be $200, the simple payback is less than 5 months.

Because the efficiency characteristics of standard induction electric motors can vary several percent, the full load efficiency of the replacement motor should always be specified. The following data should be collected from each motor at the time of the electrical or rotational speed measurements, since it must be known to select a replacement motor:

- Nameplate horsepower
- Rpm
- Amperage and voltage requirements
- Starting code
- Frame number

- Type and number of belts
- Fan and motor pulley diameters, and distance between pulley centers
- Information on the overload protection equipment, specifically amperage of circuit breaker, fuse, and thermal overload.

Kilowatt measurements indicated that 104 of the 201 motors surveyed were sufficiently underloaded to result in a payback of 5 years or less. Rpm measurements confirmed that 78 of those 104 warranted replacement. Various reasons prohibited replacement of 10 motors: size, cost, and enclosure type. Replacement of 68 motors resulted in a 17 percent decrease in electrical power demand.

**Dispensary Chiller Installation**

The dispensary is currently provided with chilled water and/or hot water to handle space cooling and heating requirements by central plant No. 4. Because of the nature and use of the building, it has sometimes been necessary to keep the dispensary air-conditioning system operating longer into the autumn than for other buildings served by the plant. Similarly, it has occasionally been necessary to start the air-conditioning system sooner in the spring than for other buildings served by the plant.

As a result, for 15 to 45 days on each end of the cooling season, the plant may be operated solely to accommodate the requirements of the dispensary. Thus for these periods, the plant (which is sized for 1,700 tons of cooling) is operated for just two 12.3-ton air-conditioning systems. Plant efficiency at such light loads is very low.

A 30-ton packaged chiller unit was installed at the dispensary to handle the loads during the "swing" seasons, thereby eliminating the need for extended operation of the central plant at extremely light loads.

To assess the chiller size requirements, plans of the building were obtained and reviewed, and a survey of the site was conducted. The results of these efforts and subsequent analyses are discussed below.

*Analysis*

A survey of the dispensary and a review of the construction drawings indicate that the building is quite well insulated and has a low percentage of glass area on external walls. Furthermore, there is a rather high level of lighting, and during certain hours of the day, a high occupancy. In general terms, these observations reflect a tendency for the building air-conditioning loads to be relatively insensitive to outdoor air temperatures. This tendency explains the observed occurrence of air-conditioning loads early in the season before outdoor air temperatures get too high, and late in the season after they have dropped off substantially from summer peaks.

Based on a design load calculation, the design cooling load of the dispensary is 33 tons. This calculation was based on an outdoor air temperature of 97 °F db (dry bulb), 77 °F wb (wet bulb), and a room setpoint of 72 °F db, 50 percent relative humidity.

The program was rerun with an 86 °F/72 °F db/wb outdoor air condition, resulting in a peak load of 28.7 tons. This load exceeds the combined design capacity of the two air-handling units (AHUs) serving the building. This provides some explanation for the comments received from building occupants that room conditions were not satisfactory during extreme weather (in cooling or heating modes).

It was observed during the survey that hot water was being supplied to the building through both the chilled water and hot water piping. Thus both decks of the multizone AHUs were being used as hot decks. Conversations with facility personnel indicated that piping modifications had been made to help alleviate the inadequate capacity of the heating and cooling systems. With the modification, both AHU coils are used for heating during the winter, and both for cooling during the summer. Based on the comments from the occupants, however, these actions have not resolved the problem.

National Oceanic and Atmospheric Administration (NOAA) weather data for Birmingham show that temperature extremes have occurred during the swing months (Table 6). Therefore, a system sized to accommodate the potential peak loads during the swing seasons would be capable of meeting design summer load conditions. Because of this situation, the size and energy saving potential of a separate packaged chiller installation at the dispensary was complicated. Whereas an undoubted efficiency improvement would occur with the installation of a fully-sized system, the ability of the system to meet currently unsatisfied loads would diminish the net energy savings.

As a fair basis for comparing a modified system to the base case system, a constant level of cooling output was assumed for the swing seasons. NOAA weather data indicates an average outdoor temperature during these seasons of 57.3 °F. Translating this into an estimated building cooling load of 16 tons, and assuming a 60-day swing season length, about 9,600 ton-hours of cooling establishes the output baseline.

To provide an average of 16 tons of cooling to the dispensary from the central plant, a distribution system loss of at least 95 tons would be incurred (estimated on the basis of a one degree water temperature rise from the plant to the dispensary, and another one degree rise from the dispensary back to the plant).

The plant chiller would therefore be required to operate at about 111 tons. At that load, its input power requirement would be about 166 kW. In addition, plant auxiliaries including chilled water, tower pumps, and tower fans, would add a constant power consumption of about 128 kW. Therefore the total plant operating input power would be 294 kW. Operating at this level for 600 hours a year, the resulting energy usage is 176,400 kWh. A new packaged chiller, on the other hand, would require approximately 1.3 kW per ton, including auxiliaries. Supplying 9,600 ton-hours during the swing season would therefore require only 12,480 kWh; a savings of 163,920 kWh. At $0.05 per kWh, this translates into annual energy cost savings of $8,196.

*Implemented Solution*

Although the peak load of the building was calculated to be 33 tons, the installed capacity of the existing AHUs is only about 25 tons. This capacity is inadequate. Therefore, for this chiller replacement, a 30-ton unit was selected. This size would enable a more satisfactory approach to meeting peak load requirements should the AHUs be upgraded to appropriate capacities in the future. The shortage of 3 tons could be tolerated more easily during the swing seasons than the summer because of the relative infrequency of peak load occurrences. The chiller replacement was $37,500 with a simple payback of 4.5 years. Other installations experiencing a large discrepancy between central plant chiller sizing and online demand during nonpeak seasons should consider individual chiller units.

**Table 6**

**Birmingham Peak Temperature Data**

| Month | Peak Temperature (°F) |
|---|---|
| March | 87 |
| April | 90 |
| May | 99 |
| September | 100 |
| October | 94 |
| November | 84 |

## 5 FAMILY HOUSING

### Combustion Optimization of Gas-fired Heating Equipment

Because of their relatively small size (50,000 to 250,000 Btu/hr), forced-air furnaces and hot water heaters of the type normally installed in barracks, mess halls, offices, and family housing units are often overlooked as candidates for installation-wide energy conservation measures. However, when the number of these heating units in operation at any installation is considered, their combined energy consumption is significant. At Fort McClellan, the combined heating output for all residential-size heating equipment is of the same magnitude as that produced by a central boiler plant.

Since the rapid increase in fuel prices experienced in the mid-1970's, numerous add-on devices and modifications for residential gas-fired heating equipment have been marketed as energy-saving measures. These devices include flue gas heat extractors, recirculators, and automatic vent dampers. While the concepts involved in the design of such devices are valid, very few operate effectively in the field. In a study of 20 such devices performed by the New York State Energy Research and Development Authority, it was determined that the most reliable and effective method of reducing fuel consumption in residential heating equipment is furnace derating. Furnace derating involves installing flue gas and vent dampers and adjusting the firing rate of the appliance. The effect is to reduce the output of the heating device. A reduced output is allowable because most furnaces are oversized for their particular applications. While typical efficiency improvements with furnace derating are approximately 8 percent, major modifications to existing equipment would have been too costly to perform in this research. However, projects to adjust the firing rate, or optimize combustion were inexpensive and had a rapid payback.

Previously, the only way to optimize combustion in heating equipment of this type was by sight. Although it is possible to set the proper combustion rate by observing the color, intensity, and other characteristics of the flame, this method provides no information about the actual operating efficiency either before or after the adjustment. Orsat meters have been used to measure concentrations of oxygen and carbon monoxide in the flue gas, but this method is very time consuming and requires sufficient training and experience.

Recently however, several electronic micro-based devices have been marketed which allow rapid measurement of flue gas composition. These devices are portable and can be used in the field with a minimum of training. Costs range from $1,000 to $2,000 (1986 dollars), depending on the model and number of options present. Measurements of flue gas temperature, oxygen and carbon monoxide concentrations, and efficiency are taken either continuously or as a sample, and are displayed in the proper units on a digital display. Using this instrument, heating equipment can be adjusted, and the energy savings quantified, in less than 30 min per unit. This is a considerable improvement over methods which were used in the past and makes combustion optimization a very cost-effective, rapid payback project.

Combustion in any heating device occurs when fuel and air are ignited. The most efficient combustion (stoichiometric combustion) occurs when the correct proportion of fuel and air are burned. The combustion products consist only of water vapor and carbon dioxide. If there is not enough oxygen present, the combustion process is incomplete. Some of the fuel does not burn and is lost through the exhaust gases. Some of the fuel that does burn forms carbon monoxide rather than carbon dioxide. Incomplete combustion is not only less efficient, but also presents a danger to the environment and possibly to the occupants of the heated space.

When too much air is mixed with the fuel, a portion of the heat of combustion is used to heat the excess air. Since this excess air is ejected through the flue pipe with the rest of the combustion products, some of the heat of combustion is wasted. Because of the inefficiency of the combustion process, all heating devices require some excess air and carbon monoxide will always be present in the flue gas.

Most furnaces and water heaters contain a sleeve or disk that is used to adjust the amount of air introduced into the combustion process. It is relatively easy to fabricate an adjustment device for equipment that does not contain one.

An additional source of energy loss in residential heaters is air leakage after the combustion process. Because of the high temperature of the exhaust products, a stack effect is created and the exhaust rises into the flue and out to the atmosphere. This movement creates a partial vacuum in the heater, which causes additional air to be drawn into it. If the heater is located in the heated space of a building, the effect is to draw in heated air and eject it to the outdoors. This occurs more often with furnaces than water heaters because of differences in the system designs.

Although oxygen in the air is the driving force of the combustion process, flue gas carbon monoxide concentration is a more reliable predictor of combustion efficiency. This is because, as stated above, some excess air will always leak into the system and appear in the flue gas. Figure 2 shows a curve of typical carbon monoxide and oxygen measurements taken as a function of the position of the adjustable air gate in a furnace. As the air gate is closed off, the oxygen concentration tends to decrease and approach a bottom limit where the air leakage after the combustion process becomes the dominant source of oxygen in the flue gas. Measuring the oxygen concentration at this point provides little information on the efficiency of combustion.

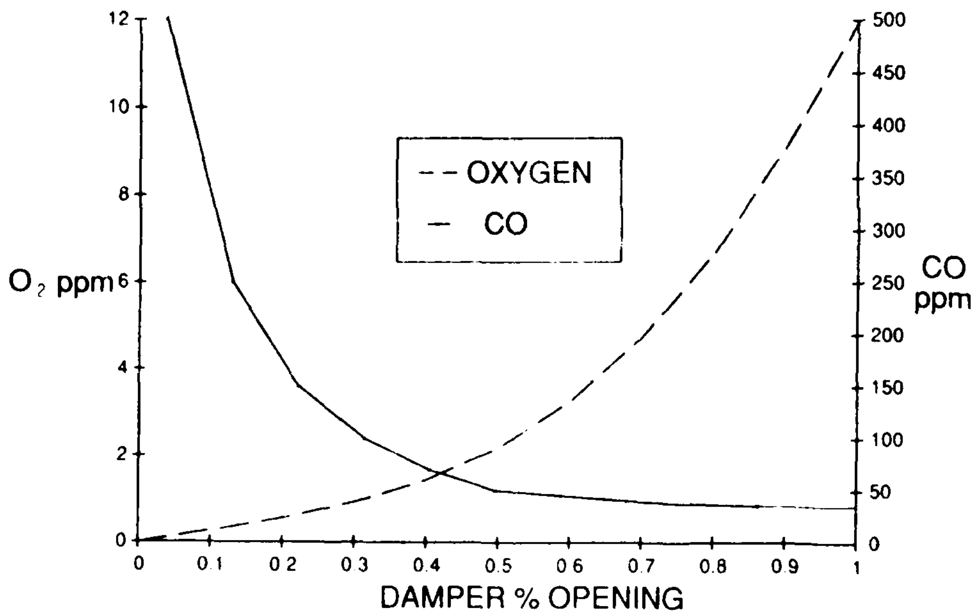


**Figure 2. Oxygen and carbon monoxide vs damper position.**

On the other hand, air leakage after the combustion process has very little effect on carbon monoxide concentrations in the flue gas. Figure 2 also shows that up to a certain point, the carbon monoxide concentration increases only slightly as the air damper is closed. This is due mostly to a decrease in dilution by the other combustion gases. However, as the air damper is closed further, the concentration of carbon monoxide begins to rise very rapidly because there is insufficient oxygen present in the combustion process. At this point, oxygen concentrations in the flue gas may still be quite high, but are the result of air leakage after the flame. Therefore, measuring carbon monoxide is more important than measuring oxygen to determine the combustion efficiency. Optimum combustion occurs just before the point at which carbon monoxide concentration begins to increase rapidly. Generally, a carbon monoxide concentration of 200 to 250 parts per million (ppm) indicates optimal combustion in the heating equipment normally encountered at military and civilian installations.

Measuring and adjusting the combustion air will be different for each type of device. However, USA-CERL researchers developed a general method which can be used as a guideline for measuring and adjusting most heating equipment. This method is discussed in the following paragraphs.

1. All heating devices contain a safety relief which prevents the combustion products from reentering the combustion area if the flue becomes blocked. On hot water heaters, the relief often consists of a vent in the flue pipe near the outlet of the device. On forced air furnaces, a vent is placed above the burners. During normal operation, ambient air is drawn into the safety relief due to the stack effect of the combustion gases. Because this air will affect the oxygen and carbon monoxide concentrations measured in the flue, all measurements should be taken before the safety relief. If this is not possible, the safety relief should be temporarily blocked using aluminum foil or some other inexpensive fireproof material, and a small hole should be drilled in the flue pipe to accomodate the measurement probe.

2. To ensure a steady state of operation, run the system for at least 15 minutes before the first measurements are made. This may require setting the thermostat to a higher temperature to ensure continuous operation of the system. Usually, the carbon monoxide and oxygen concentrations will stabilize after a few minutes, but flue gas temperature requires additional time to reach a steady state.

3. After a steady state has been reached, measure and record the efficiency, temperature, oxygen concentration, and carbon monoxide concentration.

4. During adjustment, monitor the carbon monoxide concentration. If the initial concentration was above 250 ppm, begin decreasing the combustion air by adjusting the air gate or other device on the burner until a point is reached where carbon monoxide concentration begins to increase rapidly. At this point, the readings will be unstable and will vary widely. Begin increasing combustion air slowly until the carbon monoxide concentration stabilizes between 200 to 250 ppm. If the initial reading was below 250 ppm, increase the combustion air until a reading between 200 to 250 ppm is achieved.

5. When adjustments are completed, measure the efficiency, temperature, oxygen concentration, and carbon monoxide concentration to quantify energy savings. If the oxygen concentration is above 10 percent after the adjustment, excess air may be entering the flue after the flame. Make sure the safety relief is blocked, check for other leaks, and repeat steps 2 through 4.

6. If temporary blocking of the safety relief was required, remove the blocking material when the adjustment and measurements are completed. If a hole was drilled in the flue pipe, cover it with aluminum furnace tape or furnace cement. Reset the thermostat to the original room temperature setting.

While this procedure applies to most heating equipment at military and civilian installations, some units may present special problems. For example, some heaters will produce high concentrations of both oxygen and carbon monoxide no matter what adjustments are made. This may be due to clogged or dirty heat exchanger surfaces. Periodic maintenance may reduce the concentration levels.

To demonstrate the effectiveness of combustion adjustment, 204 furnaces and hot water heaters at Fort McClellan were evaluated. These units had a combined heating capacity of 88,952,000 Btu/hr. The combustion efficiency of more than half of the units was improved, with efficiency increases of as much as 25 percent in some cases. The average improvement for all the units was 2.5 percent. Based on the average heating season and fuel costs for Fort McClellan, this project is estimated to have saved $16,000 with an investment of $17,000 in labor and equipment costs, hence a simple payback of 1.1 years was achieved.

### Infiltration Study of Family Housing

Infiltration of outdoor air in family housing is a major source of energy losses not only at Army installations, but in the private sector as well. Many local power utilities have developed programs for their consumers whereby company representatives will perform an energy analysis of the home and suggest ways to improve energy efficiency. These suggestions usually involve weather stripping outside doors, caulking windows, and other measures designed primarily to reduce outdoor air infiltration. Because of the relatively low cost and ease of installation, most of the conservation measures performed under such programs fall into the category of reduction of outdoor air infiltration.

To test the likelihood of energy conservation through reduction of infiltration in family housing units at Fort McClellan, 30 housing units (6 each of 5 different construction types) were tested using a standard fan pressurization/depressurization method. Since major leakage sites and problem areas were consistent within each specific construction type, corrective measures were applied to four representative units, and the reduction in the leakage rate documented. Using an infiltration model, this reduction in leakage rate was compared to the associated savings of heating energy.

A blower door apparatus was used to measure the airtightness of each housing unit. The apparatus consists of a variable-speed fan attached to a calibrated flow measuring nozzle. The fan is sealed to the door opening by a coated nylon fabric that is stretched across an adjustable frame. Differential pressure gauges are used to measure the pressure across the nozzle and the pressure in the house with respect to the outdoors.

The representative units selected for infiltration reduction were: a two-story, three-bedroom townhouse (1,195 sq ft); a two-story, two-bedroom townhouse (922 sq ft); a one-story, three-bedroom duplex (1,456 sq ft); and a one-story, two-bedroom duplex (1,103 sq ft). Corrective measures consisted of four retrofit packages. The first retrofit was to seal the interior door to the heater room closet (furnace and/or hot water heater). This was done only when a sufficient amount of combustible air was provided per the National Fire Protection Association code. Most of the housing units at Fort McClellan are equipped with a large attic fan that allows a considerable amount of cold air to infiltrate from the vented attic. The second retrofit was to seal the interior louvers to these fans. The third package was to seal air leakage sites associated with utility services (e.g., hot and cold water supplies, drains, electrical main breaker box, switches and receptacles, and heating ducts). The fourth retrofit package was to seal other air leakage sites in the building envelope (doors and frames, windows and frames, walls and baseboards, vents, kitchen cabinets, and attic hatch), which is the method of energy reduction most often suggested by utility companies.

*Heating Load Reduction*

The calculated reduction in heating load for each of the four construction types and each of the four retrofit packages is summarized in Table 7.

The results of the infiltration study show the average reduction in leakage area was 37 percent for the four units retrofitted, with the majority of the reduction as a result of retrofit package four. For approximately 3 hours labor and $60 in materials per unit, an average annual savings of $38 per unit can be recognized. This represents an approximate simple payback period of 3 years. It is anticipated that the retrofit would have to be performed at least every 2 years. Because this does not meet the requirement of rapid payback, an installation-wide infiltration reduction program was not implemented as part of this project. However, such a program may be applicable to installations with larger heating requirements.

**Table 7**

**Calculated Reduced Heating Loads**

| | 2St3Br Twnhse* | 2St2Br Twnhse** | 1St3Br Duplex*** | 1St2Br Duplex‡ |
|---|---|---|---|---|
| Original annual load(MMBtu) due to infiltration | 9.92 | 12.06 | 7.73 | 7.35 |
| Load after retrofit 1 | 9.46 | 11.31 | 7.34 | 6.02 |
| Load after retrofit 2 | 9.34 | 9.30 | 7.09 | 5.33 |
| Load after retrofit 3 | 8.86 | 9.91 | 6.41 | 4.94 |
| Load after retrofit 4 | 8.70 | 6.36 | 4.12 | 3.99 |
| Total reduction (MMBtu) | 1.22 | 5.70 | 3.61 | 3.36 |
| Dollar savings (based on energy cost of $10.86 per MBtu) | $13.24 | $61.90 | $39.20 | $36.49 |

*Two-story, three-bedroom townhouse (1,195 sq ft)
**Two-story, two-bedroom townhouse (922 sq ft)
***One-story, three-bedroom duplex (1,456 sq ft)
‡One-story, two-bedroom duplex (1,103 sq ft)

## 6 ENERGY MANAGEMENT

### Computer Decision Support for Energy Management

Fort McClellan has various sources of energy consumption information (oil, gas, coal, and electricity). Most of this information provides consumption data for the entire installation; it provides little information on energy performance of individual buildings. However, the information was not being used to analyze the installation energy consumption and determine if energy management could enhance energy conservation. If this information were organized, it could be used to evaluate conservation decisions.

USA-CERL researchers developed an energy information management system that maintains annual consumption data for each building located at the installation. The program uses input from DEIS II report, meter readings, boiler logs, and utility bills. The actual consumption is recorded for metered buildings; linear regression equations are used to estimate consumption for buildings that are not metered.

*System Overview*

The management system was developed in Lotus 1-2-3 for use on a DEC Rainbow personal computer. Lotus' macro language and data base functions were used extensively to create a user-friendly, menu-driven program.

The management system is divided into three major components with various subcomponents and interfaces. The building data base, consumption files, and analysis module interact to provide a comparison of consumption rates for building groups or time periods selected by the user.

The building data base is the most critical element in the system. It contains selected items from the installation real property inventory, including the building number, building category code, square footage, heating and cooling equipment, structural materials, year built, number of stories, and fuel used. Energy consumption estimates and all analysis procedures are based on this information.

The consumption files contain monthly consumption values for each facility. The files are listed in order of the facility number. Consumption values are computed monthly when the meter information becomes available. Degree day information (HDD and/or CDD) also in this file is used with linear equations to estimate consumption values. The estimates are adjusted to reflect actual use for metered buildings and are stored in fiscal files sorted by heating, cooling and electrical costs. The sum of these files is equal to the installation's total energy consumption.

The analysis module allows the user to compare consumption values against historical data (located in the fiscal files). The comparison can be based on individual building performance or facility group consumption.

*Building Data Base*

Each field within the building data base provides information used to calculate consumption or identify building characteristics that are used to define building groups.

The building's category code determines which linear regression parameters are used when estimating energy consumption. Parameters have been developed[9] for nine major building functions: family housing, troop housing, administration/training, dining, medical/dental, production/maintenance, fieldhouse/gym, commissary, and storage.

The regression equations are in the following form:

$$E_h \text{ (Btu/sq ft/day)} = a1 + b1 \times (HDD_d) \qquad \text{[Eq 12]}$$

$$E_e \text{ (kWh/sq ft/day)} = a2 + b2 \times (CDD_d) \qquad \text{[Eq 13]}$$

where $E_h$ = heating energy
$E_e$ = electric energy
$HDD_d$ = average daily Heating Degree Days
$CDD_d$ = average daily Cooling Degree Days

The values of a1 and a2 for each of the building functions represent the baseline energy consumption. Heating degree days (HDD) and cooling degree days (CDD) multiplied by their respective parameter (b1, b2), indicate how much addition energy use is required as a function of the weather.

All the regression equations are normalized to represent an energy use per square foot. It is necessary to multiply the respective equation by the square footage (obtained from the building data base) and the number of days in the month to obtain the monthly consumption estimate for a building.

The heating and cooling equipment information is used to identify sources of information and adjust consumption estimates. For example; consumption for buildings that get heating and cooling from a central plant is estimated using the regression equations but is adjusted to reflect actual use as recorded in the central plant boiler logs.

The regression equations were developed using meter information for a large group of buildings with the same function. The accuracy is limited when the equations are used to estimate the energy consumption (particularly electrical consumption) of one building. The structural material information identifies variations in construction that are linked to differences in infiltration and overall building thermal performance.

*Consumption Files*

When new meter information is obtained, CDD and HDD must be gathered for the metering period. This new data is then used to estimate comsumption rates for each building in the data base. Area meter information is used to check the accuracy of the estimates. As an example, if the estimates for a group of family housing units exceed the area meter consumption, each estimate is adjusted downward (based on a percent of energy consumption per square foot) until the estimate equals the metered consumption. Likewise, if the estimates are low, they are adjusted upward to reflect actual consumption. These adjustments are made whenever aggregate consumption information is available from central boiler production logs, family housing area metering, or main utility distribution meters.

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[9]Benjamin J. Sliwinski, et al. (1979).

The new consumption estimates are then appended to the appropriate fiscal files. The heating fiscal file contains Btu consumption for each building. The cooling fiscal file consists of the weather-dependent portion of the electrical regression equation, and the electrical fiscal file contains the electrical baseline. All values are converted to Btu so the total energy consumption can be determined.

*Analysis*

Once the consumption files have been developed, the user can select a specific building or a group of buildings for analysis. Up to five of the nine predefined groups can be chosen for analysis at one time.

The user can construct groups using any combination of the fields in the building data base. For example, the user could choose all masonry two-story buildings of greater than 6,000 sq ft. This group could be compared to all one-story masonry buildings. By analyzing different groups of buildings, the energy manager can determine how energy consumption varies among construction types.

By defining the time frame for analysis, the user can determine historical performance for the defined groups. New consumption values can be compared to information from previous months within the current fiscal year or to consumption totals for the same month of prior fiscal years. By analyzing energy use in this fashion, rapid changes in energy consumption (increase or decrease) would not be overlooked.

The final results are printed tables of facility numbers, consumption information for the defined time period, and a bar graph.

*Conclusion*

The energy management system allows the user to answer many questions that are critical to making decisions about energy conservation. The resulting program is relatively time consuming and requires consistent input and updating by the energy manager.

Accuracy of the estimates still presents a problem. Without field metering, estimates cannot be compared to actual consumption. To increase the usefulness of the energy management system, field verification is necessary and the regression equations need to be adjusted when applicable. Familiarity with the programming becomes essential.

Despite the limitations, the management system provides immediate feedback on how the energy use is distributed among the various different functions at an installation. Target areas for energy reduction can be readily identified and the extent of the feasibility of an energy conservation project (and the associated savings) can be determined.

By using this system, energy managers are able to perform consistent analysis on energy consumption and make correct decisions on the course to take in energy conservation.

## Radio Control of Exterior Lighting

USA-CERL researchers investigated using a radio control unit, currently used for load shedding, to reduce lighting during late night hours. A Military Police survey identified the effect of reduced lighting on safety and security and selected 250 lights located in parking or recreational areas for this project. Although lighting is required in these areas after dark, the levels can be safely reduced during low activity hours.

A radio frequency digital control switch was purchased for each light. The lights are controlled by the control switch in series with an optical sensor. The optical sensor controls daytime/nightime operation. Sunlight above a certain level of intensity will cause the relay to open (turning off the light). The digital control switch is used to override the optical sensor and turn the light off for low activity hours. The radio receiver in the switch operates on a frequency of 139.650 megahertz (MHz). A microprocessor monitors this frequency for a digital code. If a received digital code matches the programmed code, the switch is opened. The digital code is sent by a computer controlled transmitter that is part of the radio control system tied to the Energy Maintenance and Control System (EMCS). The system is designed so that if any component fails, the normal operation of the light will not be affected.

Three types of lights can be controlled at Fort McClellan. The majority are 175-watt mercury (Hg) vapor lamps, while the remainder are either 100- or 400-watt high pressure sodium (HPS) vapor lights. The amount of electrical energy saved by turning off these lights during low activity hours can easily be computed by using the following equation:

$$\$saved = (Watts/1000) \times \text{number of fixtures} \times \text{hours} \times \text{days} \times (cost/kWh) \qquad [Eq\ 14]$$

*Lighting Control Annual Savings*

The lights at Fort McClellan can be turned off for 5-1/2 hours, from 11:00 p.m. to 4:30 a.m. every day. The annual savings, based on an average cost of $0.05/kWh, are shown in Table 8. The digital control switches mounted on the lights cost $76 each and can be installed by an electrician in approximately 1 hour.

Facilities Engineers should investigate any radio control system already in use for possible expansion and application to lighting control. Most manufacturers of radio control systems sell a remote switch comparable in operation and cost to the digital control switch used in this project.

**Table 8**

**Lighting Control Annual Savings**

| Lamp Type | Dollars |
|---|---|
| 400-W HPS | 42.56 |
| 175-W Hg | 18.62 |
| 100-W HPS | 10.64 |

## 7 PROJECTS NOT IMPLEMENTED

The initial cost, payback period, and Army-wide applicability were the major factors used to determine project implementation. Although some proposed major modifications were too costly to implement during this research, projects were not automatically eliminated because they were expensive. Projects that offered the best combination of payback and energy savings received additional evaluation before implementation. In some cases, the results of this additional evaluation showed that the project was not feasible. These projects are discussed in this chapter.

### Central Plant No. 4 Cooling Towers

Facility personnel at Fort McClellan have experienced difficulty in maintaining sufficiently cool chilled water to satisfy the air-conditioning loads on central plant No. 4. The effectiveness of the two centrifugal chillers in the plant is limited on very hot days. The maximum compressor head pressures were being reached before the rated chilled water tonnage was reached. Facility personnel believe this is the result of inadequate performance by the cooling towers serving the two chillers.

*Analysis*

Temperatures and pressures throughout the plant were recorded hourly and measurements were made to determine the following operating parameters:

- Tower water flow rates
- Tower fan current draw
- Tower pump current draw
- Tower fan cfm.

Because the analysis was conducted during January when the chillers were not in use, full performance data could not be measured. However, the tower pumps and fans were run to collect data on the parameters listed above. Flow rates were measured by venturis installed in the two condenser water lines and were compared with flow indications from pressure drops across the two condensers. Additional data on equipment performance was obtained from plant logs. Table 9 is a collection of data taken from these logs and weather records showing conditions of chiller 1 during times of varying degrees of operational trouble.

*Data Interpretation*

Review of plant log data (Table 9) helped pinpoint the source of operational difficulties as failure of the chiller tower system in central plant No. 4 to produce adequate chilled water tonnages under normal operation during very hot days. Excerpts presented in the table show excessive head pressures exhibited by chiller 1 on a number of occasions when the chiller was only producing a 10 °F water temperature drop (12 °F is the design water temperature drop).

## Table 9

## Chiller No. 1 Log Data Excerpts

| HEAD (psig) PRESSURE | CONDENSER ENTERING WATER TEMPERATURE (°F) | EVAPORATOR ΔT (°F) | AMBIENT WET BULB (°F) | DATE | TIME |
|---|---|---|---|---|---|
| 165 = 110°F. | 92 | 10 | 74 | 8-5 | 1300 |
| 165 | 92 | 8 | 72 | " | 1500 |
| 165 | 91 | 10 | 78 | 8-21 | 1000 |
| 165 | 91 | 10 | 78 | " | 1100 |
| 165 | 91 | 10 | 79 | " | 1200 |
| 165 | 89 | 10 | 76 | " | 1700 |
| 165 | 89 | 10 | 76 | " | 1900 |
| 165 | 90 | 9 | 71 | 8-22 | 1900 |
| 165 | 90 | 10 | 78 | 8-23 | 1300 |
| 165 | 90 | 10 | 78 | 8-23 | 1500 |
| 165 | 92 | 11 | 78 | 8-24 | 1300 |
| AVG. 165 | 90.6 | 9.8 | 76.2 | | |
| 160 = 108°F. | 88 | 10 | 75 | 8-21 | 2100 |
| 160 | 87 | 10 | 71 | 8-22 | 2100 |
| 160 | 90 | 10 | 78 | 8-23 | 1100 |
| 160 | 90 | 10 | 78 | 8-24 | 1100 |
| 160 | 91 | 10 | 78 | 8-24 | 1200 |
| 160 | 87 | 10 | 75 | 8-4 | 1500 |
| 160 | 90 | 10 | 76 | 8-5 | 1100 |
| 160 | 90 | 10 | 72 | 8-5 | 1900 |
| AVG. 160 | 89.1 | 10 | 75.4 | | |
| 155 = 106°F | 86 | 10 | 78 | 8-24 | 1000 |
| 155 | 85 | 10 | 77 | 8-23 | 0900 |
| 155 | 84 | 10 | 74 | 8-23 | 2100 |
| 155 | 84 | 10 | 74 | 8-23 | 2200 |
| 155 | 87 | 11 | 73 | 8-1 | 1300 |
| 155 | 87 | 11 | 72 | 8-1 | 1500 |
| 155 | 87 | 10 | 73 | 8-1 | 1700 |
| 155 | 84 | 10 | 70 | 8-4 | 2400 |
| 155 | 87 | 10 | 70 | 8-3 | 2100 |
| 155 | 86 | 10 | 70 | 8-3 | 2200 |
| AVG. 155 | 85.7 | 10.2 | 73.1 | | |

The evaporator pressure drop data indicate that flows no greater than the design flow (1,700 gpm) were being circulated. Furthermore, the temperature of the system chilled water supply tends to increase during times of high head pressures, indicating a failure to satisfy loads. Therefore it may be concluded that the chiller's ability to meet loads is in fact being limited by the excessive head pressures.

The degree of excessive head pressure closely tracks the degree of excess in condenser entering water temperature (EWT) above the design maximum value of 85 °F. Also, rising condenser EWTs tend to track increasing ambient wet bulb air temperatures. These relationships suggest that the excessive head pressures in the chillers result from an inadequate heat rejection performance by the cooling towers. This failure in heat rejection performance could stem from improper tower water flow, inadequate tower fan loading, or insufficient physical tower capacity.

The measured flows of 2,900 and 3,300 gallons per minute (gpm) exceed the design tower flow of 2,550 gpm (based on 3 gpm/ton for 850-ton towers). The maximum recommended flows are 2,600 and 3,750 gpm for chillers 1 and 2, respectively. The measured flows approximate these limits, indicating more than sufficient flow through the condensers. Inadequate tower water flow was not the source of the problem.

The two-tower systems serving central plant No. 4 have three fans: one on the large eastern tower serving chiller 2, and one on each of the two towers serving chiller 1. The nameplate horsepower of the large tower fan is 50. The smaller fans are 30 hp. The current drawn by the large tower fan was 67 amps. The smaller fans drew 32 and 38 amps.

Assuming a power factor of 90 percent, and motor efficiencies of 85 percent, these current draws reflect output horsepowers calculated as follows:

$$hp = \frac{\frac{VA\sqrt{3}}{1000} \times 0.85 \times 0.90}{0.74} \qquad \text{[Eq 15]}$$

where V = volts
A = amperes.

Thus, 67 amps = 55 hp, 32 amps = 26 hp, and 38 amps = 31 hp. The proximity of the output horsepowers to the nameplate motor horsepowers indicates that the fan motors were operating near full load at the time of the measurements. Tower fan loading was not the problem source.

Therefore, the inadequate tower performance was attributed to the physical limitations of the tower itself as a heat exchanger. These cooling towers have had a history of unsatisfactory service since their installation in 1977.

To help quantify the extent of tower insufficiency, the tower performance was compared with the theoretical performance on the basis of information from the 1983 *ASHRAE Handbook*,[10] and the design specification of 850 tons of refrigeration with a cooling tower drop from 95 °F to 85 °F at 79 °F wet bulb temperature. The present tower performance level results in tower leaving water temperatures from 4 °F to 11 °F warmer than the theoretical design, depending on the ambient wet bulb temperature.

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[10] *ASHRAE Handbook, Equipment*, (American Society of Heating, Refrigerating, and Air-conditioning Engineers., Inc. [ASHRAE] 1988), pp 21.11 - 21.13.

*Solution Alternatives*

Four possible solutions to the existing cooling tower inadequacy were identified:

1. Demolish the existing tower structures (excluding pumps and basins) and install new towers

2. Replace the fill, fans, motors, gearboxes, and distribution basin of the existing towers, but keep the existing casing and collection basins, and reuse the existing pumps

3. Keep the existing towers and accessories as they are, but add a new tower to augment their performance. Pump water to the added tower in parallel with the existing towers

4. Same as 3, but pump water to the new tower in series with the existing towers.

5. If the tower is of an old design (having wood fill and splash bars) the installation of PVC fill, splash bars, and spray heads and a fan venturi (if needed) may provide an increase in tower performance or may provide the same performance with lower fan runtime.

Alternatives 1, 2, and 4 could be designed to meet the desired performance levels of heat rejection. However, alternatives 2 and 4 involve interfacing between new equipment and existing equipment. This may be difficult since the manufacturer of the existing equipment is no longer in business. The performance level of the retained equipment components in the new combined equipment configuration would be an undefined quantity. This would introduce some performance risk and the modifications manufacturer may be less willing to ensure the system's performance. These factors would also exist with alternative 3; however, unlike the other alternatives, it is doubtful that the desired performance level could be practically achieved with any substantial portion of the flow being sent through the existing towers. Therefore alternative 3 was evaluated on the basis of sizing the added tower to handle half the flow from each of the existing towers, and to individually have the capacity to reduce that water quantity temperature to required levels. With this sizing, it is estimated that the composite parallel tower configuration would return water to the condensers at approximately 89 °F, under full load, at 79 °F ambient wet bulb temperatures.

*Potential Energy Savings*

Compared to existing conditions, any of the four alternatives would improve the efficiency of the chiller plant. This efficiency improvement would result from three possible sources:

- Improved chiller Coefficient of Performance (C.O.P.) because of lower condenser EWTs

- Reduced cooling tower pumping horsepower because of reduced water flow rate requirements

- Potential reduced cooling tower fan energy because of improved tower heat transfer efficiency. (An extra tower means another fan. All fans need to be in operation to achieve increased performance at full load.)

Any of the alternatives would allow the current cooling load to be met at a reduced electrical consumption. Because the existing plant fails to meet the full load imposed on it, modifications which increase plant capacity would result in greater loads being satisfied. This would reduce the net energy savings compared to the base case of existing conditions. However, for comparison, the current level of chilled water production was taken as the baseline. The energy savings potential of each option was estimated on the basis of the difference in electrical energy input required to achieve this baseline level of chilled water output.

To estimate the chiller related energy saving potential of alternatives 1, 2, or 4 (all of which would meet the cooling tower performance level), performance curves from *ASHRAE Handbook*[11] were used as a reference to reflect typical cooling tower performances under varying load and weather conditions. The curves approximate the performance of the new or modified tower systems.

To use approximate performance data to estimate potential chiller energy savings, it was necessary to identify the seasonal air-conditioning loads which are currently met by the plant. Since no metered data was available, these loads were estimated on the basis of log and weather data. Chiller loads were estimated as a function of ambient dry bulb temperature from May through September.

Based on manufacturers data for chiller C.O.P. improvement as a function of condenser EWT, the base case and alternate case chiller input energies were estimated to be as follows:

| | |
|---|---|
| Base Case: | 2,653,000 kWh |
| Alternative Case: | 2,510,000 kWh |
| Savings: | 143,000 kWh |

Additional savings based on the assumption that the new towers could function with 3 gpm/ton (instead of the existing 3.65 gpm/ton) were also estimated. Assuming a pump efficiency of 68 percent, a motor efficiency of 80 percent, and a seasonal operating period of 3,720 hr, it was estimated that reducing the tower water flow from 6,200 to 5,700 gpm would save about 92,000 kWh of energy each year.

Finally, by comparing the specified fan horsepower for four new efficient 425-ton cooling tower cells to the total existing fan horsepower, it was found that a reduction of about 10 hp could be achieved. Assuming a 50 percent duty cycle and a 3,720 hr total operating period, potential energy savings of about 14,000 kWh were calculated. Therefore, the annual potential energy savings possible from alternatives 1, 2, or 4 are:

| | |
|---|---|
| Chiller Energy Savings: | 143,000 kWh |
| Pumping Energy Savings: | 92,000 kWh |
| Fan Energy Savings: | 14,000 kWh |
| Total Energy Savings | 249,000 kWh |

The energy savings is the same for all three alternatives; only the technical solutions differ.

Although the performance of the equipment configuration in alternative 3 is less adequately defined, it is estimated that the peak condition condenser EWT would

[11] *ASHRAE Handbook, Equipment*, pp 21.11-21.13.

probably be around 89 °F, or roughly half the improvement offered by the other alternatives. Therefore, the projected chiller energy savings were halved for this alternative. Also, because the new configuration would circulate half the flow through existing equipment and half through new equipment, pumping and fan energy savings are also halved. As a result, it is estimated that alternative 3 would enable annual energy savings of about 124,500 kWh.

*Cost Economics*

Of the four alternatives, replacing the existing towers would be most beneficial. Estimates of the costs associated with each alternative are presented in Table 10. Because boiler plant No. 4 was shut down during the summer, the decision was made not to modify the cooling towers.

## Hospital Plant Investigation

The hospital at Fort McClellan is currently supplied with chilled water from a mechanical plant. The plant uses two 200-ton centrifugal chillers with a 2-cell cooling tower as the primary equipment. The plant also has three 50-ton reciprocating chillers with associated air-cooled condensers for backup. Facility personnel indicated that on very hot days, the chiller head pressure had increased to a point near the high pressure cutout (125 psig). On such occasions, suspecting cooling tower insufficiency, the tower was flooded with city water.

Unfortunately, logs are not maintained for this plant because it is not continually manned. Facility records that could help identify or quantify the specifics of the problem involved do not exist. In spite of the lack of documentation, researchers decided that that a simple check of the tower water flows might illuminate the problem. This check was accomplished during a field survey of the plant in January 1984.

*Analysis*

The flow measurements in the hospital plant indicate a tower water flow rate of about 750 gpm per pump. This represents about 125 percent of the design tower water flow rate of 602 gpm per cell. Although overpumping can cause some rise in the tower leaving water temperature, the effect is minimal.[12] The 25 percent excess in flow through the tower would increase the tower leaving water temperature less than one degree.

**Table 10**

**Cost Data for Cooling Tower Alternatives**

| Alternative | Cost ($ FY84) | Annual Savings ($) | Simple Payback (yr) |
|---|---|---|---|
| 1 | 124,800 | 12,450 | 10.0 years |
| 2 | 124,900 | 12,450 | 10.0 years |
| 3 | 103,500 | 6,225 | 16.6 years |
| 4 | 134,300 | 12,450 | 10.8 years |

[12] *ASHRAE Handbook, Equipment.*

During the survey, a 2-in. bypass line in the tower (which was operating) was bypassing water around the distribution basin and directly into the collection basin. At the same time, one of the two float control valves was set up to keep makeup water flowing while the basin was overflowing to drain.

These two control conditions have opposite effects on tower heat rejection. The bypass reduces heat transfer to the air, while the makeup water, being cooler than the drained water it replaces, tends to lower the tower leaving water temperature. If the bypass valve is always left open, it could diminish tower capacity during summer operation as well.

However, the small quantity of water handled by the 2-in. line would have only a minimal effect on overall tower performance. With both pumps running, an estimated 90 gpm are being bypassed. This is less than 8 percent of the total tower flow, and would explain less than a one degree increase in tower leaving water temperature.

Some water was overflowing the distribution basin and running down the outside of the tower. Although this proves that there was some overpumping, it does not explain the reduction in tower capacity, since makeup water to replace the spilled water would be cooler than the spilled water.

Relocating the tower basin drain to a higher level than the inital design could also contribute to a tower capacity reduction. Because the resulting basin water level would be higher, water would occupy some of the space originally intended for air flow. Manufacturers representatives indicate that this would have a small deleterious effect on tower capacity.

*Conclusions*

The problem with the hospital cooling plant is not one of tower water flow inadequacy. Because of this, and the lack of data reflecting tower performance during trouble periods, it was considered premature to replace the existing tower.

**Central Plant No. 4 HTHW Reset**

The objective of this project is to reduce the temperature of HTHW leaving the plant as much as possible while still satisfying all loads on the system. Service is currently provided to nine buildings (including the plant itself). These buildings demand HTHW for space heating, domestic hot water heating, and kitchen steam.

*Alternative Approaches*

A number of parameters could be monitored to establish the appropriate level of HTHW setback including:

- HTHW return temperature
- HTHW system pressure
- Time of day
- Outside air temperature.

Although the HTHW return temperature is indicative of system load, it could provide a misleading reset signal to the boilers if loads are low because a block of system customers are off-line. In such a case, lowering the HTHW temperature would diminish the heating capacity of the on-line buildings even though their loads might be high. Therefore, return water temperature was rejected as a primary control input.

The performance curves for the system distribution pumps indicate that significant flow variations correspond to quite small pressure variations. Furthermore, plant logs show that the system HTHW temperature drop historically diminishes with decreasing loads, thus diminishing the impact of load variations on system flow rate. These two system characteristics indicate that the pump discharge pressure (system pressure) would be a poor indicator of system load. It, therefore, was also rejected as a reset control input.

The time of day has a significant impact on kitchen steam use and domestic hot water requirements. Based on metered data and conversations with facility personnel, these loads occur primarily between 5:00 a.m. and 8:00 p.m. daily. During this time, an estimated 300 °F HTHW supply must be provided to the steam generators to ensure steam production for the kitchen. Even at this temperature, however, the capacity of the generators is reduced. This reduced capacity could be a problem at times of peak kitchen loads. Therefore, time of day must be taken into account in establishing a reset shedule.

The fourth candidate parameter, outside air temperature, was also considered a reasonable indicator of space heating thermal loads. As shown by plant log data, the boiler output does vary with outside air temperature in a fairly predictable manner. Based on evaluation of the parameters, the reset program should be based on outside air temperature, but should contain an override to lock out the reset signal during hours when the kitchen steam loads dictate HTHW temperature requirements.

The time of day limitation imposed by the kitchen steam requirements could be eliminated by installing booster heaters in the HTHW lines supplying the steam generators. Booster heaters would permit an outside air temperature reset program to be used. Using the satellite boilers (currently being installed for summertime operation) to serve the kitchen and domestic hot water loads throughout the winter was also considered. This option was rejected primarily because of the expenses associated with operating and maintaining both the central plant and the satellite plants simultaneously.

Another alternative was to have the reset continually engaged but subjected to an adjustable limit during daytime hours. In this manner, some savings could possibly be achieved during periods when the kitchen equipment is in use. The limiting reset magnitude would be determined and adjusted based on operating experience. This alternative offers the best combination of payback and energy savings, and the cost allowed it to be implemented by installation personnel.

*Potential Fuel Cost Savings*

Plant logs were reviewed to determine the relationship between outside air temperature and plant load. Based on this relationship and the historical effectiveness of system heat exchangers in extracting heat from the HTHW circuit, a reset schedule was determined and corresponding potential energy savings were estimated. The results for the basic reset schedule and the various alternatives are given in Table 11. Only winter operation was considered since the central HTHW system is shut down during the summer.

Table 11

**Cost Data for the Reset at Central Plant No. 4**

| Reset Alternative | Annual Fuel Energy Savings (MBtu) | Annual Fuel Cost Savings ($) |
|---|---|---|
| Night only (9 hr/day) | 603 | 6,548 |
| Continuous Reset (24 hr/day) | 1,436 | 15,595 |
| Complete Reset at Night, limited reset during day ($\geq$ 300 $^{o}$F) | 756 | 8,210 |

*Summary Economics*

The three reset programs (identified in Table 11) require different investments for implementation. The greatest cost differential comes from including new booster heaters in the second alternative. The heaters are designed to remove significant time-of-day constraints from the reset schedule. The investments for the first and third alternatives basically cover new boiler combustion controls capable of accepting a reset signal from a new outside air temperature sensor and transmitter. (The existing controllers do not have this capability.)

The costs, savings, and payback periods for each alternative are summarized in Table 12. Based on data from the table, the third alternative offers the best combination of payback and energy savings.

Table 12

**HTHW Reset Summary**

| Reset Alternative | Implementation Cost ($) | Annual Fuel Cost Savings ($) | Simple Payback (yr) |
|---|---|---|---|
| Night only (9 hr/day) | 16,500 | 6,548 | 2.5 |
| Continuous Reset (24 hr/day) | 92,000 | 15,595 | 5.9 |
| Complete Reset at Night, limited reset during day ($\geq$ 300 $^{o}$F) | 18,000 | 8,210 | 2.2 |

*Conclusions and Recommendations*

The savings potential does not reflect plans to shut down central plant No. 4 during the summer (see project titled Boiler Plant No. 4 Summer Shutdown). Also, variable speed regulators have been installed on the system circulation pump motors. This modification is intended to vary the pump speed in response to changes in system temperature drop (i.e., the difference in temperature between HTHW leaving the plant and that returning). The energy savings the speed controller achieves depends on minimizing the flow for a given load. But lowering the HTHW supply temperature would require increasing the flow above that minimum for a given load. Therefore, temperature reset savings occur at the expense of pump energy savings and pump energy savings occur at the expense of temperature reset settings. Assuming operation during winter months only, the variable speed pump drives could save approximately $3,000 per year in electricity costs. Since this is comparable to the potential reset savings estimated earlier, but cannot be achieved with the temperature reset program in place, it was concluded that the reset project should be not be implemented.

**Central Plant No. 2 Pressure Reset**

The objective of this project is to maintain a steam pressure at the plant no greater than that necessary to satisfy all loads. Satisfaction of all loads can generally be ensured by maintaining a steam pressure of approximately 50 psig at the hospital. Sterilization units at the hospital have a minimum requirement of 50 psig. Therefore, the proposed project entails providing a means by which the steam pressure at the hospital can be monitored and used to control the boiler firing at plant No. 2 so the pressure remains at 50 psig at the hospital.

*Alternative Approaches*

1. Radio transmission of hospital steam pressure to plant control device

2. Hard-wire connection of a hospital steam pressure sensor to the plant controls

3. Periodic manual reset of plant pressure controls based on observed hospital pressure conditions and prevailing weather conditions

4. Use of outside air temperature to reset plant pressure setpoint.

The first approach, radio telemetry, could be implemented in a variety of ways, depending on the equipment type desired. The options include:

- Commercially available FM transmitter/receivers sending digital signals
- Built-up electronic package to transmit/receive analog signal over FM radio
- Built-up electronic package to transmit/receive analog signal using microwaves
- System of commercially available components to transmit/receive an analog signal using pulse width modulation with VHF frequencies.

Based on cost, availability, and reliability, the last of these possibilities was considered the optimum choice for the radio alternative.

The hard-wire connection could be completed during the programmed replacement of the steam distribution line from the plant to the hospital. Armored cable could be laid in the pipe trench during this work and save on trenching costs.

With no additional capital investment, a significant savings could be achieved by manually resetting the plant pressure control for summer and winter conditions. A review of plant logs indicates a fairly predictable relationship between outside air temperature and steam production by the plant. Because of this, outside air temperature can also be used as an indicator of the magnitude of distribution system pressure losses. Therefore, it was considered as a candidate control input to establish a pressure reset schedule.

*Potential Fuel Cost Savings*

In a manner similar to the analysis of central plant No. 4 reset savings potential, estimates were developed for the savings potential of this measure. The estimates are listed in Table 13.

*Conclusions*

Implementation costs were estimated for each of the alternatives being considered. They are summarized in Table 13 with simple paybacks based on the above energy cost savings estimates. Each alternative exhibits a quick payback. The project titled Steam Pressure Reduction in the District Heating System incorporates the seasonal manual reset.

**Central Plant No. 1 Automatic Blowdown**

The objective of this project is to provide a means to automatically ensure that adequate, but not excessive quantities of boiler water are blown down to keep the boiler water quality within recommended levels.

**Table 13**

**Central Plant No. 2 Reset Summary**

| Alternative | Implementation Cost ($) | Annual Fuel Energy Savings (MBtu) | Annual Fuel Cost Savings ($) | SPB |
|---|---|---|---|---|
| Radio Telemetry | 14,300 | 1,662 | 18,050 | 0.79 |
| Hard-wire Telemetry | 12,000 | 1,662 | 18,050 | 0.66 |
| Seasonal Manual Reset | 0 | 1,289 | 13,998 | 0 |
| OSA Temperature Based Pressure Reset | 7,100 | 1,473 | 15,997 | 0.44 |

*Alternative Approaches*

Three approaches to automating the blowdown procedures at plant No. 1 were considered:

1. Continuous control based on makeup water flow rate

2. Continuous control based on sensed boiler water conductivity

3. On/off (shot type) control based on sensed boiler water conductivity.

The first option uses the cooling effect of makeup water on a heat sensitive valving arrangement to admit blowdown when makeup is flowing, and to terminate blowdown when makeup stops. One shortcoming of this approach is its dependence on a constant solids concentration in the makeup water to ensure proper blowdown quantities. The heat sensitive valving arrangement of this system is an integral part of a heat recovery package built into the equipment. It is therefore impossible to install this system without paying for new heat recovery capability as well.

The second alternative makes use of a sensing probe installed in the boiler near the water line to monitor conductivity. The output from this sensor modulates a control valve to regulate blowdown.

The third alternative is similar to the second, except it operates a valve in an on-off, rather than modulating fashion. This type of system has the advantage of a flushing effect when the valve opens, which helps minimize blockage problems. It was therefore selected as the preferred equipment for this project.

*Potential Fuel Cost Savings*

Fuel cost savings from this project result to the extent that the automatic controls eliminate excessive blowdown which may currently be occurring. It is difficult to estimate this quantity accurately since blowdown quantities are not recorded on plant logs. The cost of implementing the project was estimated at $14,200 with an annual savings of $1,000. Based on a simple payback period of 14.2 years, the project was not recommended for implementation.

# 8 CONCLUSIONS AND RECOMMENDATIONS

The installation-wide energy conservation demonstration at Fort McClellan contributed to the 14.4 percent reduction in baseline (weather independent) energy consumption since 1984. Throughout this demonstration, a consistent policy regarding metering and feasibility analysis was employed before implementing a project. Hence, a number of proposed projects with a modest return on investment were rejected. Generally the implemented projects have wide applicability to other Army installations.

## Energy Metering

The Army does not have a policy that mandates metering of energy use in buildings served by central plant heating systems. However, due to the number of such systems in existence throughout the Army, and the amount of information which can be gained, it is recommended that installation of metering equipment in future construction as well as in existing buildings be considered.

## Steam Pressure Reduction in the District Heating System

Predicted values for building steam demand and information gathered through condensate metering enabled USA-CERL researchers to develop an accurate means of computing energy savings. The efficiency of the district heating system is approximately 53 percent. In 1982, the fuel use in boiler plants 1 through 3 was approximately 180,000 MBtu. Assuming a boiler efficiency of 75 percent, 135,000 MBtu enter the steam distribution system in the form of steam. Of this, 47 percent (63,450 MBtu) is lost to conduction between the pipes and the ground and as live steam from leaks and faulty equipment. Natural gas has a unit cost of $10.86 per MBtu. This results in losses of $689,067 annually.

The savings were calculated by attributing 75 percent of the losses to conduction and the remainder to leaks and faulty valves. The project cost was $100,000; hence, a simple payback of 1.5 years was achieved. It is recommended that other agencies with district heating systems operating at much higher than end-use pressures consider this type of retrofit. In any case, a detailed engineering analysis must be performed before implementation.

## Steam Vault Drainage

Random inspection of steam vaults revealed that they were often filled with water and that the piping was in contact with the water. In some cases, the vaults were not equipped with pumps. In others, the pumps (steam powered) were inoperative. Draining this water is saving approximately $4,600 per year. The estimated heat loss from a bare, 6-in. steam pipe carrying 338 °F steam and covered by 150 °F water is 50,000 Btu/hr-ft. A general recommendation to keep the vaults dry applies to all Army installations.

## Oxygen Trim Systems for Central Plant Boilers

Oxygen trim systems require a large investment of time and effort. The correct setup procedure includes installing the appropriate linkages, ensuring proper interface with existing controls, accurate installation of the controller, and periodic calibration of the oxygen sensor and the controller. Verification is necessary to ensure the success of the investment. If all equipment is functioning correctly, however, an oxygen trim system can improve combustion control. For general application, it is recommended that installation personnel become thoroughly familiar with the system before implementation.

## Boiler Plant No. 4 Summer Shutdown

This investigation identified no significant technical constraint which would limit the ability to shut down the HTHW system at Fort McClellan each summer. Other HTHW systems are shut down on a regular basis with no net damaging effects. Typically, district heating systems operate with an efficiency between 50 and 75 percent. This energy represents the savings associated with use of the on-site boilers. For the months of interest, approximately 37,500 MBtu enter the system and, assuming a district efficiency of 75 percent, the result is a loss of 9,375 MBtu annually. A cost of $10.86 per MBtu results in an annual savings of $101,812.50.

In addition to significant savings in energy and operating costs, a regular system shutdown allows time for preventive and corrective maintenance. However, the number of system leaks will increase as a result of the thermal cycles. The leaks are expected to be relatively minor and the additional maintenance required for repair will be more than offset by the savings realized during the extended shutdown period. Finally, the heat transfer from the HTHW piping to the chilled water piping will be alleviated.

## Reduction in Outdoor Air Ventilation

Outdoor air reduction to conserve energy at this installation was first suggested in 1979. Based on estimates of outdoor air flow rates, and adjustment in accordance with ASHRAE Standard 62-1973 (a predecessor of the standard used in this study), a simple payback period of about 1 month was projected. Since the current standard allows less outdoor air, thereby saving additional energy at virtually the same cost, an even shorter payback period would have been expected using the more recent standards. While the actual 2-year payback is acceptable, it does point out the value of using accurate field measurements rather than blanket estimates.

## Replacement of Inefficient Electric Motors

Kilowatt measurements indicated that 104 of the 201 motors evaluated were underloaded enough to yield a payback of 5 years or less. The rpm measurements confirmed that 78 of those 104 warranted replacement. Ten motors were not replaced because of restrictions on size, cost, enclosure type, or plant design. Replacement of 68 motors resulted in a 17 percent decrease in electrical power demand. The results are conclusive: oversized motor replacement can result in not only a significant decrease in electrical power demand but also an increase in power factor.

### Dispensary Chiller Installation

Although the peak load of the dispensary was calculated to be 33 tons, the installed capacity of the existing AHUs is only about 25 tons. This capacity has been shown to be inadequate in service. Therefore, a 30-ton chiller unit was installed. This unit will allow peak load requirements to be met should the AHUs be upgraded to appropriate capacities in the future. The shortage of 3 tons, as compared with the calculated peak of 33 tons, could be tolerated more easily during the swing seasons than the summer because of the relative infrequency of peak loads. The chiller replacement was $37,500 with a simple payback of 4.5 years. It is recommended that other facilities experiencing a large discrepancy between chiller sizing and demand consider this type of retrofit.

### Combustion Optimization of Gas-fired Heating Equipment

To demonstrate the effectiveness of combustion adjustment, 204 furnaces and hot water heaters at Fort McClellan were evaluated. These units had a combined heating capacity of 88,952,000 Btu/hr. The combustion efficiency of more than half the units was improved, with efficiency increases as high as 25 percent in some cases. The average improvement for all the units was 2.5 percent. Based on the average heating season and fuel costs for Fort McClellan, this project is estimated to have saved $16,000 with an investment of $17,000 in labor and equipment costs; hence, a simple payback of 1.1 years was achieved.

### Infiltration Study of Family Housing

The results of the infiltration study show that the average reduction in leakage area was 37 percent for the four units retrofitted--mostly the result of sealing leaks in the building envelope. By investing 3 hours of labor and $60 in materials per unit, an average of $38 annual savings per unit can be recognized. This represents an approximate simple payback period of 3 years. Because this did not meet the requirement of rapid payback, an installation-wide infiltration reduction program was not implemented as part of the demonstration. However, such a program may apply to installations with larger heating requirements.

### Computer Decision Support for Energy Management

The management system provides immediate feedback on how energy is distributed to and used by the various functions at an Army installation. Target areas for energy reduction can be readily identified and the extent of the potential application of an energy conservation opportunity and associated savings can be computed.

By using this system, energy managers can perform consistent analyses on energy consumption and make correct decisions on the course to take in energy conservation.

### Radio Control of Exterior Lighting

It is recommended that Facilities Engineers investigate any radio control system already in use for possible expansion and application to lighting control. Most radio control system manufacturers sell a remote switch which is comparable in operation and cost to the digital control switch described in this report.

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