CALIFORNIA ENERGY COMMISSION

Design and Optimization of Solar Absorption Chillers

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Legal Notice

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Preface

The Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The PIER Program, managed by the California Energy Commission (Commission), annually awards up to $62 million to conduct the most promising public interest energy research by partnering with Research, Development, and Demonstration (RD&D) organizations, including individuals, businesses, utilities, and public or private research institutions.

PIER funding efforts are focused on the following six RD&D program areas:

- Buildings End-Use Energy Efficiency
- Industrial/Agricultural/Water End-Use Energy Efficiency
- Renewable Energy
- Environmentally-Preferred Advanced Generation
- Energy-Related Environmental Research
- Strategic Energy Research.

What follows is the final report for the "Design and Optimization of a Solar Absorption Chiller" project, Contract Number: 500-02-035, conducted by Bergquam Energy Systems, 8611 Folsom Blvd., Sacramento, CA 95826. The report is entitled "Design and Optimization of Solar Absorption Chillers." This project contributes to the Renewable Energy program.

For more information on the PIER Program, please visit the Commission's Web site at: http://www.energy.ca.gov/research/index.html or contact the Commission's Publications Unit at 916-654-5200.

Executive Summary

This project is concerned with the design and optimization of solar-fired, double effect absorption chillers. As part of a previous California Energy Commission project, Bergquam Energy Systems, and other project participants, demonstrated the use of a solar-fired, double effect absorption chiller as part of a complete HVAC system on an 8,000 ft^2 commercial building in Sacramento, California. The chiller purchased for that project was manufactured and marketed by McQuay-Sanyo, had a nominal capacity of 20 tons and was driven by natural gas. The chiller was converted to solar fired by removing the natural gas-fired high temperature generator and all of the associated combustion equipment.

A new hot water-fired, high temperature generator was designed and built and installed in the McQuay chiller. The new generator is a shell and tube type heat exchanger with 61 stainless steel tubes and a stainless steel shell. The tubes are 0.5 inches diameter and 60 inches long. The shell is 6 inches in diameter and 56 inches long. A lithium bromide / water solution was heated in the tubes by hot water that is pumped through the shell. The energy to heat the water is from an array of Integrated Compound Parabolic Concentrator (ICPC) solar collectors. The cost of the original Mc Quay-Sanyo chiller was about $1,700 per ton. It is estimated that the new high temperature generator could reduce the cost of the chiller to approximately $1,400 per ton.

The nominal operating temperature of the LiBr / H_2O solution in the high temperature generator was about 260°F. The average log mean temperature difference between the fluids was 40-50°F. As a result, the collector array and the solar storage tank had to operate at a temperature in excess of 300°F and a pressure of 100 psi. The initial costs and the maintenance expenses are both very high for equipment operating under these conditions.

Objectives

This project seeks to improve the performance and reduce the cost of solar driven space cooling technology for small to medium-sized commercial buildings. The project addresses the PIER program objective of improving electricity system reliability by reducing electrical consumption during peak demand created by cooling commercial buildings. Specifically, this technology seeks to replace packaged, compression air conditioning systems, which are typically used for cooling small to medium-sized buildings (20 to 100 tons), with a cost effective solar absorption HVAC system.

Technical Objectives

The overall technical objective of the project is to optimize the design of the high temperature generator in a solar fired, double effect absorption chiller. The performance objectives for the chiller are:

- COP in the range of 1.2 to 1.4 when the solar system is operated at high temperature (approximately 300°F)
- COP in the range of 1.1 to 1.2 when the solar system is operated at low temperature (approximately 250°F)
- A cooling capacity of 16 to 18 tons at both high and low operating temperatures

Economic Objectives

The overall economic objective of this project is to lower the cost of the solar fired absorption chiller, and the complete solar HVAC system, by lowering the operating temperature of the equipment to approximately 250°F. The present objectives are:

- Solar fired chiller cost of $1,100 / ton
- Complete solar HVAC system cost of $4,500 / ton
- Simple payback for the complete solar HVAC system of four years or less.

Computer models have been developed to perform thermodynamic and heat transfer calculations for a double effect, absorption chiller. The models analyze the performance of the individual components of the chiller and of the complete absorption cycle. The most important component is the high temperature, or first stage, generator.

The results from a literature search and from the computer modeling performed by the contractor indicate that twisted metal strips, inserted in the individual generator tubes, will have the biggest impact on improving the performance of the high temperature generator. The twisted inserts significantly augment the heat transfer with a very small pressure drop, Ref. [1].

The work completed during the project includes 1. Constructing a generator tube test device and performing laboratory bench tests to measure the improvement in performance by installing twisted stainless steel inserts in the individual tubes in the generator, 2. Fabricating and installing inserts in all 61 of the tubes in the high temperature generator of a 20 ton, solar-fired absorption chiller and 3. Field testing the solar-fired HVAC system as it provided the air conditioning load for an 8,000-ft2 commercial building.

In order to confirm and quantify the effect of twisted inserts in solar HVAC applications, a generator tube test device was built and bench tests were conducted on full-scale generator tubes. The device made it possible to test three individual generator tubes operating simultaneously and in parallel. Experimental data were obtained with both water and a lithium bromide / water solution in the generator tubes. In both cases the heat was supplied by hot water in the shell.

Experimental results from the generator tube test device showed that:

- With water in the tubes, the twisted inserts increase the inside heat transfer coefficient by an average of 175 percent.
- With the LiBr / H_2O solution as the working fluid, the inserts increase the inside heat transfer coefficient by an average of 47 percent.
- The presence of the inserts caused the boiling process to be much more stable.

In an effort to improve and optimize the performance of the solar fired, 2E absorption chiller, twisted stainless steel inserts were fabricated and installed in all 61 of the tubes in the water-fired, high temperature generator. Throughout the summer of 2001, the solar-fired HVAC system, with the optimized generator, was field tested as it provided the air conditioning load for our 8,000 ft^2 commercial building. The important conclusions are to compare the heat transfer coefficients, the log mean temperature differences, the COPs and the cooling capacities of the chiller with the original and the modified generators. It is also important to compare the operating characteristics and the costs of the equipment and of the HVAC systems.

Outcomes

The performance of the optimized chiller was significantly improved over that of the original solar-fired chiller. Essentially all of the technical objectives of the project have been achieved. As improvements are made in the commercially available equipment required for solar HVAC systems, it will be possible to achieve the economic objectives of this project.

Technical Outcomes

- With the original chiller, the collector array and storage tank had to operate at a minimum of 310°F in order to achieve an average generator temperature of 260°F. This is a temperature difference of 50°F. The chiller achieved a COP of 1.2 and provided up to 17 tons of cooling.
- With the optimized generator, we are able to operate the collector array and storage tank at 278°F and still achieve a generator temperature of 260°F. This reduces the temperature difference to 18°F.
- With the optimized generator, the 20-ton chiller achieved an actual measured COP of 1.05 and a full-load COP of 1.2.
- The improved chiller is able to provide the entire cooling load for an 8,000-ft2 building. The load was approximately 12 tons for most of the field-testing.

The average value of the overall heat transfer coefficient with the new generator was 212.6 Btu / hrft^{2}F. With the original generator, the range was 140 to 175 Btu / hrft^{2}F. This is as much as a 50 percent increase in the overall heat transfer coefficient.

The average value of the log mean temperature difference for the new generator was about 18°F. With the original generator the LMTD was between 40°F and 50°F. This means that the difference between the average generator temperature and the average firing water temperature was reduced by 20 to 30°F.

With the new generator, the highest COP averaged over a 10-minute time interval was 1.45. Under steady state conditions, the average COP was approximately 1.0. Similar results were achieved with the original generator.

With both generators, the absorption chiller provided the entire cooling load of the building, which was usually about 12 tons. In field testing, the load is not controlled. With the original generator, there were some periods when the chiller provided in excess of 16 tons. However, on average, the cooling capacity was about the same.

Both HVAC systems are designed with high temperature storage. A 1,000 gallon storage tank is used so the systems will operate during cloudy periods of up to one hour. The cost of an insulated high temperature storage tank at 300°F is about $10,000. At 250°F the cost is approximately $6,000.

The total installed cost of a solar HVAC system with the collector array and storage tank operating at temperatures above 300°F is about $7,500 per ton. The goal with the optimized generator is to lower the installed cost to about $5,000 per ton.

Economic Outcomes

During the course of the project we met with several manufactures of absorption chillers and related equipment. We have received a draft Memorandum of Understanding for a Technical Collaboration Agreement from one of the manufactures. The cost of the original double effect, absorption chiller was $1,700 per ton. It is estimated that the new high temperature generator could reduce the cost of the chiller to under $1,400 per ton. As improvements are made in the commercially available equipment required for solar HVAC systems, the economic objectives of the project should be met.'

Conclusions

- The performance of the optimized chiller was significantly improved over that of the original solar-fired chiller.
- The cost of the original double effect absorption chiller purchased by the contractor was about $1,700 per ton. It is estimated that the new high temperature generator could reduce the cost of the chiller to under $1,400 per ton. This is based on the following facts.
- The water fired chiller uses about half of the lithium bromide / water solution of the original chiller
- The design of the new generator is simpler and does not require any combustion equipment.
- The net reduction in demand resulting from the use of solar HVAC systems is about 1.3kW per ton or 26kW for a 20-ton system. The installation of just 50 solar HVAC system per year, with and average capacity of 20 tons, would result in an annual peak demand reduction of 1.3 MW.
- As improvements are made in the commercially available equipment required for solar HVAC systems, it will be possible to achieve the economic objectives of this project.
- A recently completed market study by Sun Utility Network identified two target markets for solar HVAC. The study concluded that solar HVAC must be part of a comprehensive energy strategy and that a large number of systems could be installed by 2010.

Benefits to California

The main advantage of solar HVAC systems is that they displace electrically driven compression air conditioners. These are the cause of the summertime peak demand problem experienced by many electric utilities in California. The potential benefits to California from the widespread implementation and use of solar chiller technology are significant.

Abstract

This report presents the results of a PIER project that involved the design and optimization of a solar-fired, double effect absorption chiller.

Solar powered absorption chillers use water heated in an array of solar collectors to boil a solution of lithium bromide and water. The energy transfer process between the heating water from the collectors and the LiBr / H_2O solution was the focus of this project. The work completed involved optimizing the design of the high temperature generator. A method of augmenting the heat transfer process in the generator was developed, bench tested and implemented in an operating 20 ton solar HVAC system. The optimized design involved installing twisted stainless steel inserts in the tubes where the LiBr / H_2O solution boils and the refrigerant vapor is generated. The inserts augment the overall heat transfer coefficient between the heat medium in the shell side of the generator and the working fluid in the tubes.

A solar-fired, double effect absorption chiller requires the collector array and storage tank to operate at temperatures in excess of 300°F. At these temperatures, the heating water must be at a pressure of 100 psi to prevent it from boiling. This combination of high temperature and high pressure requires that the collectors, storage tanks, pumps, valves and piping be designed according to pressure vessel codes. This increases the initial cost of the system and also requires significant maintenance.

The main objective of this project is to develop a method of lowering the requirement of 300°F heating water. The ultimate goal is to operate at about 250°F while maintaining the Coefficient of Performance (COP) and the cooling capacity of the absorption chiller.

The results presented in this report show that the generator with twisted inserts can operate with an average temperature difference of 18°F. The average COP is about 1.0 and the chiller provided all of the cooling required by an 8,000 ft^2 building. Without the inserts, the generator operated with a temperature difference of 40 to 50^{o}F. The average COP was also approximately 1.0.

The main advantage of solar HVAC systems is that they displace electrically driven compression air conditioners, which are the cause of the summertime peak demand problem experienced by many electric utilities in California. The potential benefits to California resulting from the widespread implementation and use of solar chiller technology are enormous. Compression air conditioners require about 1.5 kW per ton.

The pumps and fans used to operate a solar HVAC system require about 0.2 kW per ton. As a result, the net demand reduction is 1.3 kW per ton or 26 kW for a 20 ton system. The installation of even 50 solar HVAC systems per year, with an average capacity of 20 tons, would result in an annual peak demand reduction of 1,300 kW or 1.3 MW. There are very few renewable energy technologies that have the potential to reduce peak electrical demand by this amount.

1.0 Introduction

For the past 17 years, Bergquam Energy Systems (BES) has been involved in the design, installation, operation, and maintenance of solar absorption chillers and solar HVAC systems. BES has also been actively involved in research and development projects that are designed to improve the performance, lower the cost and simplify the maintenance of both single effect and double effect solar absorption systems. (Single effect chillers have one generator and double effect chillers have two generators)

The overall goal of this project was to improve the performance and reduce the cost of solar driven space cooling technology for small to medium-sized commercial buildings. The project addresses the PIER program objective of improving electricity system reliability by reducing electrical consumption during peak demand created by cooling commercial buildings. Specifically, this technology seeks to replace packaged, compression air conditioning systems, which are typically used for cooling small to medium-sized buildings (20 to 100 tons), with a cost effective solar absorption HVAC system.

1.1. Background and Overview

Successful projects that have been completed by Bergquam Energy Systems include the following:

- A 10 ton single effect system installed in 1985 on a 10,000 ft^2 commercial building in Sacramento, CA.
- A 4 ton single effect system installed in 1985 on a 3,000 ft^2 residence in El Dorado Hills, CA.
- A 10 ton single effect system installed in 1990 on an 8,000 ft^2 commercial building in Sacramento, CA.
- A 20 ton double effect system installed in 1998 on the 8,000 ft^2 commercial building.

All of these systems have been in continuous operation since they were installed and BES has provided all of the required maintenance. BES has also been the project director for research and development work funded by the California Energy Commission, the National Renewable Laboratory, Sandia National Laboratories, the Sacramento Municipal Utility District, and the South Coast Air Quality Management District.

These projects have demonstrated the technical feasibility of solar HVAC systems. The main advantage of these systems is that they displace compression air conditioners, which require approximately 1.5 kW per ton of electric utility provided power. Equally important is the fact that the electricity is displaced during the cooling season when electric utilities experience their peak demand for power. Solar HVAC systems provide significant reductions in the charges for both electric energy and peak demand. While there is widespread interest in the technology, there are still very few operating systems in California. This is primarily because of the high initial cost of the equipment for a solar HVAC system.

1.2. Project Objectives

This project is concerned with the design and optimization of a solar fired, double effect (2E) absorption chiller, which uses a Lithium Bromide / Water solution as the working fluid. The work involved improving the performance of the chiller by optimizing the design of the high temperature generator. A method of augmenting the energy transfer within the high temperature generator was developed, bench tested and implemented in the 20 ton double effect system on the 8,000 ft^2 building in Sacramento. The optimized design involved installing twisted stainless steel inserts in the tubes where the LiBr / H_2O solution boils and the refrigerant vapor is generated. The inserts augment the overall heat transfer coefficient between the solar heated water in the shell side of the generator and the working fluid in the tubes.

A solar fired, double effect absorption chiller requires the collector array and the storage tank to operate at temperatures in excess of 300°F. At these temperatures, the heating water must be at a pressure of 100 psi to prevent it from boiling. The combination of high temperature and high pressure requires that the collectors, the storage tank and the pumps, valves and piping be designed according to pressure vessel codes. This increases both the initial cost and the maintenance expenses associated with the system.

Therefore, the aim of this project is to develop a method of lowering the required 300°F temperature of the collector array and storage tank. The ultimate goal is to operate the high temperature generator at about 250°F while maintaining the Coefficient of Performance (COP) and the cooling capacity of the absorption chiller.

This project seeks to improve the performance and reduce the cost of solar driven space cooling technology for small to medium-sized commercial buildings. The project addresses the PIER program objective of improving electricity system reliability by reducing electrical consumption during peak demand created by cooling commercial buildings. Specifically, this technology seeks to replace packaged, compression air conditioning systems, which are typically used for cooling small to medium-sized buildings (20 to 100 tons), with a cost effective solar absorption HVAC system.

1.2.1. Technical Objectives

The overall technical objective of the project is to optimize the design of the high temperature generator in a solar fired, double effect absorption chiller. The performance objectives for the chiller are:

- A COP in the range of 1.2 to 1.4 when the solar system is operated at high temperature (approximately 300°F)
- A COP in the range of 1.1 to 1.2 when the solar system is operated at low temperature (approximately 250°F)
- A cooling capacity of 16 to 18 tons at both high and low operating temperatures

1.2.2. Economic Objectives

The overall economic objective of this project is to lower the cost of the solar fired absorption chiller, and the complete solar HVAC system, by lowering the operating temperature of the equipment to approximately 250°F. The present objectives include:

- solar fired chiller cost of $1,100 / ton
- complete solar HVAC system cost of $4,500 / ton
- simple payback for the complete solar HVAC system of 4 years or less

1.3. Report Organization

Section 2.0 Project Approach and Outcomes describes the following:

- Design and testing of the high temperature generator, the generator tube test device that was built and the bench testing that was completed;
- Modification and testing of the double effect chiller; and
- Testing of the solar HVAC system using the 2E chiller with the optimized generator.

Section 3.0 Conclusions and Benefits to California discusses our conclusions and the benefits to California of this research.

2.0 Project Approach and Outcomes

The work on this project started, with the kick-off meeting, on October 1, 1999 and was completed, on schedule, on March 31, 2002. The budget for the project includes $150,000 in PIER expenditures and match fund expenditures from the Contractor, the Sacramento Municipal Utility District and the South Coast Air Quality Management District as follows:

BES		$122,500
SMUD		20,000
SCAQMD		7,500
	Total	$150,000

2.1. Design and Testing of High Temperature Generator

In the heat transfer analysis of the high temperature generator, three thermal resistances in the path of heat flow from the solar heated hot water to the LiBr / H_2O solution, are combined into an overall heat transfer coefficient, U, by a procedure described below. The generator is a shell and tube heat exchanger with hot water in the shell and LiBr / H_2O inside of the tubes. The total thermal resistance, R, to heat flow from the hot water, across the tubes to the LiBr / H_2O solution, is composed of three individual thermal resistances, Ref. [2]. These are:

- Thermal resistance of the outside flow
- Thermal resistance of the tube material
- Thermal resistance of the inside flow

In equation form, the terms are given by:

$$R = \frac{1}{A_o h_o} + \frac{t}{k A_m} + \frac{1}{A_i h_i}$$

Where

A_o, A_i are the outside and inside surface areas of the tubes

A_m is the logarithmic mean area

h_o, h_i are the inside and outside heat transfer coefficients

k is the conductivity of the tube material

t is the thickness of the tubes

R is the total thermal resistance from outside to inside

In this analysis, the thermal resistance is expressed as an overall heat transfer coefficient, U, based on the outside surface area of the tubes. The thickness of the tubes is small and the thermal conductivity is high enough so that the tube resistance is negligible. Using the fact that $A_i \approx A_o = A$, the result is:

$$U = \frac{1}{AR} = \frac{1}{\frac{1}{h_o} + \frac{1}{h_i}} \qquad (1)$$

The equations used to calculate the rate of heat transfer in a shell and tube heat exchanger are:

$$\dot{Q} = UA\Delta T_{ln} \qquad (2)$$

U is the average overall heat transfer coefficient and A is the total heat transfer area.

ΔT_{ln} is the logarithmic mean temperature difference (LMTD) between the hot and cold fluids over the entire length of the heat exchanger. The formula is:

$$\Delta T_{ln} = \frac{\left(T_{h,i} - T_{c,i}\right) - \left(T_{h,o} - T_{c,o}\right)}{\ln\left[\frac{T_{h,i} - T_{c,i}}{T_{h,o} - T_{c,o}}\right]} \qquad (3)$$

The subscript h denotes the hot water and c the colder fluid (LiBr / H_2O).

The total heat transfer rate between the fluids is also determined from

$$\dot{Q} = \left(\dot{m}c_p\right)_h \left(T_{h,i} - T_{h,o}\right) \qquad (4)$$

Note that a similar equation can not be used for the LiBr / H_2O because there is a phase- change process taking place as refrigerant vapor is boiled out of the solution.

$T_{h,i}$, $T_{h,o}$, $T_{c,i}$ and $T_{c,o}$ are the temperatures of the hot and cold fluids in and out of the heat exchanger.

2.1.1. Testing Procedure

The objective of the testing is to determine the effect of twisted inserts on the inside heat transfer coefficient. During each test the following steps were followed.

1. Measure the flowrates m_h and m_c and the temperatures

$T_{h,i}$, $T_{h,o}$, $T_{c,i}$ and $T_{c,o}$

2. Calculate Q using equation (4)

3. Use (3) to calculate ΔT_{ln}

4. Calculate the overall heat transfer coefficient using equation (2). This takes the form:

$$U = \frac{\dot{Q}}{A\Delta T_{ln}} \qquad (5)$$

5. Calculate the outside heat transfer coefficient. This involves calculating the Reynolds number for the flow:

$$Re = \frac{VD_H}{\nu}$$

V is the average velocity, D_H the hydraulic diameter and ν the kinematic viscosity at the average fluid temperature. In the test device, the hot water is flowing in the annular space between two concentric tubes. The hydraulic diameter is the difference between the outside diameter and the inside diameter of the annulus

$$D_H = D_o - D_i$$

After the Reynolds number has been determined, the next step is to calculate the Nusselt number (Nu) for the fluid flow. The equation used is:

$$Nu = 0.023 Re^{0.8} Pr^{0.3} \qquad (6)$$

where Pr is the Prandtl number at the average temperature of the fluid. With the Nusselt number for the flow, the outside heat transfer coefficient can be found. By definition

$$Nu = \frac{h_o D_H}{k}$$

where k is the thermal conductivity of the fluid. Re-arranging this equation to solve for h_o gives:

$$h_o = \frac{k}{D_H} Nu$$

with the calculated outside surface heat transfer coefficient and the measured overall heat transfer coefficient, from (5), the final step is to calculate the inside heat transfer coefficient using equation (1). Solving for h_i, (1) takes the form

$$h_i = \frac{1}{\frac{1}{U} - \frac{1}{h_o}}$$

For all of the test runs, this is the procedure that was used to determine h_i.

The main objective of this project is to lower the operating temperature of the collector array and the high temperature storage tank in a solar fired, double-effect absorption HVAC system. One method of lowering this temperature is to increase the inside heat transfer coefficient in the high temperature generator. Increasing the inside heat transfer coefficient decreases the thermal resistance between the solar heated hot water and the LiBr / H_2O solution on the inside of the tubes. This in turn lowers the log mean temperature differences (LMTD) between the hot water and the LiBr / H_2O solution and allows the generator to transfer the required amount of heat with a much smaller temperature difference.

2.1.2. Generator Bench Testing

This section presents the results of the bench testing work that was performed to determine the effectiveness of using twisted, stainless steel inserts in the inner tubes of water fired generators, Ref. [3]. These are the types of generators that are used in solar-fired absorption chillers.

The generators designed and built by BES are shell and tube type heat exchangers. Solar heated water is pumped through the shell with outside heat transfer coefficient h_o. A lithium bromide / water solution circulates through the tubes with inside heat transfer coefficient h_i. In this application h_o is much greater than h_i (the conductive resistance of the tube is negligible). In order to improve the performance of the generators, it is necessary to augment h_i. This will lower the required temperature of the heating water and improve the performance of the generator. The use of twisted stainless steel inserts was chosen as the most favorable method of increasing the turbulence of the solution as it is heated in the tube and water vapor is boiled out of the solution.

Twisted strip inserts, manufactured from stainless steel sheet metal, were used. The sheet metal is 22-gauge stainless steel, which is sheared into strips 13/32(0.406) inch wide and 60 inches long. The inserts are fabricated by a device known as - "The Twister". A strip is placed in the Twister and weights are added to put the strip in tension. The mechanical advantage of the device is 6.5 and as a result a weight of 15 pounds results in a tension force of 97.5 pounds.

The chiller modification and optimization work involved augmenting the heat transfer to the LiBr/H_2O solution on the inside of stainless steel tubes in the high temperature generator. The augmentation was accomplished by inserting twisted stainless steel inserts in all 61 of the tubes. As part of this project, a Generator Tube Test Device was built to perform bench tests on generator tubes with and without inserts. The test device simulates the actual conditions inside the high temperature generator.

Figure 1 shows a schematic of the Generator Tube Test Device, with instrumentation locations. Table 1 lists the instrumentation.

Table 1. Generator Testing Sensor Description

1	T_{Ref}	Refrigerant Temperature
2	T_{S1}	Strong Solution Temperature Tube 1
3	T_{S2}	Strong Solution Temperature Tube 2
4	T_{S3}	Strong Solution Temperature Tube 3
5	T_{w}	Weak Solution Temperature
6	$T_{w\ out}$	Weak Solution Temperature out of Mixer
7	T_{In}	Water Inlet Temperature
8	T_{out1}	Water Outlet Temperature Tube 1
9	T_{out2}	Water Outlet Temperature Tube 2
10	T_{out3}	Water Outlet Temperature Tube 3
11	FM_{s1}	Solution Flow Meter Tube 1
12	FM_{s2}	Solution Flow Meter Tube 2
13	FM_{s3}	Solution Flow Meter Tube 3
14	FM_{w1}	Water Flow Meter Tube 1
15	FM_{w2}	Water Flow Meter Tube 2
16	FM_{w3}	Water Flow Meter Tube 3

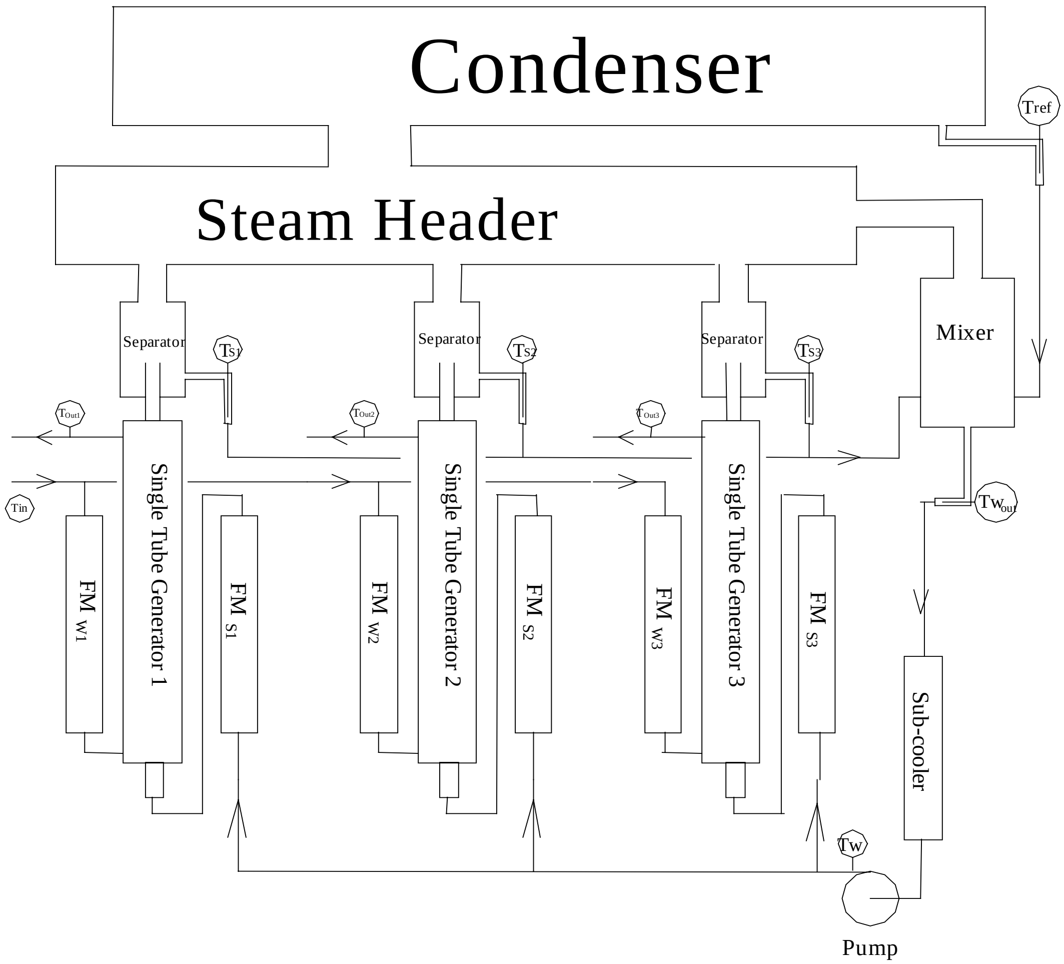

Figure 1. Instrumentation for Generator Tube Test Device

This test device was built to simulate the conditions inside the generator of an absorption chiller. The generator operates under a vacuum and a completely airtight testing apparatus had to be built. Shell and single tube heat exchangers (Figure 2), were used to simulate the heat transfer conditions that take place in the generator. The working fluid flows inside the inner tube and the heating water is pumped through the annulus between the tubes. The inside diameter of the annulus is 1/2 inch and the outside diameter is 5/8 inch.

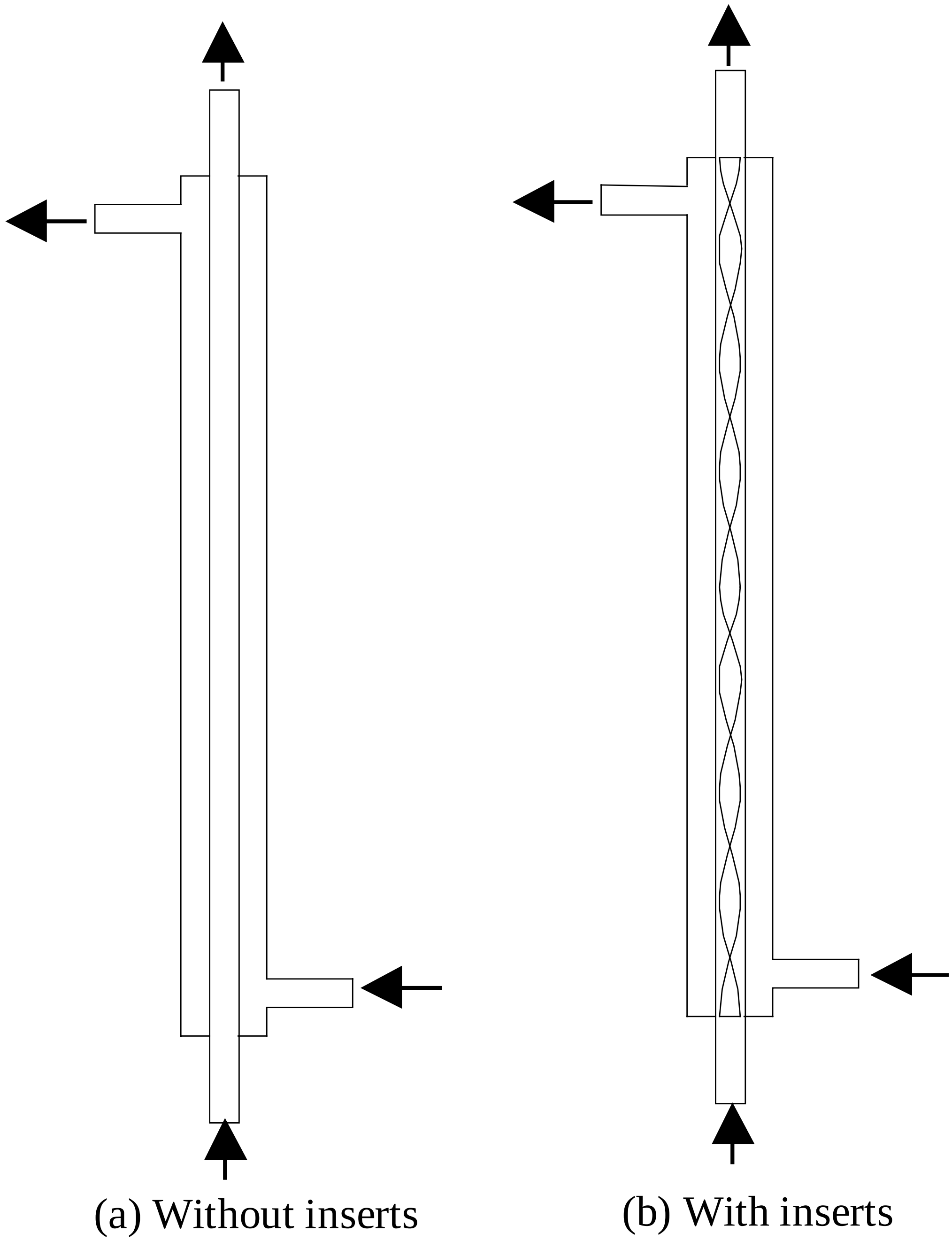

Figure 2. Views of Generator Tubes

In the actual generator, there are a total of 61 inner tubes, each ½ inch in diameter. The tubes are installed vertically inside a single 6-inch diameter shell. In the test device, three identical concentric tube heat exchangers (Figure 3) are installed in parallel. The data from all of the tubes were compared and averaged to account for variations in flow and heat transfer among the individual tubes.

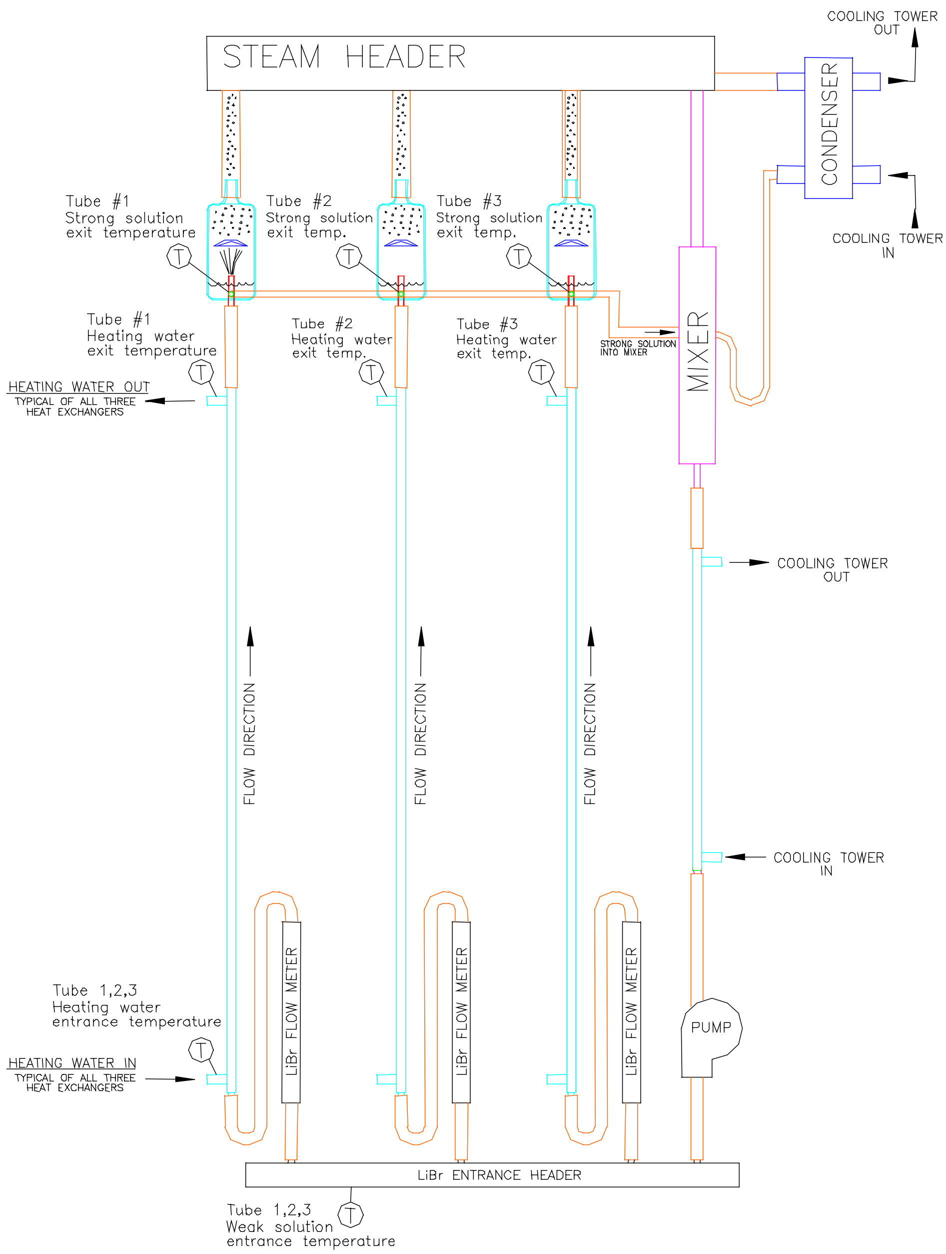

Figure 3. Generator Tube Test Device

The temperatures of the heating water, at the inlet to the device and at the outlet of each outer tube, were measured. In the same way, the temperatures of the working fluid (H_2O or LiBr / H_2O), at the inlet to the device and the outlet of each inner tube, were measured.

The volumetric flow rates of the heating water and of the working fluid were monitored, using rotameters, and maintained at constant levels. The flow of both the heating water and the working fluid were maintained at levels that duplicate the conditions inside the generator of an absorption chiller. By measuring the flow rate and the temperature change of the heating water, the rate of heat transfer from the heating water to the working fluid can be calculated. In addition, the heat transfer coefficient between the heating water and the outside of the tube can be calculated using standard formulas for turbulent flow in an annulus.

These results are then combined to determine the inside heat transfer coefficient. This is the quantity that the Generator Tube Test Device is designed to measure. The detailed calculation procedure is given in the previous sections.

At the top of each tube is a glass cylinder, referred to as the separator (Figure 1). LiBr / H_2O solution as well as steam that boiled off of the solution enters the separators from the bottom. The boiling action in the tubes causes the fluid to be pushed into the separator at high speed. The separators allow the steam to move up into the header tube, located above the separators, but prevents any liquid from entering the header.

Near the bottom of each separator, a small tube leads to a second header behind all three separators. Through these tubes the remaining solution from the inner tubes in the heat exchangers flows to the mixing chamber. This LiBr / H_2O mixture is referred to as the "Strong Solution", because the LiBr concentration increases when water vapor is removed from the solution.

The steam from all three separators enters a common header where it mixes and moves on to the condenser. In the condenser, heat is removed and the steam temperature drops to the saturation point. This causes the steam to condense and become liquid H_2O. The water then flows out of the condenser into a chamber called the absorber (mixer). In the absorber the water combines with the strong LiBr / H_2O solution coming from the separators. The strong solution and the water combine to produce the Weak Solution, named for its lower LiBr concentration. The Weak Solution leaves the mixer and flows through a small, shell and tube, heat exchanger that can be used to sub-cool the solution. After the sub-cooler, the solution passes through a pump, where it is pushed on through the flow meters of each tube. At this point the process begins all over again.

The spiral strip inserts are made of 22 gauge, type 308 stainless steel. The steel was sheared into strips 13 / 32(0.406) inch wide by 60 inch long. For the test device, 3 inserts were made. The flat, thin strips were placed under tension by 90 lbs. hanging weight. Under this tension, the weight and steel strip were rotated on one end, creating a spiral insert for the inner tubes in the test device. Each strip was rotated 180^o a total of 48 times in order to achieve the desired final twist ratio (y) of 3.0.

The twist ratio is calculated using the following formulas:

$$y = \frac{H}{D}$$

and

$$H = \frac{L}{N}$$

Where D is the diameter (width) of the strips and H is equal to the length (L) divided by the number of 180° turns (N). For these tests, the length is 60 inches, the number of 180° turns is 48 and the width of the strip is 13/32 (0.406) inches. This results in a twist ratio of 3.

In order to create a baseline for comparison, the Generator Tube Test Device was first run with water as the working fluid and no inserts to augment the heat transfer. The flow rates of the heating water and the working fluid were at levels simulating the conditions inside the generator of a 20 ton absorption chiller. These conditions are as follows. For the heating water a volumetric flow rate of 0.6 gallons per minute (GPM) was used for each of the three generator tubes. This was set manually at the beginning of the experiment and monitored throughout the test time period. The flow rate of 0.6 GPM creates turbulent flow on the outer surface of the inner tube, creating an outside heat transfer coefficient that is 8 to 10 times larger than the inside heat transfer coefficient that is being measured. This large difference insures that any anomalies caused by unforeseen imperfections in the device will not greatly effect the results of the calculations for the inside heat transfer coefficient.

For the working fluid, a volumetric flow rate of 0.045 GPM was set using a high precision flow meter. These conditions closely simulate the flow on the inside of the generator tubes.

A temperature range for the heating water of 160°F to 210°F was used during the tests. The temperatures were maintained by a combination of parabolic trough and ICPC collector arrays with a natural gas back up boiler. While the spiral inserts have a large effect on the average values of the inside heat transfer coefficient, the variation with heating water temperature appears to be smaller. As a result, this temperature range is adequate for this study. The variation in temperature during an individual test was usually 15-20°F. This is a small enough change to assume steady state conditions.

After the baseline data were collected, inserts were placed in the inner tubes and the same test conditions were used for the next set of experiments. After an appropriate number of runs, the water and the inserts were removed and a $LiBr/H_2O$ solution with a 56 percent concentration of LiBr was placed in the test device. The same conditions used for the water runs were repeated for the $LiBr/H_2O$ solution and data were collected. For the final set of tests, the inserts were placed in the tubes and data were collected.

The generator tube test device operates with the same flow rates of both the heating water and the working fluids as in the generator of an actual absorption chiller. As is seen in Figure 3, three shell and tube type generator tubes are used in parallel. Each tube has temperature sensors at the inlets and outlets of the heating water and the working fluid. Three tubes in parallel were used in order to obtain an average from the results of all three. This is required due to the inherent instability of the boiling process and the difficulty of maintaining consistent conditions. The test device was run in four configurations in the inside tubes as follows:

- Water only, no inserts
- Water with inserts
- LiBr / H_2O only, no inserts
- LiBr / H_2O with inserts

The no insert data are used to establish a baseline for the inside heat transfer coefficient in a standard absorption chiller. Then the spiral inserts were placed inside the inner tubes in all three of the generator tubes and additional tests were run. For all four test configurations, temperature readings were recorded at on-minute time intervals by the data acquisition system. The flow rates were recorded manually from the rotameters at the beginning and end of each test. Temperature and flow rate data were placed in a spreadsheet that calculates the overall and internal heat transfer coefficients for each tube. Section 3.01 describes the calculation procedure.

In comparing the data with water as the working fluid, it was found that the inside heat transfer coefficient increased by as much as 200 percent, with an average increase of 175 percent. The enhanced mixing created by the spiral inserts caused this increase in heat transfer coefficient. Furthermore, in observing the boiling fluid leaving the top of the inner tubes into the separators, it was clear that the presence of the inserts caused the boiling process to be much more stable. This stability is the result of both the augmented heat transfer coefficient and the path that the slugs of steam had to follow in flowing through the tubes. Without an insert, slugs of steam would form erratically and push their way straight up to the outlet of the tube. Each slug would also push a quantity of liquid out the top of the tube. This resulted in instability in the boiling as well as in the heat transfer rates. With the presence of the inserts, the slugs of vapor that form follow a spiral path as they flow through the tubes. This reduces the amount of liquid that is pushed out of the top of the tube and greatly stabilizes the boiling and heat transfer process.

In comparing the experimental data with LiBr / H_2O as the working fluid, it is found that the inside heat transfer coefficient is increased by an average of 47 percent due to the presence of the inserts. The same observations were made concerning the fluid flowing out of the tubes in the test device as with water as the working fluid. The large difference in the increased heat transfer coefficient between the water runs and the LiBr / H_2O runs can be attributed to the physical characteristics of the fluids. The data presented here confirm the well-known fact that water is an excellent medium for transferring heat. However, we believe that the enhancement achieved with LiBr / H_2O as the working fluid is a significant result in improving the performance of solar driven absorption chillers.

All of the data from the Generator Tube Test Device are summarized in Table 2. Listed in the table are the average values of temperatures and heat transfer coefficients for each test run. The device has a subcooler section that can be used to cool the working fluid before it enters the inner tube. During some of the test runs, the subcooler was used to lower the temperature of the working fluid before it entered the tubes. It was found that in the cases where LiBr / H_2O was the working fluid, the subcooling caused the boiling process to become more erratic and unstable. It was determined that the data collected in these test runs were unreliable and, therefore, were not included in the calculated averages discussed below. As for the test runs with water as the working fluid, the subcooler had little effect, so the data were used in calculating the averages.

From Table 2 it is seen that for water as the working fluid with no inserts, the average inside heat transfer coefficient for the three runs listed is 41.0 Btu / $hrft^2F$. For the five runs where water is the working fluid and inserts are used, the average heat transfer coefficient is 112.7 Btu / $hrft^2F$. This constitutes an average increase in inside heat transfer coefficient of 175 percent, due to the presence of the inserts. Looking at the data for LiBr / H_2O solution as the working fluid, with no inserts and no subcooling of the fluid, the average heat transfer coefficient is 41.2 Btu / $hrft^2F$ (note that this is the same as the runs with water as the working fluid). With inserts the average inside heat transfer coefficient increases to 60.7 Btu / $hrft^2F$. This is an average increase of 47 percent. As mentioned above, the lower percentage increase can be attributed to the poorer heat transfer characteristics of the LiBr / H_2O solution compared to that of pure water.

Table 2. Generator Tube Testing Summary

Date	Fluid	Inserts	Avg Tgen	Avg Tcond	Avg h_i	Avg Tin	Sub-Cooling
1/18/01	H_2O	No	155.0	55.5	42.5	133.7	No
1/22/01	H_2O	No	158.9	61.0	35.0	134.3	No
1/23/01	H_2O	No	142.5	69.0	45.6	94.0	Yes
2/6/01	H_2O	Yes	153.6	63.5	93.5	112.5	Yes
2/8/01	H_2O	Yes	145.9	62.5	123.8	108.6	Yes
2/27/01	H_2O	Yes	154.2	65.6	117.2	113.7	Yes
3/1/01	H_2O	Yes	133.0	66.2	97.8	104.5	Yes
3/13/01	H_2O	Yes	192.7	72.6	131.0	131.8	Yes
5/1/01	LiBr	Yes	183.9	75.4	46.1	157.5	Yes
5/3/01	LiBr	Yes	186.6	73.8	51.6	126.0	Yes
5/10/01	LiBr	Yes	182.2	73.2	38.8	119.4	Yes
5/17/01	LiBr	Yes	198.0	76.8	47.5	131.1	Yes
5/23/01	LiBr	Yes	205.8	80.9	44.8	137.4	Yes
5/24/01	LiBr	Yes	206.5	81.2	42.5	139.2	Yes
5/30/01	LiBr	Yes	206.3	80.6	47.2	136.1	Yes
6/7/01	LiBr	Yes	205.9	81.1	46.0	135.0	Yes
6/14/01	LiBr	Yes	207.9	82.5	60.7	178.5	No
6/20/01	LiBr	Yes	206.5	82.6	43.6	131.1	Yes
6/21/01	LiBr	Yes	203.7	83.3	55.5	173.4	No
7/3/01	LiBr	Yes	201.6	83.7	66.1	174.5	No
7/5/01	LiBr	Yes	202.1	83.3	66.8	173.4	No
7/10/01	LiBr	Yes	201.8	82.2	54.4	170.1	No
7/24/01	LiBr	No	184.1	82.6	47.5	160.0	Yes
7/26/01	LiBr	No	198.5	80.4	43.7	164.3	No
8/22/01	LiBr	No	180.6	78.1	34.9	145.3	No
8/23/01	LiBr	No	190.6	79.2	44.3	158.6	No
9/20/01	LiBr	No	191.1	80.5	41.3	162.4	No
10/2/01	LiBr	No	200.0	81.6	42.0	164.1	No
						Avg.Increase	
Highlighted dates used to form averages for LiBr-H2O.						h_i =	47%
Highlighted dates used to form averages for water.						h_i =	175%

Figure 4 shows the average heat transfer coefficients for the test runs with, and without, inserts and water as the working fluid. The 175 percent average increase can be observed for the whole test run period.

Figure 5 shows the average heat transfer coefficients for the test runs with and without inserts and LiBr / H_2O as the working fluid. The 47 percent increase in heat transfer coefficient can be seen over nearly the entire period of the tests.

A copy of the spread sheets for the testing on June 21, 2001 is included in Appendix I as a sample of the testing and calculation procedures.

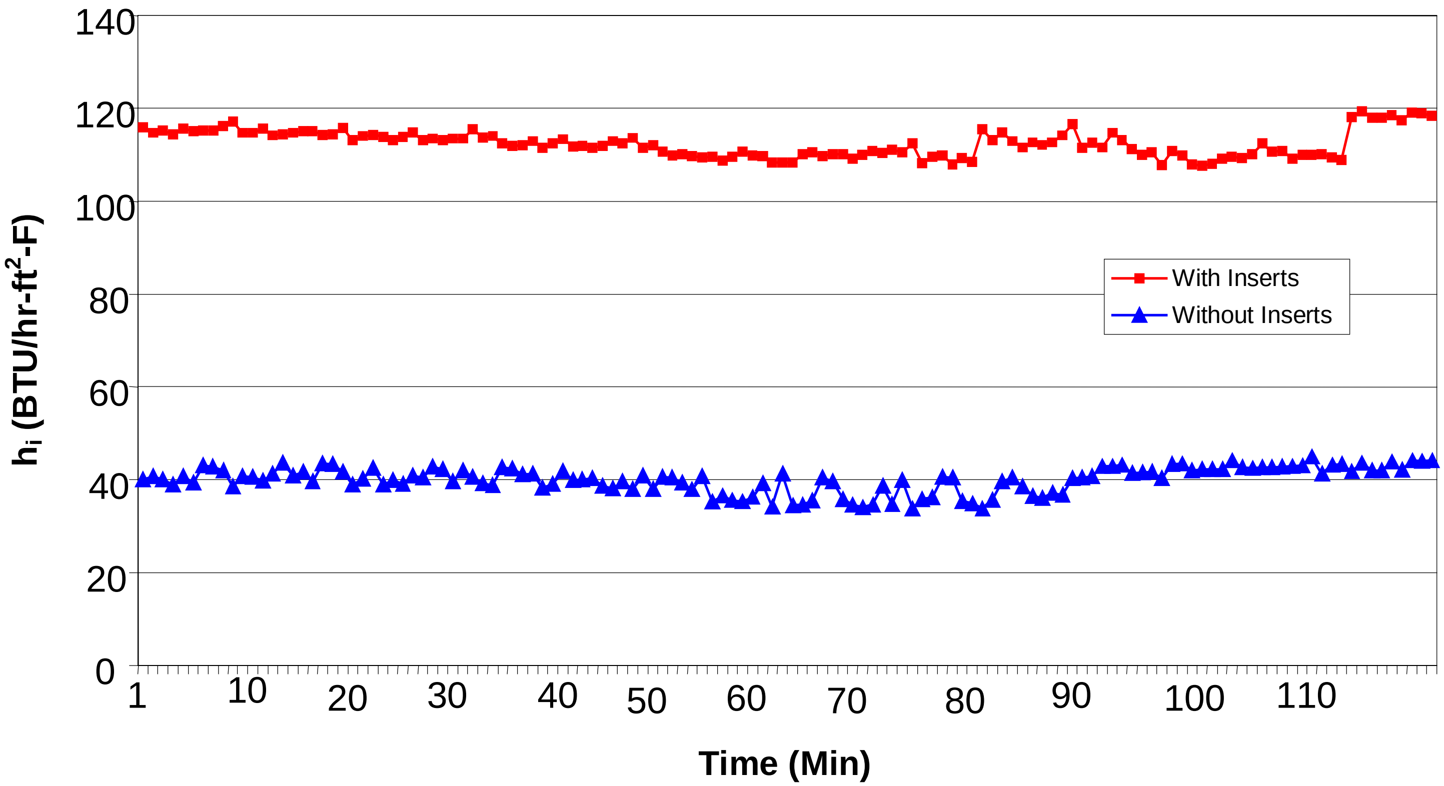

Figure 4. Average Internal Heat Transfer Coefficients for Water

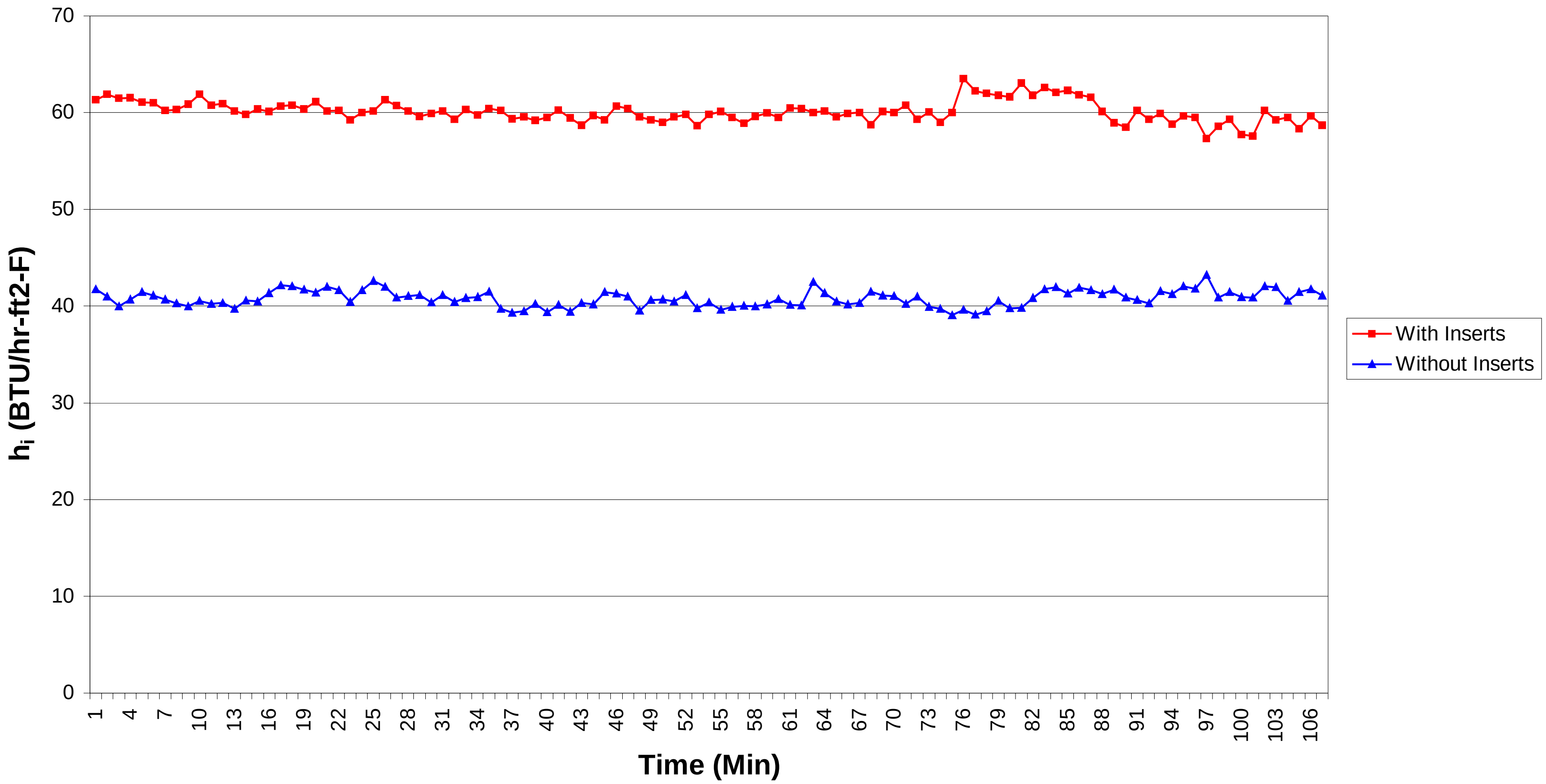

Figure 5. Average Internal Heat Transfer Coefficients for Li-H_2O Solution

2.2. Modification of the Double Effect Chiller

As part of a previous California Energy Commission project, the contractor modified a 20-ton McQuay-Sanyo 2E, natural gas fired absorption chiller. The modification involved replacing the original natural gas-fired high temperature generator with a water-fired high temperature generator.

The main objective of this project is to optimize the design of a solar-fired, double-effect (2E) absorption chiller for use in commercial HVAC systems. The most important component of a 2E-absorption chiller is the first-stage, or high temperature, generator. The generator, designed and built and tested during the previous project, was a shell and tube type with solar heated water on the shell side and a LiBr / H_2O solution in the tubes. For this project, computer models have been developed to predict the performance of the high temperature generator and to evaluate the effect of various augmentation techniques.

The results of the calculations indicate that the optimum design is obtained by using twisted strip inserts to augment the heat transfer on the inside of the generator tubes. As described in the previous section, a Generator Tube Test Device has been designed and built by the contractor to experimentally determine the effect of incorporating twisted inserts in the generator tubes. The results of the testing indicate that a 47 percent increase in the inside heat transfer coefficient can be achieved with these swirl type inserts.

There are two distinct heat transfer regions within the generator tubes. The first is a single phase, laminar flow heat transfer region and the second is a two phase, or boiling, region. The computer models use appropriate correlations and calculation procedures to predict the heat transfer coefficients for the different regions. Refrigerant vapor is generated in the boiling region and the effectiveness of the generator is improved by minimizing the single-phase region and, thereby, starting the boiling process as soon as possible. Results from the computer models, which have been verified experimentally, indicate that installing twisted, stainless steel strips to produce a swirl flow reduces the laminar flow region from as much as 70 percent of the tube length to as low as 25 to 30 percent of the total length. This means that without augmentation, up to 70 percent of the tube has single phase, laminar flow. The use of inserts to improve the heat transfer inside the tubes causes the transition to boiling to occur much sooner. In this case, only 25 to 30 percent of the tube has single phase flow. This also decreases the magnitude of the log mean temperature difference and makes it possible to operate the solar collectors and the solar storage tank at significantly lower temperatures. These are the type of results that this project was designed to achieve and are very important in terms of optimizing the performance of solar fired absorption chillers.

The high temperature generator in the 20-ton McQuay, water fired, double effect absorption chiller has 61 stainless steel tubes that are 1/2 inch in diameter and 5 ft. long. The tubes are installed in a 6-inch diameter, 5-ft. 8-inch long stainless steel shell. A lithium bromide / water solution is inside the tubes with solar heated water in the shell. The optimization of the generator involves installing twisted, stainless steel strips inside the tubes to augment the heat transfer between the LiBr / H_2O in the tubes and the solar heated water in the shell.

2.2.1. Generator Tube Test Device

In order to determine the effect of spiral inserts in the generator tubes, a generator tube test device was designed and built as part of the project. A detailed description of the device is presented in Section 3.0. The purpose of the device is to experimentally determine the effect of installing twisted stainless steel strips in the stainless steel tubes to augment the heat transfer and improve the performance of the high temperature generator.

A five-part testing program has been completed. The test plan is as follows:

1. Testing with water – no inserts

For the first phase of the testing, the 3 identical tubes in the Generator Tube Test Device were installed, without inserts, and were tested with water, under vacuum, in the tubes. These tests were designed to verify the procedure and to provide baseline test data for the device.

2. Testing with water – with inserts

For the second phase of the testing, twisted stainless steel inserts were installed and tests were conducted with water in the tubes. Data are available for heat transfer in circular tubes with water and with twisted inserts. This provided experimental data that relates to the augmentation achieved by the inserts and to the potential improvement in the performance of the high temperature generator.

3. Testing with LiBr / H_2O – with inserts

For this phase of the testing, the water was removed from the tubes and a 56 percent LiBr / H_2O solution was added to the tubes. The inserts remained in place. Tests were conducted for a range of operating temperatures and heat transfer data were obtained. This is the configuration that was used in the final design of the optimized, high temperature generator.

4. Testing with LiBr / H_2O – no inserts

The 4th phase of the testing involved draining the LiBr / H_2O solution, removing the inserts from the tubes and then replacing the LiBr / H_2O. These tests provided experimental data on the tubes as they were originally installed in the generator. Data from 3 and 4 have been compared to determine the effect of the inserts on the performance of the optimized generator.

2.2.2. Chiller Testing with Optimized Generator

The final phase of the testing program involved field-testing of the HVAC system with the modified generator installed in the McQuay 2E-absorption chiller. The work performed in completing this task included

- Removing the original generator from the double-effect chiller;
- Installing twisted, stainless steel strips in all 61 of the generator tubes;
- Reinstalling the generator in the chiller and recharging the system with the LiBr / H_2O solution.

For most of the 2001 summer, the HVAC system was used to provide air conditioning for our 8,000-ft^2 commercial building with the modified chiller. The objective of this testing was to determine the effect of the optimized generator on:

- Performance and operational stability of the chiller
- Effectiveness and log mean temperature difference of the generator
- Cooling capacity of the chiller
- Coefficient of performance of the chiller

In order to complete the chiller modification work, it was necessary to fabricate the stainless steel inserts that were installed in the tubes of the high temperature generator. The twisted strip inserts are made of 22-gauge type 308 stainless steel. A stainless steel sheet was sheared into strips 13/32 (0.406) inches wide by 60 inches long. The inserts were twisted in a device we built that we call “The Twister”. It is shown schematically in Figure 6. The Twister has a mechanical advantage of 6.5 so an applied force of 15 pounds puts the strip under tension with a force of 97.5 pounds. The individual strips were installed in the device, the load was applied and the wheel turned 25 complete revolutions. The wheel would then turn backward two revolutions leaving 48 twists of 180^o each. This resulted in strips with a twist ratio of 3. A total of 61 of the 13/32 inch by 60 inch long strips were twisted and then inserted in the 1/2 inch stainless steel tubes in the high temperature generator. Figure 7 is a picture of the generator with most of the inserts installed. Figure 8 shows the completed generator, with twisted inserts installed and spot-welded in all 61 of the solution tubes. The generator was then reinstalled in the McQuay chiller. A vacuum pump was used to remove all of the air from the chiller and the chiller was recharged with the LiBr/H_2O solution. The chiller was leak tested, by checking the vacuum over a 1-week period, and determined to be operational and ready for field-testing.

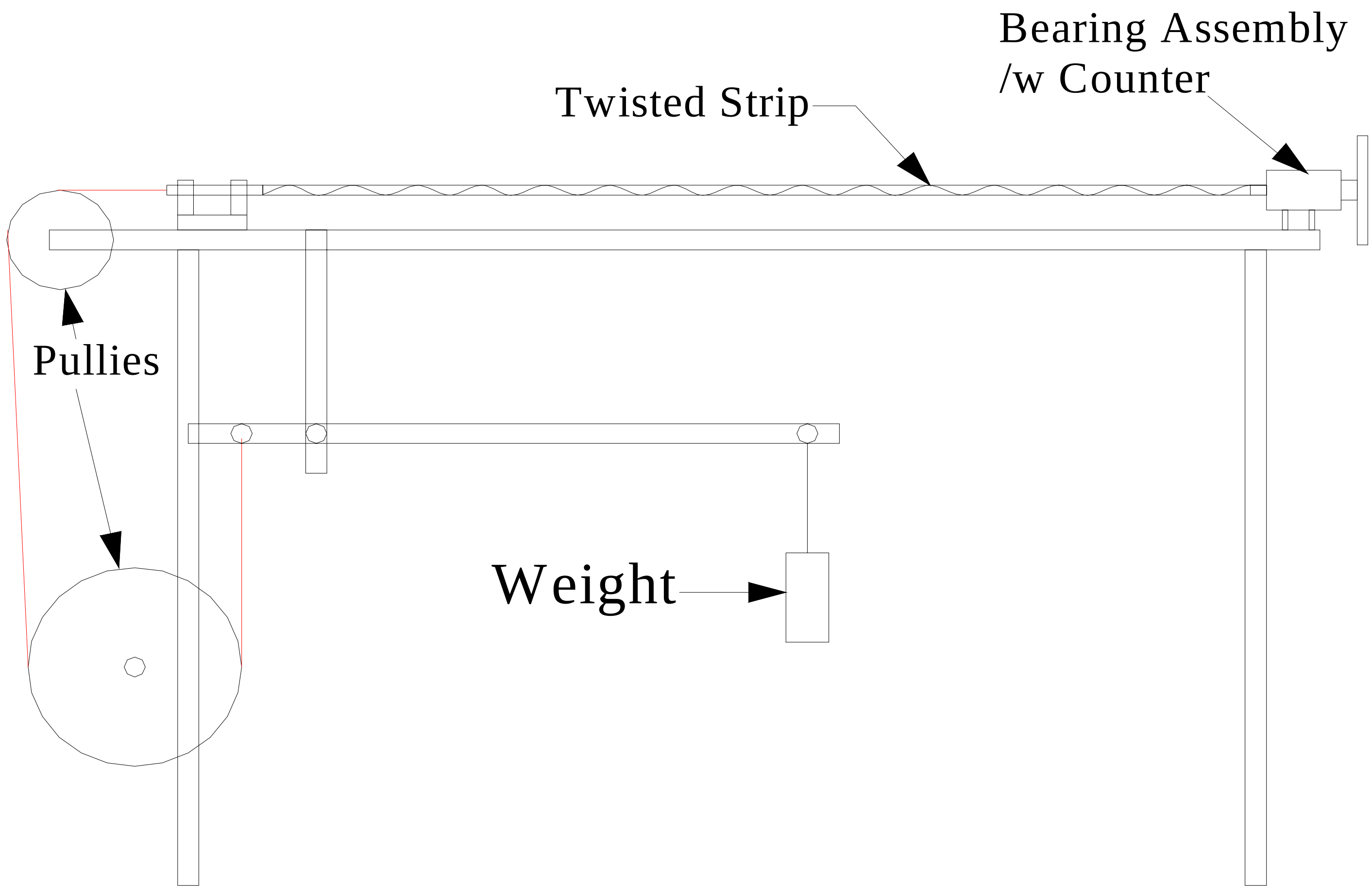

Figure 6. The Twister

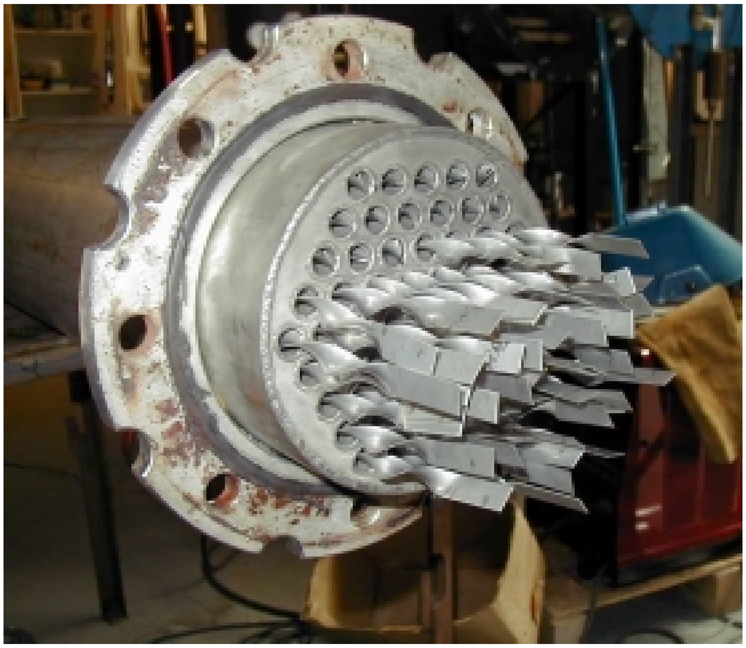

Figure 7. Installing Twisted Strips in High Temperature Generator

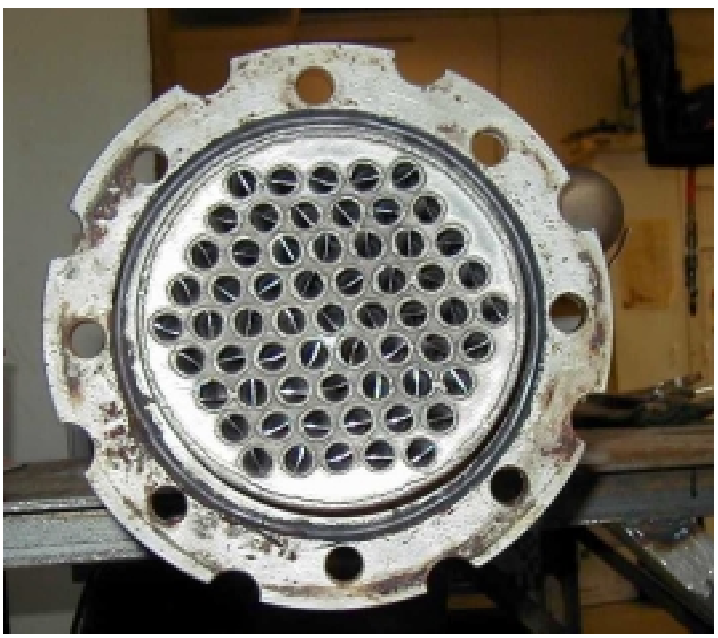

Figure 8. Generator with 61 Twisted Strips Installed

2.3. Test Results with Optimized Generator

The instrumentation that was installed in the chiller is listed in Table 3. A schematic of the double effect McQuay chiller with the location of the sensors is shown in Figure 9. The chiller was operating with the modified generator and this instrumentation starting at the end of June 2001.

Figure 10 shows the results of the chiller testing for August 18, 2001. The average value of the overall heat transfer coefficient, U, is 212.6 Btu / hrft^{2}F. The average value of the log mean temperature difference is 18.3°F.

For comparison, data for the chiller with the original generator are shown in Figure 11 and Figure 12 for July 29, 1998 and August 5, 1998. The fluctuations in the overall heat transfer coefficient are the result of the instability of the boiling process. The operation of the optimized generator is much more stable. In addition the average of the overall heat transfer coefficients were in the range of 140 to 175 Btu / hrft^{2}F compared to the 212.6 value noted above. This is as much as a 50 percent increase in the overall heat transfer coefficient. Equally significant are the average values of the log mean temperature difference. These were between 40°F and 50°F compared to the current value of 18.3°F. This means that the difference between the average generator temperature and the average firing water temperature is reduced by 20 to 30°F.

Another way of evaluating this is that with a LMTD of 50°F the collector array and storage tank have to operate at 310°F in order to achieve an average generator temperature of 260°F. With the optimized generator, when the generator temperature is 260°F, the collector array and storage tank can now operate at an average temperature of 278°F. These are the types of results that we were looking for in the process of optimizing the performance of solar fired double effect absorption chillers. The objective of this project is to lower the operating temperatures of the collectors and storage tanks that are used in conjunction with the absorption chillers. These data indicate that the operation of the modified chiller is significantly improved in comparison to the original chiller. A lot of progress has been made in reaching our ultimate objective of operating this type of equipment at temperatures under 250°F.

2.3.1. Chiller COP

As discussed above, the original water fired generator had 61 stainless steel tubes that were 1/2 inch in diameter and 5 feet long. The stainless steel tubes were installed inside a 6-inch diameter stainless steel shell. A lithium bromide / water solution on the inside of the tubes is heated by high temperature water that is pumped through the shell. The high temperature water is heated by an array of Integrated Compound Parabolic Concentrator (ICPC) vacuum tube solar collectors. There are 336 ICPC collectors with a total area of 1,145 ft^2.

Table 3. McQuay Testing (Optimized Generator) Sensor Description

TC1	Evaporator Temperature
TC2	High Temperature Generator Saturation Temperature
TC3	Condenser Saturation Temperature
TC4	Absorber Weak Solution Temperature
TC5	Low Temperature Generator Outlet Solution Temperature
TC6	High Temperature Generator Inlet Solution Temperature
TC7	High Temperature Generator Outlet Solution Temperature
TC8	Condenser Outlet Cooling Water Temperature
TC9	Not Used
TPR1	High Temperature Generator Water Outlet Temperature
TPR2	Evaporator Chiller Water Temperature
TPR3	Condenser Cooling Water Outlet Temperature
TP1	Delta Generator Water Temperature
TP2	Delta Evaporator Chiller Water Temperature
TP3	Delta Cooling Water Temperature

TC – Thermocouple
TP – Thermopile
TPR - Thermopile Reference Temperature

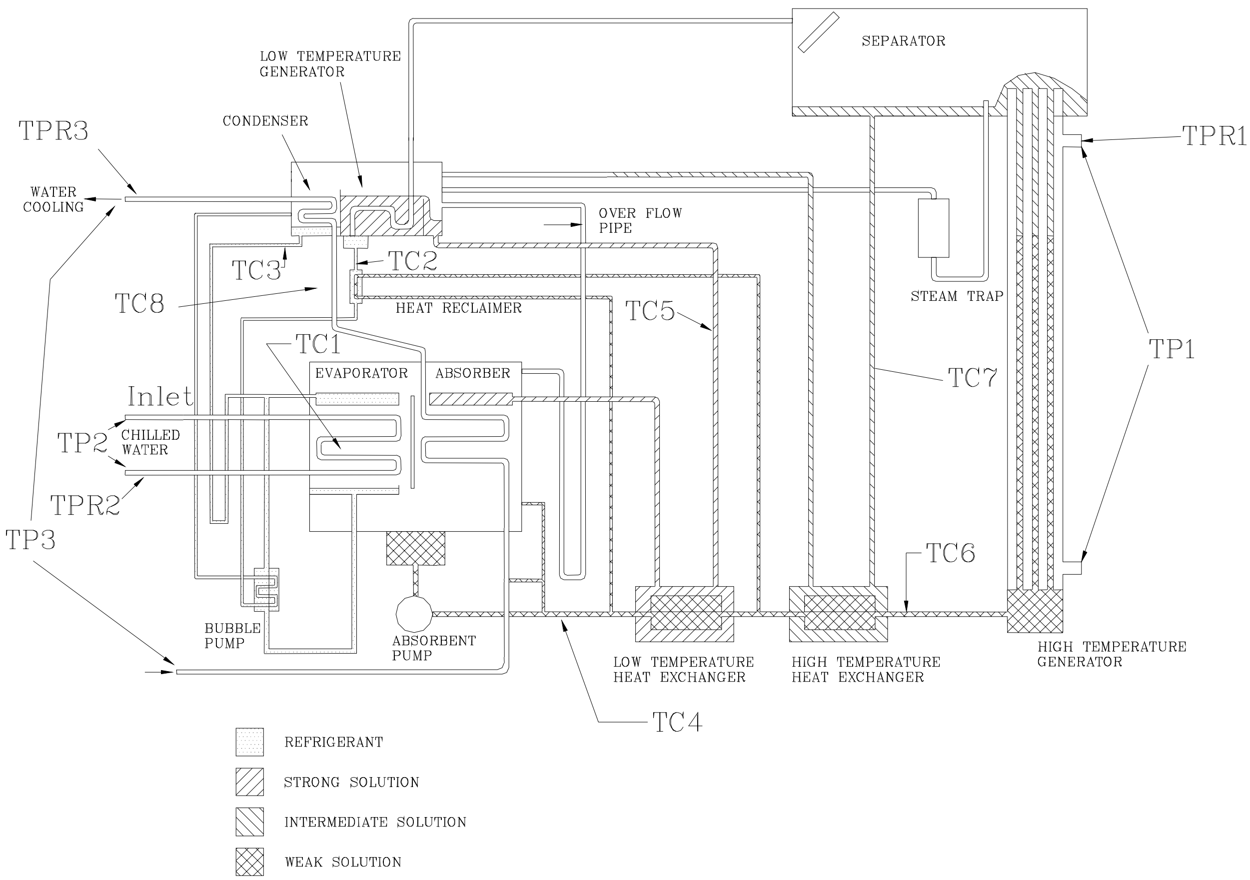

Figure 9. McQuay Chiller with Optimized Generator

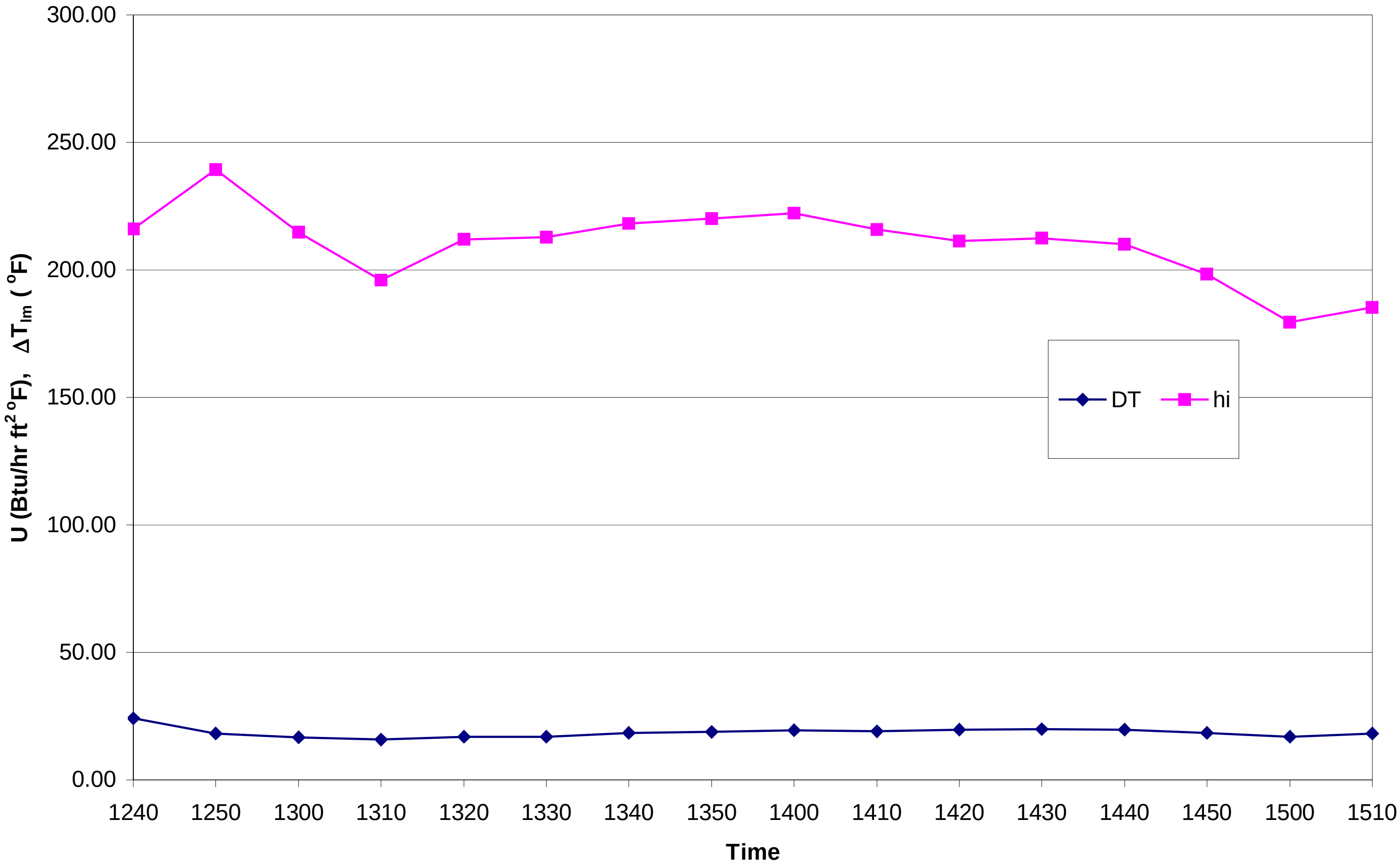

Figure 10. McQuay (08/18/01

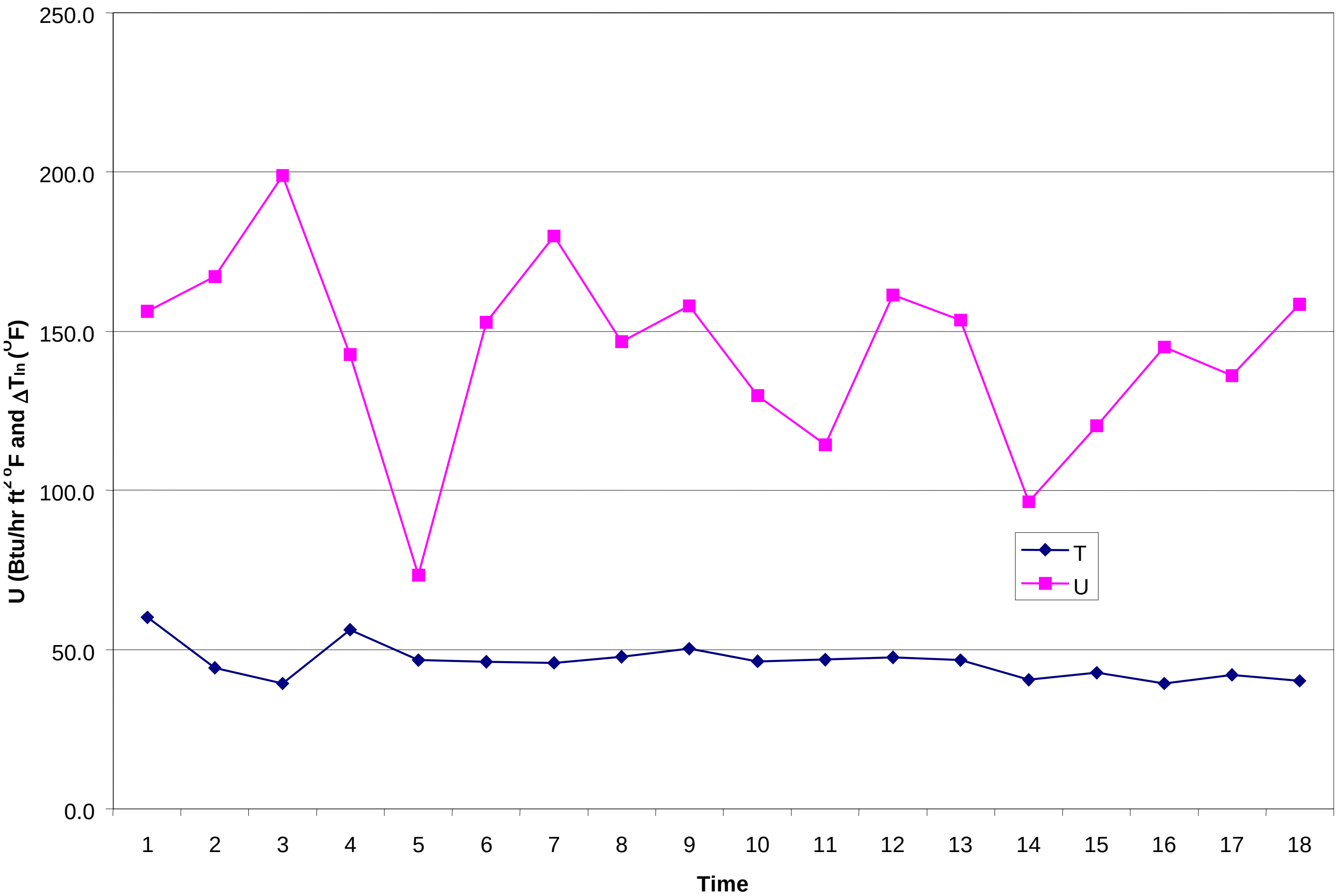

Figure 11. McQuay (07/29/98

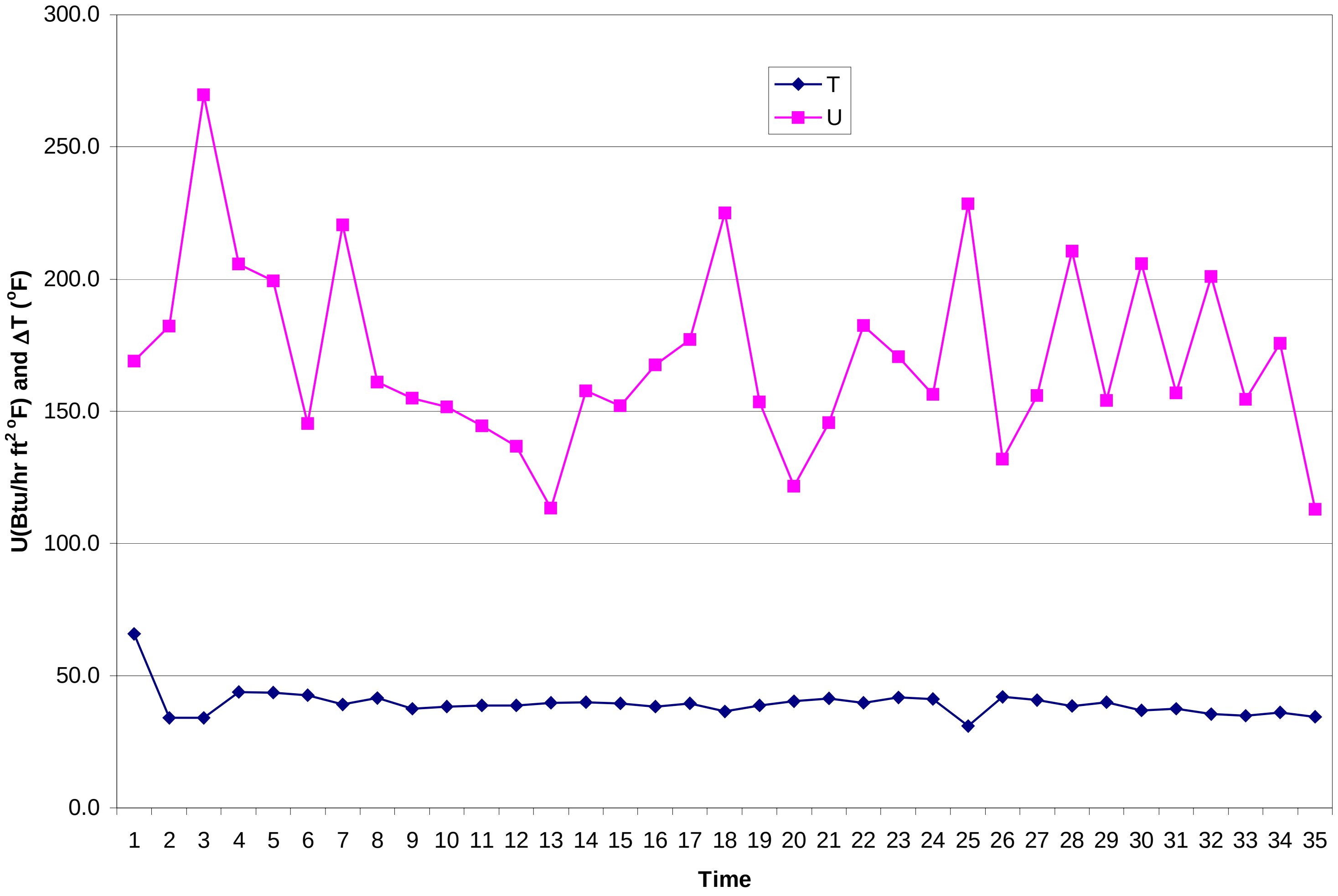

Figure 12. McQuay (08/05/98)

The operating temperature of the collector array and storage tank was typically 310 to 320°F. The primary objective of this project is to optimize the high temperature generator in the McQuay double effect absorption chiller. The modification involved installing twisted stainless steel strips in all 61 of the generator tubes. The purpose of the spiral strips is to augment the heat transfer in the tubes and thereby increase the effectiveness of the high temperature generator. This will allow the collector array and the storage tank to operate at a lower temperature and still maintain a chiller COP of approximately 1.0.

Figure 9 shows a schematic of the McQuay chiller with the water fired high temperature generator and the thermocouples (TC1-TC8), the thermopiles (TP1-TP3) and the thermopile references (TPR1-TPR3) that were used in the testing phase of the project. Additional instrumentation includes turbine and vortex shedding flow meters to measure the flowrates of the hot water through the high temperature generator, the chilled water through the evaporator and the cooling water through the absorber, condenser and cooling tower. The cooling capacity of the chiller is the rate of heat transfer in the evaporator Q_E. This is determined by multiplying the mass-flowrate and specific heat of the chilled water by the temperature drop across the evaporator. The heat input to the chiller is the rate of heat transfer in the high temperature generator Q_G. This is determined by multiplying the mass flow rate and specific heat of the hot water by the temperature difference across the high temperature generator. The Coefficient of Performance (COP) of the chiller is:

$$COP = \frac{\dot{Q}_E}{\dot{Q}_G}$$

For all of the data presented in this report, the measured heat transfer rates are summed over 10-minute time intervals. The calculated COP is then the average value for the same time interval.

During the summer of 2001, the McQuay 20 ton, double effect absorption chiller operated with the modified high temperature generator described above. Temperature and flow rate data were measured while the chiller was operating and actual values of COP were determined. Typical values of COP as a function of time of day are shown in Figure 13 for 8/6/01.

There are several interesting features about the COP data. Between 11:40 and 12:20 the chiller was operating in a transient mode. In the transient mode, the chiller produces colder temperatures and more refrigerant, and therefore more cooling capacity, than the building requires. If the chilled water supply temperature drops below 40°F the generator pump is turned off until the chilled water temperature rises above 45°F. While the generator pump is off the chiller still provides cooling with the excess refrigerant that was generated during the time the generator pump was on. This transient mode will typically occur when the building load is below eight tons. It also produces high COPs because the heat input is zero for the portion of the time interval that the generator is off but the chiller is still producing cooling. As shown in Figure 13 the highest COP, averaged over a 10-minute time interval, was about 1.45.

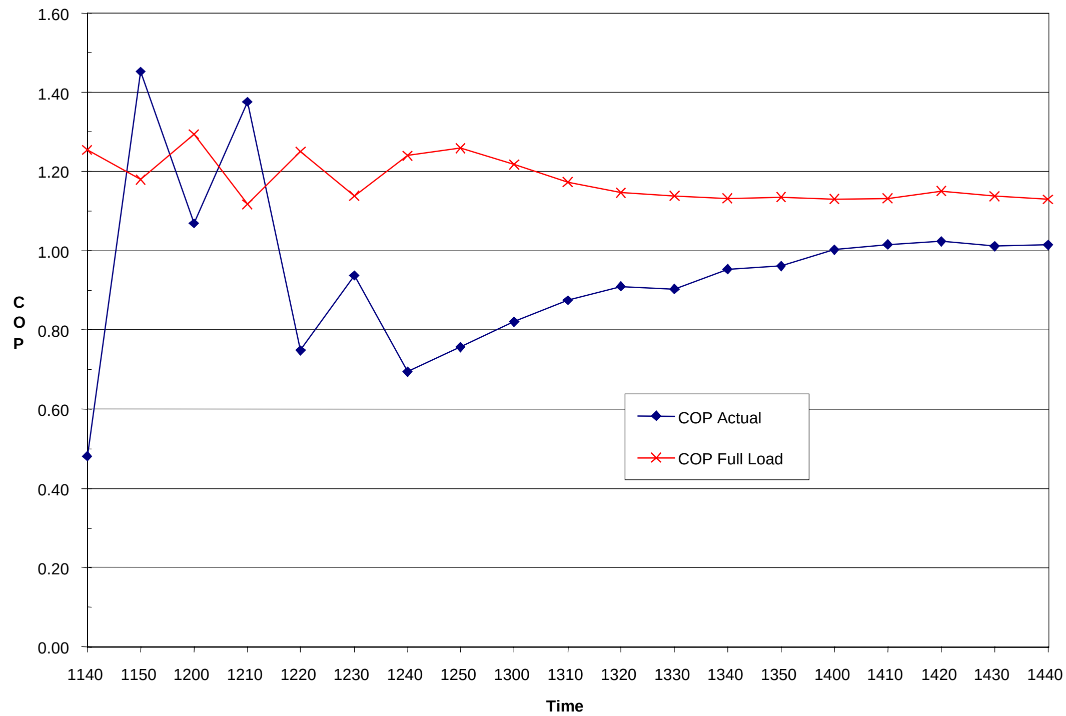

Figure 13. McQuay Testing (08/06/01)

Another important factor is that the performance of the chiller depends on the cooling load of the building. If the chiller is generating enough refrigerant to produce 20 tons of cooling but the load on the building is only 12 tons, the actual COP will be low. We have observed this in both double-effect and single-effect absorption chillers. This means that if the cooling load is increased, for example, by bringing in extra outside air on a hot day, the chiller will provide additional cooling and the COP will increase. In field testing, we do not attempt to control the load. In order to get a measurement of the potential performance of the chiller we have developed a procedure for determining the full load COP. This is described below.

2.3.2. Calculation of Full Load COP

The first step is to write the energy balance and salt mass balance equations for the high temperature generator (Figure 14). The subscripts r, i, and w refer to the refrigerant, the intermediate solution and the weak solution respectively.

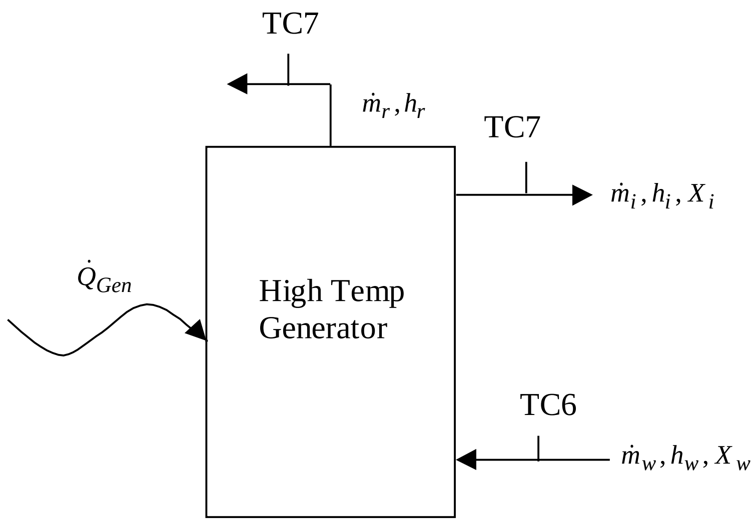

Figure 14. Energy and Mass Balance Equations

The Energy Balance and the Salt Mass Balance equations are

$$\dot{Q}_{Gen} = \dot{m}_r h_r + \dot{m}_i h_i - \dot{m}_w h_w \qquad (1)$$

and

$$\dot{m}_w X_w = \dot{m}_i X_i \qquad (2)$$

Combining (1) and (2) and solving for the refrigerant flow rate gives:

$$\dot{m}_{r,hi\ Gen} = \frac{\dot{Q}_{Gen} - \dot{m}_w \dfrac{X_w}{X_i} h_i + \dot{m}_w h_w}{h_r}$$

From double effect absorption chiller simulations and measuring the weak solution flow rate with a sonic flow meter it has been determined that the approximate weak solution flow rate is 2806 lb / hr.

The concentrations of the weak and intermediate solutions are determined by the temperatures and pressures of the solutions. Refering to Figure 9, the temperature of the intermediate solution is measured by TC7 and the pressure is the saturation pressure of the refrigerant at TC2. Similary, for the weak solution, the temperature is measured by TC4 and the pressure is the saturation pressure of the refrigerant at TC1.

The next step is to calculate the refrigerant flow rate in the low temperature generator. This is determined by the heat transferred in the condenser as follows:

$$\dot{m}_{r,low\ gen} = \frac{\dot{Q}_{Cond}}{h_g - h_l}$$

The enthalpies of the saturated vapor h_g and the saturated liquid h_l are determined at temperature TC3.

The total refrigerant flow rate is the sum of the flow rates in the low and high temperature generators.

$$\dot{m}_r = \dot{m}_{r,high\ gen} + \dot{m}_{r,low\ gen}$$

Now the full load cooling capacity of the chiller can be calculated by:

$$\dot{Q}_{Full\ Load} = \dot{m}_r (h_g - h_l)$$

The enthalpies of the saturated vapor h_g and the saturated liquid h_l are determined at temperature TC1.

The full load COP is calculated by:

$$COP_{Full\ Load} = \frac{\dot{Q}_{Full\ Load}}{\dot{Q}_{Gen}}$$

Values of the full load COP are shown on Figure 13 for 8/6/01. In the afternoon the actual measured COP has a maximum value of 1.02. The maximum value of the full load COP is 1.15. This is a valid indication of the performance of the chiller. It shows that the chiller is generating more refrigerant than is being evaporated. By adding another air handler to the HVAC system the chiller would provide more cooling to the building and would be operating with a higher COP.

Additional Field Testing Data for the last nine days in August 2001 are presented in Figures 15-32, Appendix II. For each of these days there is one plot of the inside heat transfer coefficient and the LMTD. A second graph shows the actual and full load COPs. The McQuay chiller is operating efficiently and in a repeatable manner. The inside heat transfer coefficient is approximately 200 Btu/hrft^{2}F and the LMTD is about 20^{o}F. The actual COP peaks out at 1.05 (on 8/28/01) and the full load COP is about 1.2. These are important accomplishments.

For most of the days when the chiller was operating, it was providing all of the cooling for the building. The nominal capacity of the chiller is 20 tons, but most of the time the building load peaked at about 12 tons. Figure 33 is a typical plot of cooling capacity during the day on 8/28/01. In the afternoon, the chiller is producing 12 tons of cooling with a LMTD of about 20^{o}F.

3.0 Conclusions and Benefits to California

This project was concerned with improving the performance of solar-fired, double effect HVAC systems by optimizing the design of the high temperature generators in absorption chillers. The results of bench testing of generator tubes and field testing of a nominal 20 ton absorption chiller demonstrate that the chiller performance can be improved by installing twisted inserts in the generator tubes. Specifically, the project showed that the average mean temperature difference can be reduced from between 40-50°F to 18°F by installing inserts in the tubes.

The chiller still operates with an actual COP of approximately 1.0 and provides all of the cooling required by an 8,000 ft^2 building. These are very significant results with regard to improving the performance of the chiller. The project went a long way toward achieving the ultimate objective of operating the first-stage generator at approximately 250°F

3.1. Project Technical Conclusions

The overall technical objective of this project was to optimize the performance of a solar fired, double effect absorption chiller. The specific technical objectives were to achieve

- A COP in the range of 1.2 to 1.4 when the collector array and first-stage generator are operated at high temperature (collectors at approximately 300°F, generator at 250-260°F)
- A COP in the range of 1.1 to 1.2 when the collector array and first-stage generator are operated at low temperature (collectors at 260-270°F, generator at 240-250°F).
- A cooling capacity of 16 to 18 tons at both high and low operating temperatures.

Under a previous Energy Technologies Advancement Program (ETAP) contract with the California Energy Commission, the contractor converted a nominal 20 ton, gas fired, double effect absorption chiller to a solar fired chiller. The modified chiller was fired by hot water from an array of vacuum tube type solar collectors. The solar fired absorption chiller provided up to 17 tons of cooling and achieved a maximum cooling coefficient of performance (COP) of 1.2. The collectors and the storage tank operated at approximately 300°F. The difference in temperature between the storage tank and the lithium bromide / water solution in the first-stage generator was approximately 50°F.

The work on this project involved improving the performance of the double effect chiller by optimizing the water fired, first-stage generator. The optimization work included:

- fabricating twisted stainless steel inserts
- building a generator tube test device
- performing bench tests to measure the performance of the generator tubes
- installing inserts in all 61 of the stainless steel tubes in the first-stage generator of the water fired chiller
- operating the solar HVAC system to provide air conditioning for an 8,000 ft^2 commercial building.

The HVAC system with the optimized chiller provided the entire cooling load for the building. The chiller achieved an operating COP of 1.0 and a full load COP of 1.1 to 1.25. The building was normally requiring about 12 tons of cooling during the field tests. However, the chiller was generating enough refrigerant to provide up to 15 tons of cooling.

The collector array and storage tank operated at approximately 270°F. The difference in temperature between the storage tank and the lithium bromide / water solution in the first-stage generator was approximately 20°F.

The performance of the optimized chiller was significantly improved over that of the original solar-fired chiller. Essentially all of the technical objectives of the project have been achieved.

3.2. Project Economic Conclusions

The overall economic objective of this project was to lower the cost of the chiller, and the complete solar HVAC system, by lowering the operating temperatures of the equipment to approximately 250°F. The ultimate targets were met as follows:

- A solar fired chiller cost of $1,100 / ton
- A complete solar HVAC system cost of $4,500 / ton
- A simple payback for the complete solar HVAC system of 4 years or less

The Contractor does research and development work on solar absorption chillers and designs and installs solar HVAC systems. However, the Contractor is not an equipment manufacturer.

During the course of the project the Contractor met with representatives of Trane and American Yazaki and communicated with other manufacturers of absorption chillers. We also completed preliminary design for several potential customers. There is a significant, renewed interest in solar fired absorption HVAC technology.

The cost of the original double effect absorption chiller purchased by the contractor was about $1,700 per ton. It is estimated that the new high temperature generator could reduce the cost of the chiller to under $1,400 per ton. This is based on the following facts.

- The water fired chiller uses about half of the lithium bromide / water solution of the original chiller
- The design of the new generator is simpler and does not require any combustion equipment.

As improvements are made in the commercially available equipment required for solar HVAC systems, it will be possible to achieve the economic objectives of this project.

3.3. Benefits to California

The net reduction in demand resulting from the use of solar HVAC systems is about 1.3kW per ton or 26kW for a 20-ton system. The installation of just 50 solar HVAC system per year, with and average capacity of 20 tons, would result in an annual peak demand reduction of 1.3 MW. A recently completed Market Study by Sun Utility Network identified two target markets for solar HVAC. The study concluded that solar HVAC must be part of a comprehensive energy strategy and that a large number of systems could be installed by 2010.

4.0 References

1. Horg, S.W. and Bergles A.E., “Augmentation of Laminar Flow Heat Transfer by Means of Twisted-Tape Inserts”, Journal of Heat Transfer, Volume 98, pp. 251-256, 1976.

2. Ozisik, M. Necati, “Heat Transfer – A Basic Approach”, McGraw-Hill Book Company, 1985.

3. Jensen, Andrew Sigurd, “Heat Transfer Augmentation in Absorption Chiller Generators”, Masters Thesis, CSU, Sacramento, Fall 2001.

Appendix I

Generator Tube Testing Data Reduction Spread Sheets

Test Calculations w/ LiBr on 6/21/2001 (w/ Inserts)

Heating Water Flow and Heat Transfer

Density = 62.26 lb/ft^3

Area = 0.63268 ft^2

D_o 0.05192 ft

D_i = 0.04167 ft

Length = 4.83 ft

Flow Rate Heating Water	Flow Rate Boiling Water		
0.6 gal/min	67 mm Hg	V = 6388 ft/hr	Di = 0.5 0.04167 0.00174
Vol Flow-1	Flow Rate= 0.05487 Gal/min		Do = 0.623 0.05192 0.0027
0.0013368 ft^3/sec	m = 27.41 lb/hr	Re = 5538	Area = 0.00075 ft^2
Mass Flow-1		Nu = 27.2	v = 0.01182 ft^2/hr
299.6 Lb_m/hr		h_O = 1043	Pr = 1.82
			k = 0.3930

		Tube-1	Entire Tube Length Tube-2	Tube-3		Entire Tube Length			Entire Tube Length		Tube-1	Entire Tube Length Tube-2	Tube-3	Tubes 1&3 Only
Date	Time	Q_1	Q_2	Q_3	Delta T_1	Delta T_2	Delta T_3	U_1	U_2	U_3	h_i	h_i	h_i	Avg. h_i
6/21/01	1:50:32 PM	707.1	593.3	674.2	21.56	21.36	21.67	52	44	49	**55**	**46**	**52**	53
6/21/01	1:51:32 PM	707.1	608.2	707.1	21.69	21.36	21.45	52	45	52	**54**	**47**	**55**	55
6/21/01	1:52:32 PM	707.1	608.2	674.2	21.56	21.22	21.45	52	45	50	**55**	**47**	**52**	53
6/21/01	1:53:32 PM	725.1	608.2	707.1	21.55	21.45	21.52	53	45	52	**56**	**47**	**55**	55
6/21/01	1:54:32 PM	707.1	608.2	674.2	21.49	21.18	21.72	52	45	49	**55**	**47**	**51**	53
6/21/01	1:55:32 PM	740.1	641.2	692.1	21.78	21.11	21.78	54	48	50	**57**	**50**	**53**	55
6/21/01	1:56:32 PM	722.1	623.2	707.1	22.03	21.32	21.79	52	46	51	**55**	**48**	**54**	54
6/21/01	1:57:32 PM	722.1	623.2	689.1	21.92	21.45	21.99	52	46	50	**55**	**48**	**52**	53
6/21/01	1:58:32 PM	740.1	641.2	707.1	21.71	21.64	21.95	54	47	51	**57**	**49**	**54**	55
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6/21/01	2:04:32 PM	740.1	626.2	725.1	21.75	21.45	22.09	54	46	52	**57**	**48**	**55**	56
6/21/01	2:05:32 PM	725.1	593.3	692.1	21.46	21.49	22.07	53	44	50	**56**	**46**	**52**	54
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6/21/01	3:15:31 PM	755.1	641.2	740.1	22.49	22.48	22.82	53	45	51	**56**	**47**	**54**	55
6/21/01	3:16:31 PM	773.0	656.2	755.1	22.34	22.45	22.76	55	46	52	**58**	**48**	**55**	56
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6/21/01	3:25:31 PM	773.0	656.2	755.1	22.87	22.60	22.91	53	46	52	**56**	**48**	**55**	56

6/21/01	3:26:31 PM	773.0	659.2	740.1	22.74	22.66	23.18	54	46	50	**57**	**48**	**53**	55
6/21/01	3:27:31 PM	788.0	656.2	755.1	22.68	22.88	23.22	55	45	51	**58**	**47**	**54**	56
6/21/01	3:28:31 PM	788.0	656.2	740.1	22.68	22.47	23.09	55	46	51	**58**	**48**	**53**	56
6/21/01	3:29:31 PM	758.1	659.2	758.1	22.97	22.49	23.07	52	46	52	**55**	**48**	**55**	55
6/21/01	3:30:31 PM	791.0	725.1	725.1	22.79	22.20	23.13	55	52	50	**58**	**54**	**52**	55
6/21/01	3:31:31 PM	791.0	674.2	824.0	22.67	22.33	22.91	55	48	57	**58**	**50**	**60**	59
6/21/01	3:32:31 PM	806.0	692.1	773.0	22.81	22.39	23.24	56	49	53	**59**	**51**	**55**	57
6/21/01	3:33:31 PM	788.0	689.1	773.0	22.85	22.44	23.08	55	49	53	**58**	**51**	**56**	57
6/21/01	3:34:31 PM	788.0	689.1	740.1	23.00	22.55	23.23	54	48	50	**57**	**51**	**53**	55
6/21/01	3:35:31 PM	773.0	641.2	755.1	22.63	22.49	22.87	54	45	52	**57**	**47**	**55**	56
6/21/01	3:36:31 PM	773.0	641.2	740.1	22.37	22.58	22.75	55	45	51	**58**	**47**	**54**	56
6/21/01	3:37:31 PM	773.0	692.1	758.1	22.60	22.43	22.87	54	49	52	**57**	**51**	**55**	56
6/21/01	3:38:31 PM	788.0	656.2	773.0	22.97	22.55	22.89	54	46	53	**57**	**48**	**56**	57
6/21/01	3:39:31 PM	788.0	641.2	773.0	22.88	22.54	22.95	54	45	53	**57**	**47**	**56**	57
6/21/01	3:40:31 PM	773.0	659.2	791.0	22.92	22.19	22.88	53	47	55	**56**	**49**	**58**	57
6/21/01	3:41:31 PM	788.0	689.1	773.0	22.84	22.25	22.91	55	49	53	**58**	**51**	**56**	57
6/21/01	3:42:31 PM	773.0	641.2	758.1	22.97	22.53	22.97	53	45	52	**56**	**47**	**55**	55
6/21/01	3:43:31 PM	773.0	656.2	755.1	22.72	22.30	22.83	54	46	52	**57**	**49**	**55**	56
6/21/01	3:44:31 PM	773.0	674.2	758.1	22.91	22.29	23.04	53	48	52	**56**	**50**	**55**	55
6/21/01	3:45:31 PM	806.0	725.1	758.1	22.85	22.26	23.05	56	51	52	**59**	**54**	**55**	57
6/21/01	3:46:31 PM	791.0	692.1	773.0	23.11	22.38	23.04	54	49	53	**57**	**51**	**56**	56
6/21/01	3:47:31 PM	773.0	656.2	755.1	23.06	22.34	22.90	53	46	52	**56**	**49**	**55**	55
6/21/01	3:48:31 PM	773.0	641.2	758.1	22.86	22.41	22.86	53	45	52	**56**	**47**	**55**	56
6/21/01	3:49:31 PM	773.0	641.2	758.1	22.70	22.53	23.01	54	45	52	**57**	**47**	**55**	56
6/21/01	3:50:31 PM	773.0	641.2	740.1	22.72	22.41	23.10	54	45	51	**57**	**47**	**53**	55
6/21/01	3:51:31 PM	773.0	689.1	755.1	22.97	22.39	23.18	53	49	51	**56**	**51**	**54**	55
6/21/01	3:52:31 PM	806.0	659.2	758.1	23.22	22.83	23.45	55	46	51	**58**	**48**	**54**	56
6/21/01	3:53:31 PM	791.0	659.2	773.0	23.34	22.97	23.35	54	45	52	**56**	**47**	**55**	56
6/21/01	3:54:31 PM	791.0	659.2	806.0	23.30	22.92	23.41	54	45	54	**57**	**48**	**57**	57
6/21/01	3:55:31 PM	806.0	710.1	758.1	23.35	22.62	23.69	55	50	51	**58**	**52**	**53**	55
6/21/01	3:56:31 PM	773.0	692.1	773.0	23.29	22.91	23.56	52	48	52	**55**	**50**	**55**	55
6/21/01	3:57:31 PM	773.0	626.2	755.1	23.20	22.67	23.44	53	44	51	**55**	**46**	**54**	55
6/21/01	3:58:31 PM	791.0	659.2	791.0	22.73	22.41	23.14	55	46	54	**58**	**49**	**57**	58
6/21/01	3:59:31 PM	806.0	659.2	791.0	22.81	22.66	23.43	56	46	53	**59**	**48**	**56**	58
6/21/01	4:00:31 PM	791.0	659.2	758.1	22.91	22.91	23.53	55	45	51	**58**	**48**	**54**	56
6/21/01	4:01:31 PM	773.0	641.2	758.1	23.16	22.89	23.23	53	44	52	**56**	**46**	**54**	55
6/21/01	4:02:31 PM	791.0	692.1	776.0	23.02	22.75	23.57	54	48	52	**57**	**50**	**55**	56
6/21/01	4:03:31 PM	791.0	659.2	758.1	23.09	22.92	23.53	54	45	51	**57**	**48**	**54**	55
6/21/01	4:04:31 PM	791.0	659.2	776.0	23.02	22.95	23.35	54	45	53	**57**	**47**	**55**	56
6/21/01	4:05:31 PM	791.0	659.2	758.1	22.85	22.78	23.46	55	46	51	**58**	**48**	**54**	56
6/21/01	4:06:31 PM	791.0	644.2	776.0	22.81	22.77	23.39	55	45	52	**58**	**47**	**55**	57

6/21/01	4:07:31 PM	791.0	659.2	776.0	23.09	23.03	23.26	54	45	53	**57**	**47**	**56**	56
6/21/01	4:08:31 PM	773.0	674.2	758.1	23.04	22.62	23.14	53	47	52	**56**	**49**	**54**	55
6/21/01	4:09:31 PM	791.0	692.1	773.0	22.96	22.44	23.00	54	49	53	**57**	**51**	**56**	57
6/21/01	4:10:31 PM	791.0	659.2	758.1	23.07	22.76	23.00	54	46	52	**57**	**48**	**55**	56
6/21/01	4:11:31 PM	806.0	674.2	773.0	23.15	22.70	23.05	55	47	53	**58**	**49**	**56**	57
6/21/01	4:12:31 PM	791.0	659.2	758.1	23.08	22.74	23.18	54	46	52	**57**	**48**	**54**	56
6/21/01	4:13:31 PM	773.0	692.1	773.0	22.94	22.72	23.45	53	48	52	**56**	**50**	**55**	55
6/21/01	4:14:31 PM	791.0	641.2	758.1	22.81	22.84	23.39	55	44	51	**58**	**46**	**54**	56
6/21/01	4:15:31 PM	776.0	692.1	758.1	22.70	22.42	23.14	54	49	52	**57**	**51**	**54**	56
6/21/01	4:16:31 PM	809.0	677.2	758.1	22.86	22.79	23.34	56	47	51	**59**	**49**	**54**	57
6/21/01	4:17:31 PM	773.0	659.2	758.1	23.02	22.77	23.05	53	46	52	**56**	**48**	**55**	55
6/21/01	4:18:31 PM	791.0	644.2	776.0	23.20	22.75	23.33	54	45	53	**57**	**47**	**55**	56
6/21/01	4:19:31 PM	758.1	677.2	758.1	23.10	22.58	23.30	52	47	51	**55**	**50**	**54**	54
6/21/01	4:20:31 PM	809.0	692.1	791.0	23.26	22.99	23.64	55	48	53	**58**	**50**	**56**	57
6/21/01	4:21:31 PM	791.0	692.1	758.1	22.96	22.82	23.47	54	48	51	**57**	**50**	**54**	56
6/21/01	4:22:31 PM	791.0	659.2	758.1	23.20	22.96	23.37	54	45	51	**57**	**47**	**54**	55
6/21/01	4:23:31 PM	791.0	659.2	740.1	23.00	22.79	23.31	54	46	50	**57**	**48**	**53**	55
6/21/01	4:24:31 PM	791.0	674.2	773.0	22.93	22.97	23.24	55	46	53	**58**	**49**	**55**	56
6/21/01	4:25:31 PM	758.1	644.2	758.1	23.09	22.88	23.32	52	45	51	**55**	**46**	**54**	54
6/21/01	4:26:31 PM	776.0	659.2	791.0	23.09	22.81	23.26	53	46	54	**56**	**48**	**57**	56
6/21/01	4:27:31 PM	791.0	674.2	773.0	23.33	23.06	23.54	54	46	52	**56**	**48**	**55**	56
6/21/01	4:28:31 PM	773.0	641.2	758.1	23.18	23.01	23.31	53	44	51	**56**	**46**	**54**	55
6/21/01	4:29:31 PM	809.0	710.1	776.0	23.23	22.61	23.23	55	50	53	**58**	**52**	**56**	57
6/21/01	4:30:31 PM	824.0	707.1	791.0	23.06	22.78	23.40	56	49	53	**60**	**51**	**56**	58
6/21/01	4:31:31 PM	824.0	677.2	791.0	23.06	22.92	23.51	56	47	53	**60**	**49**	**56**	58
6/21/01	4:32:31 PM	791.0	659.2	758.1	23.03	22.85	23.20	54	46	52	**57**	**48**	**54**	56
6/21/01	4:33:31 PM	776.0	644.2	776.0	23.20	22.68	23.13	53	45	53	**56**	**47**	**56**	56
6/21/01	4:34:31 PM	776.0	659.2	758.1	23.21	22.59	23.29	53	46	51	**56**	**48**	**54**	55
6/21/01	4:35:31 PM	791.0	659.2	791.0	23.16	22.71	23.16	54	46	54	**57**	**48**	**57**	57
6/21/01	4:36:31 PM	791.0	659.2	758.1	23.11	22.83	23.24	54	46	52	**57**	**48**	**54**	56
6/21/01	4:37:31 PM	773.0	641.2	758.1	23.20	23.10	23.41	53	44	51	**55**	**46**	**54**	55
6/21/01	4:38:31 PM	791.0	659.2	758.1	23.22	22.91	23.66	54	45	51	**57**	**48**	**53**	55
6/21/01	4:39:31 PM	758.1	659.2	758.1	23.12	22.71	23.36	52	46	51	**55**	**48**	**54**	54
6/21/01	4:40:31 PM	776.0	644.2	758.1	23.02	22.54	23.31	53	45	51	**56**	**47**	**54**	55
6/21/01	4:41:31 PM	806.0	659.2	773.0	22.96	22.54	23.31	55	46	52	**59**	**48**	**55**	57
6/21/01	4:42:31 PM	809.0	710.1	776.0	23.16	22.81	23.09	55	49	53	**58**	**52**	**56**	57
6/21/01	4:43:31 PM	791.0	659.2	776.0	23.23	22.78	23.46	54	46	52	**57**	**48**	**55**	56
6/21/01	4:44:31 PM	791.0	659.2	758.1	23.17	22.85	23.44	54	46	51	**57**	**48**	**54**	55
6/21/01	4:45:31 PM	791.0	659.2	758.1	23.53	22.88	23.77	53	46	50	**56**	**48**	**53**	54
6/21/01	4:46:31 PM	773.0	659.2	740.1	23.27	22.75	23.47	53	46	50	**55**	**48**	**52**	54
6/21/01	4:47:31 PM	773.0	659.2	740.1	23.06	22.99	23.41	53	45	50	**56**	**47**	**52**	54

6/21/01	4:48:31 PM	791.0	674.2	773.0	22.79	23.08	23.41	55	46	52	**58**	**48**	**55**	56
6/21/01	4:49:31 PM	773.0	641.2	758.1	22.89	23.06	23.30	53	44	51	**56**	**46**	**54**	55
6/21/01	4:50:31 PM	791.0	677.2	776.0	22.94	23.00	23.35	55	47	53	**58**	**49**	**55**	56
6/21/01	4:51:31 PM	776.0	659.2	758.1	22.95	22.78	23.36	53	46	51	**56**	**48**	**54**	55
6/21/01	4:52:31 PM	791.0	659.2	758.1	23.01	22.87	23.42	54	46	51	**57**	**48**	**54**	56
6/21/01	4:53:31 PM	791.0	659.2	758.1	23.08	22.87	23.25	54	46	52	**57**	**48**	**54**	56
6/21/01	4:54:31 PM	776.0	692.1	758.1	23.40	22.78	22.96	52	48	52	**55**	**50**	**55**	55
6/21/01	4:55:31 PM	773.0	641.2	758.1	23.09	22.85	23.39	53	44	51	**56**	**46**	**54**	55
6/21/01	4:56:31 PM	773.0	659.2	758.1	22.95	22.98	23.32	53	45	51	**56**	**47**	**54**	55
6/21/01	4:57:31 PM	773.0	659.2	758.1	22.96	22.79	23.16	53	46	52	**56**	**48**	**54**	55
6/21/01	4:58:31 PM	791.0	740.1	773.0	22.99	22.44	23.37	54	52	52	**57**	**55**	**55**	56
6/21/01	4:59:31 PM	776.0	644.2	758.1	22.83	22.73	23.48	54	45	51	**57**	**47**	**54**	55

Appendix II

Field testing Data 8/23/01 – 8/31/01

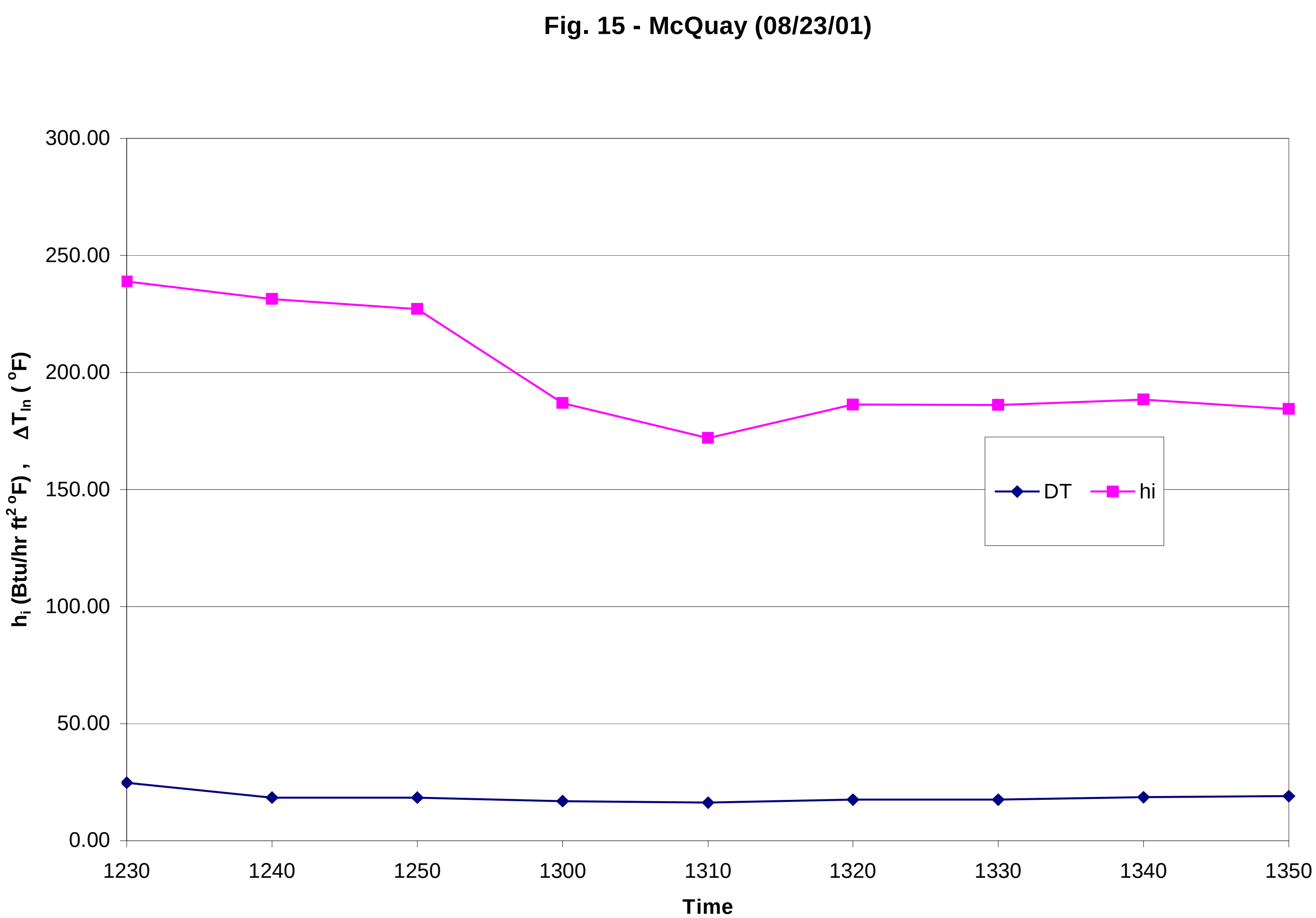
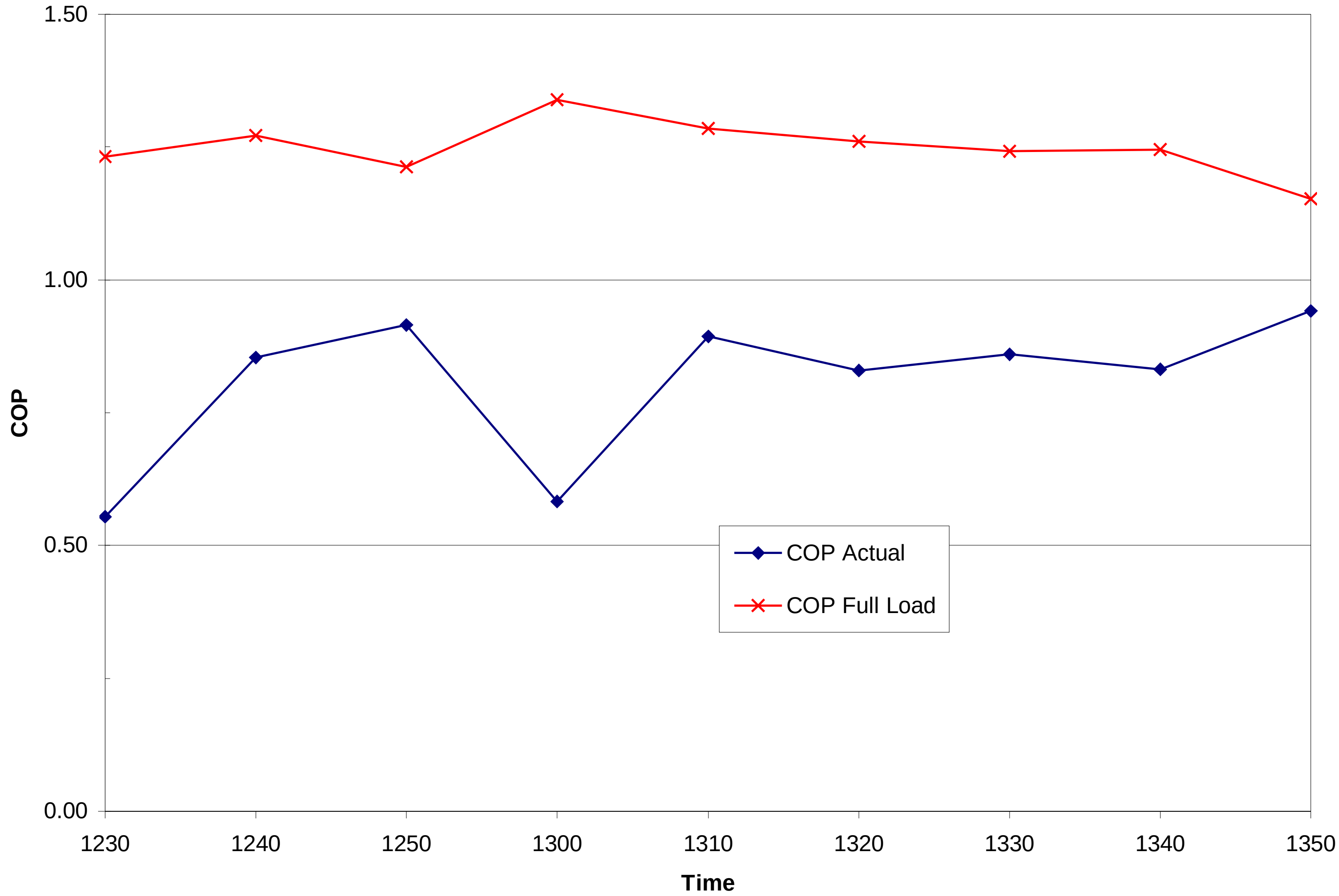


Fig. 15 - McQuay (08/23/01)
300.00
250.00
200.00
150.00
100.00
50.00
0.00
hi (Btu/hr ft2 oF) , ΔTln (oF)
DT
hi
1230
1240
1250
1300
1310
1320
1330
1340
1350
Time

Fig. 16 - McQuay (8/23/01)



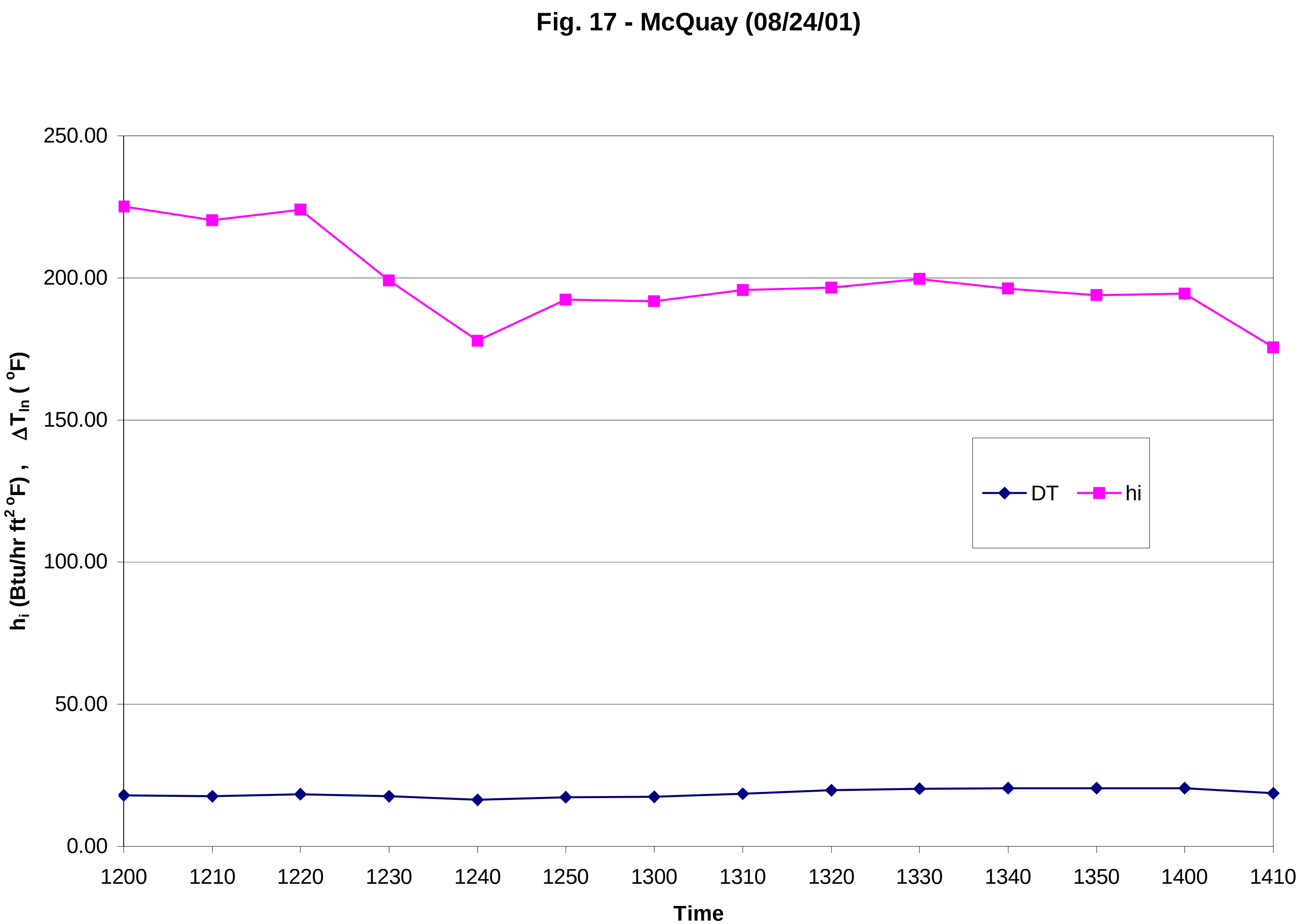


Fig. 17 - McQuay (08/24/01)
h_i (Btu/hr ft^2 °F) , ΔT_{ln} (°F)
250.00
200.00
150.00
100.00
50.00
0.00
DT
hi
1200
1210
1220
1230
1240
1250
1300
1310
1320
1330
1340
1350
1400
1410
Time

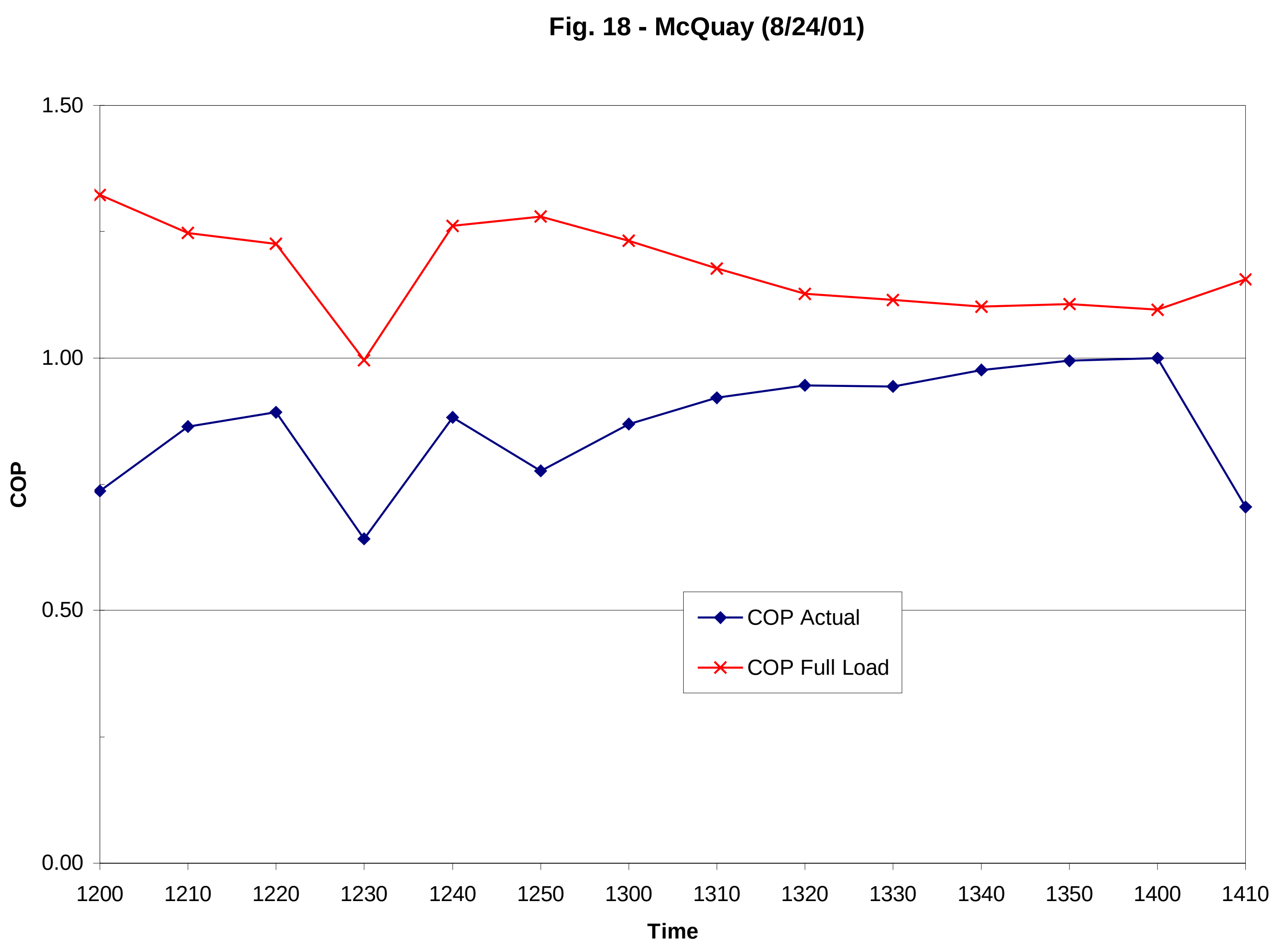


Fig. 18 - McQuay (8/24/01)
1.50
1.00
0.50
0.00
COP
1200
1210
1220
1230
1240
1250
1300
1310
1320
1330
1340
1350
1400
1410
Time
COP Actual
COP Full Load

Fig. 19 - McQuay (08/25/01)

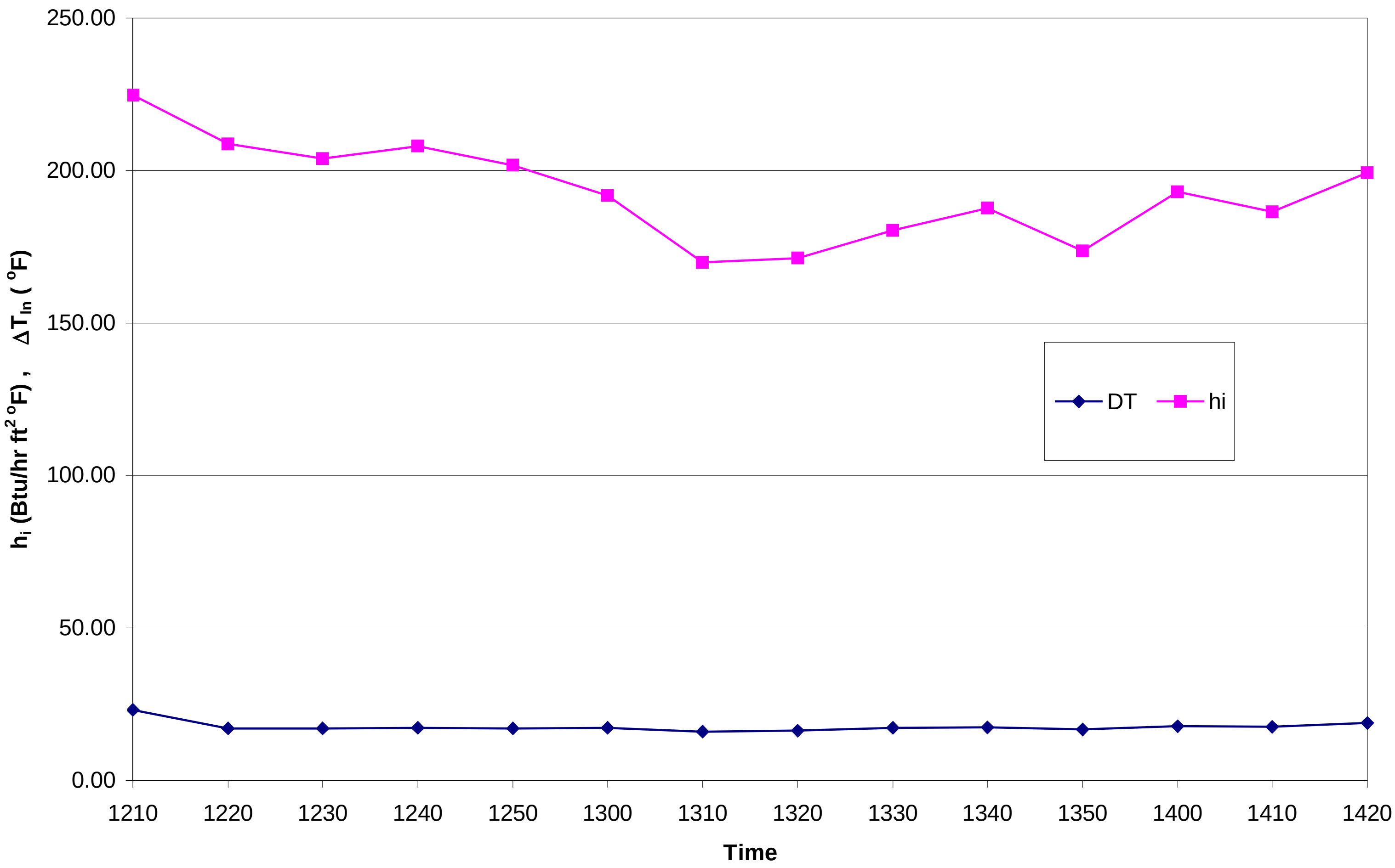
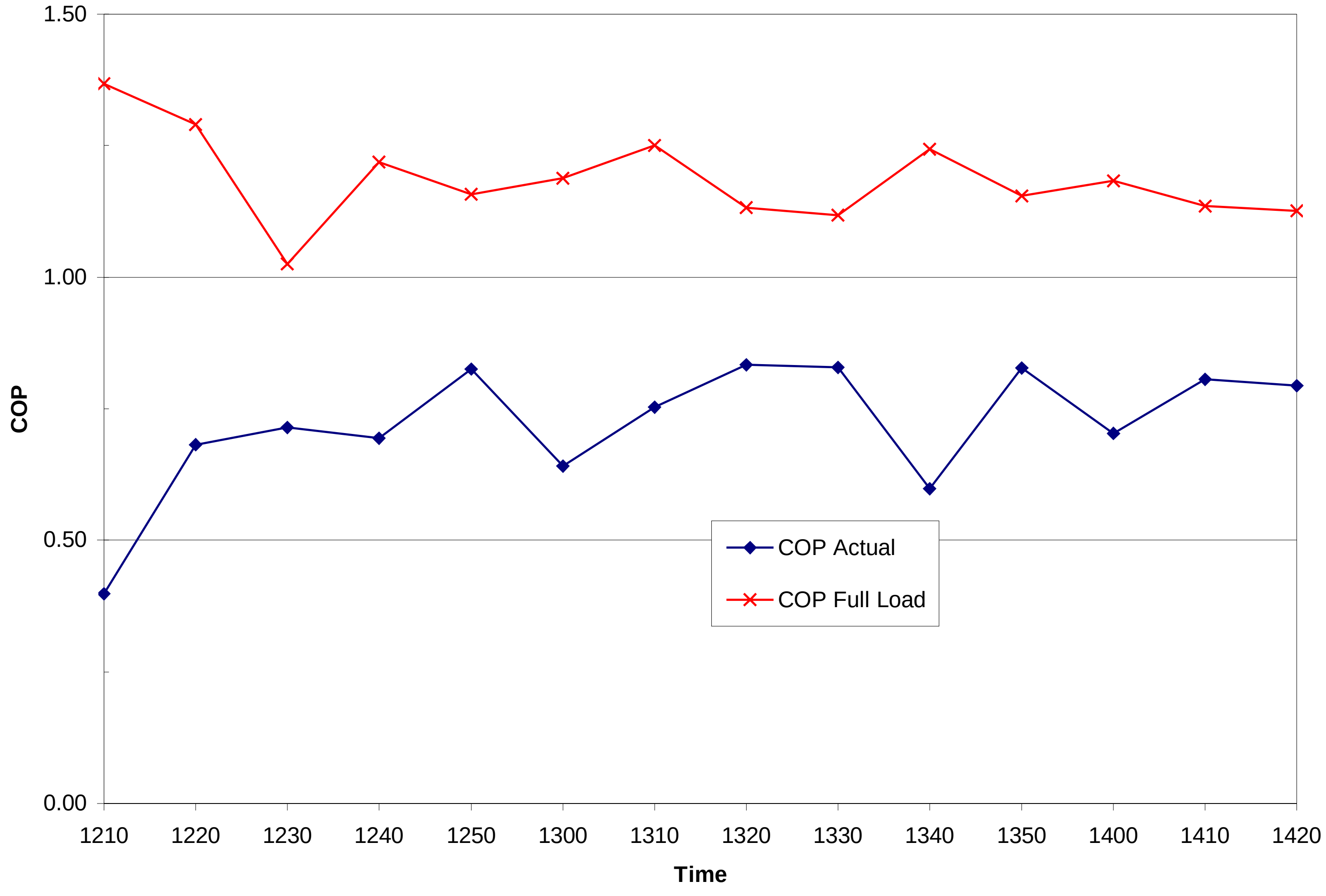


Fig. 20 - McQuay (8/25/01)



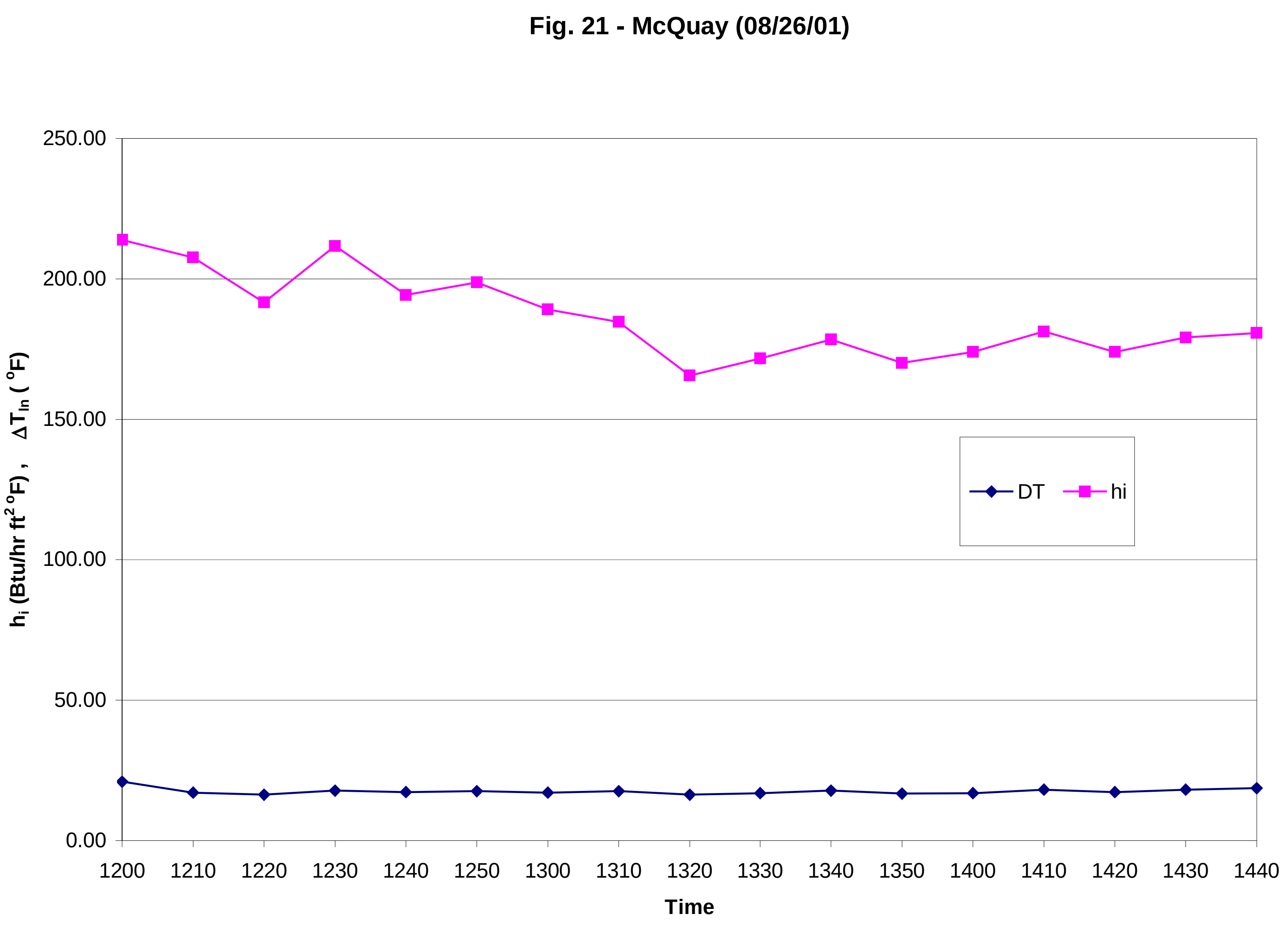


Fig. 21 - McQuay (08/26/01)
250.00
200.00
150.00
100.00
50.00
0.00
h_i (Btu/hr ft^2 °F) , ΔT_{ln} (°F)
DT
hi
1200
1210
1220
1230
1240
1250
1300
1310
1320
1330
1340
1350
1400
1410
1420
1430
1440
Time

Fig. 22 - McQuay (8/26/01)

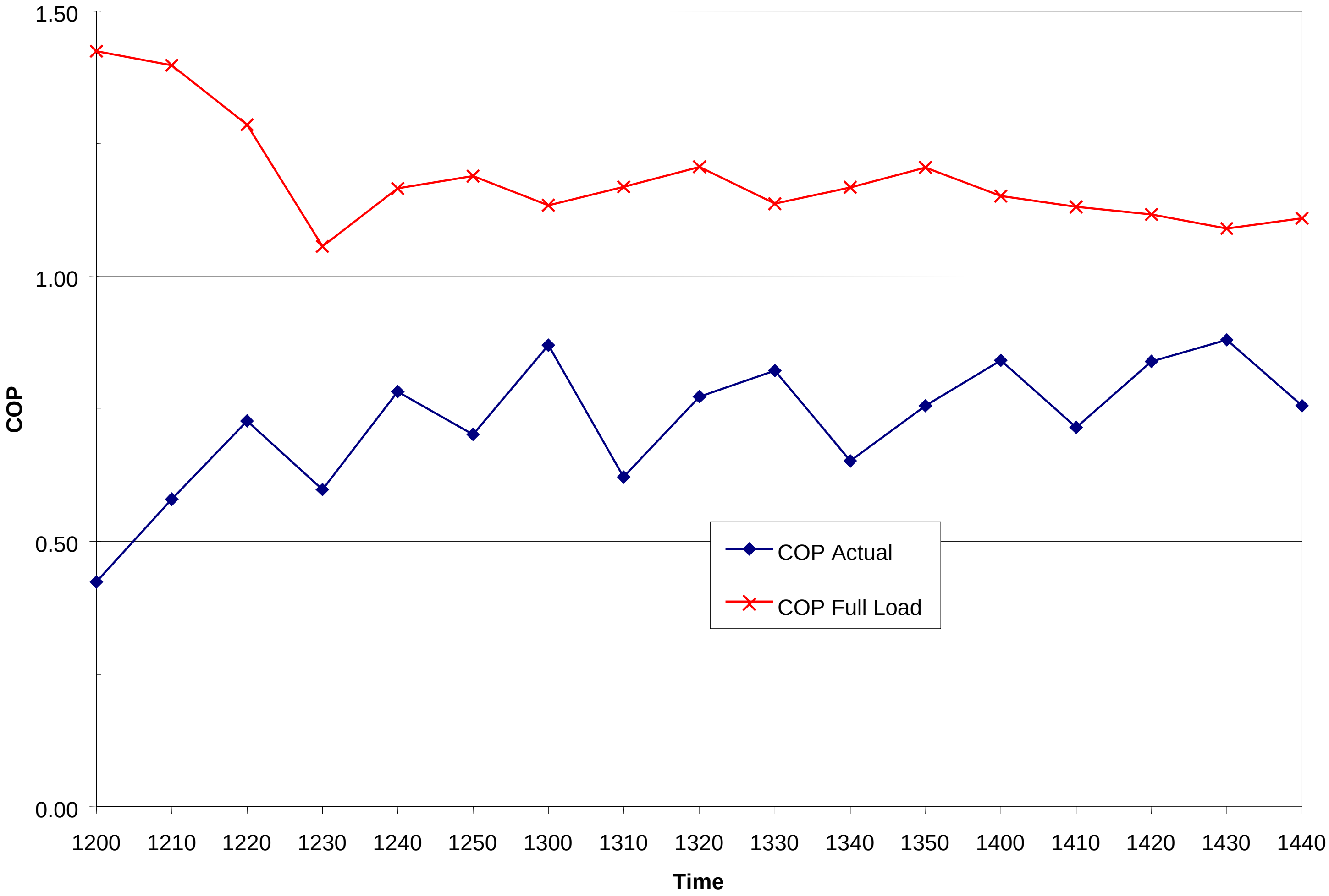


Fig. 23 - McQuay (08/27/01)

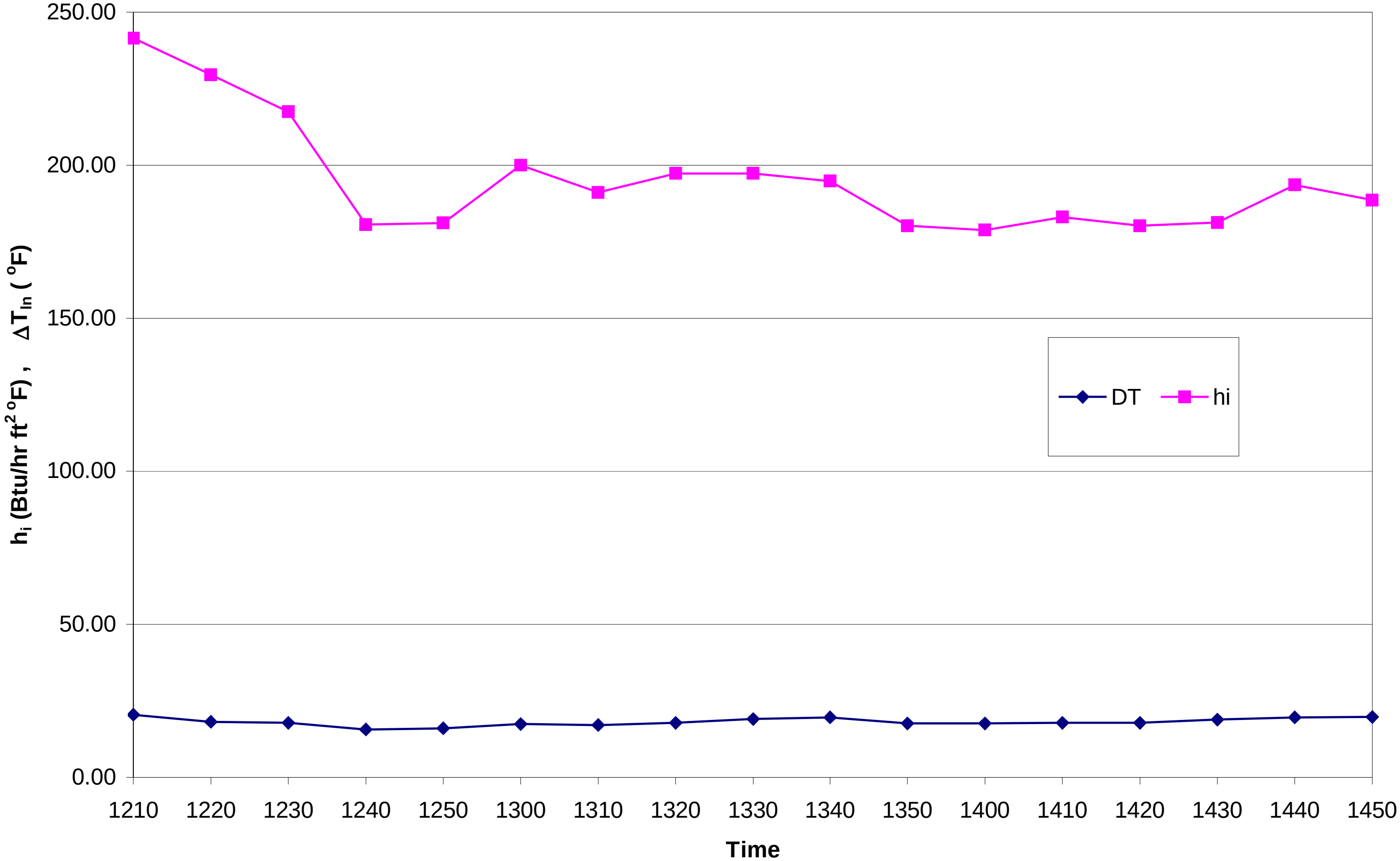


Fig. 24 - McQuay (8/27/01)

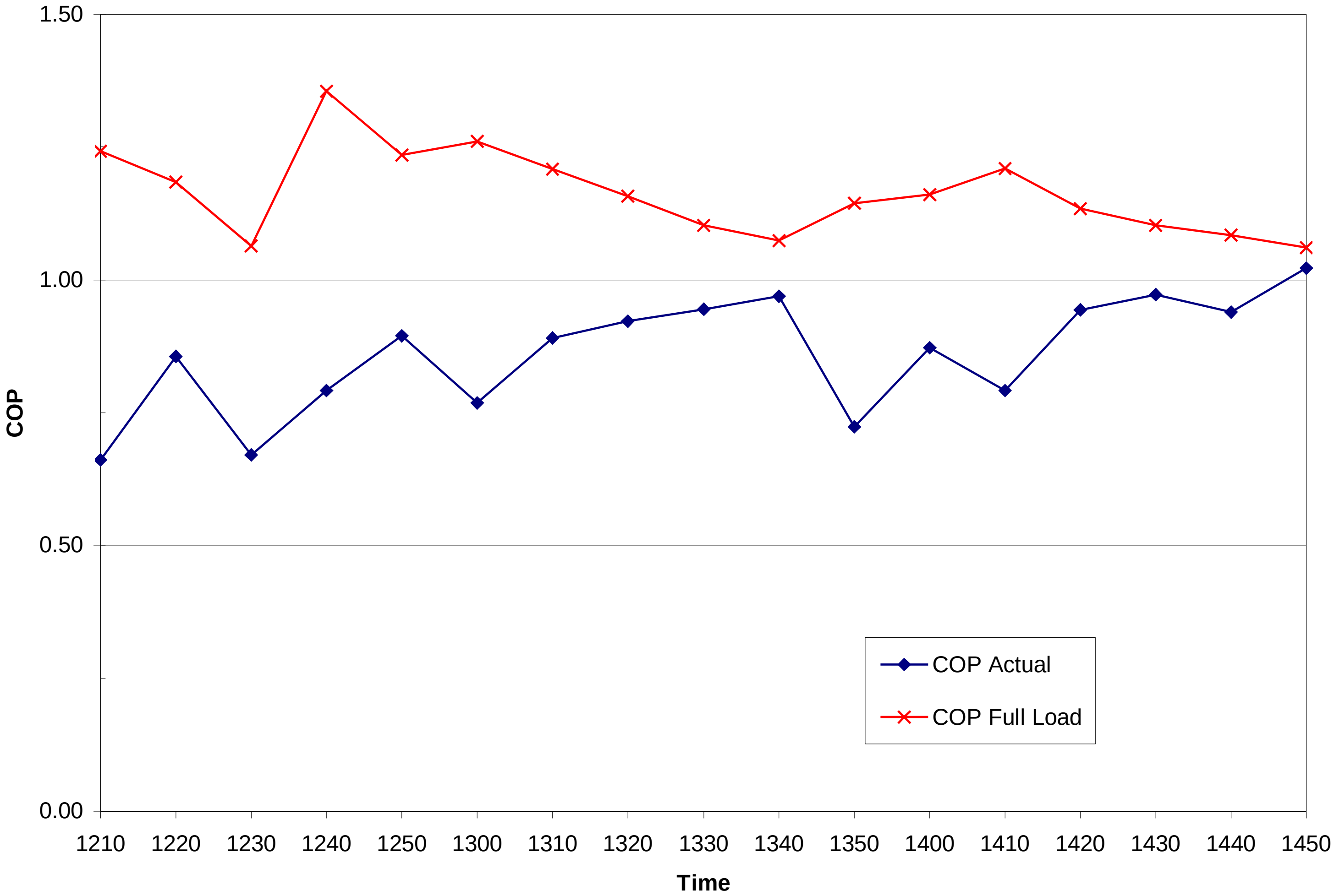


Fig. 25 - McQuay (08/28/01)

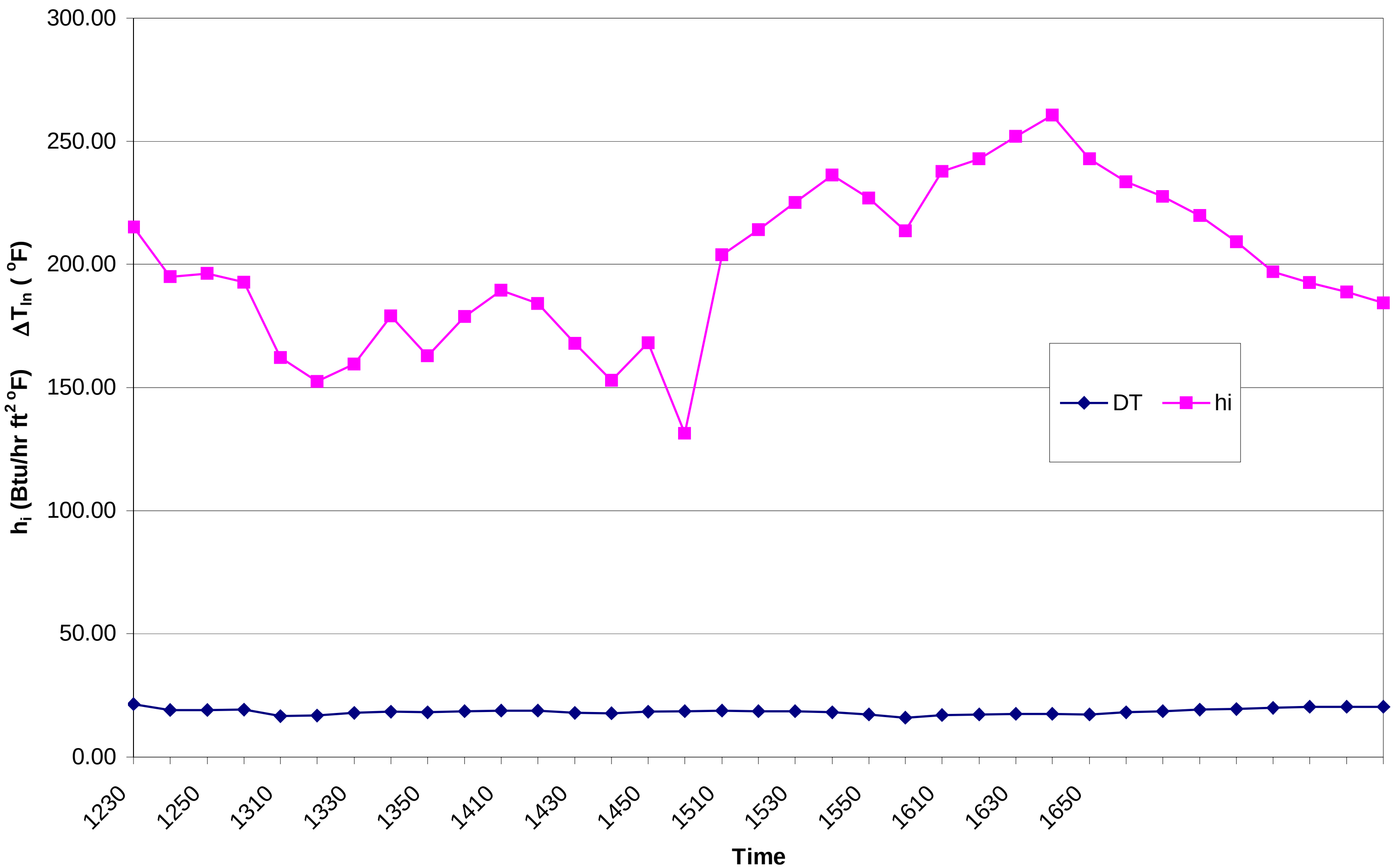


Fig. 26 - McQuay (8/28/01)

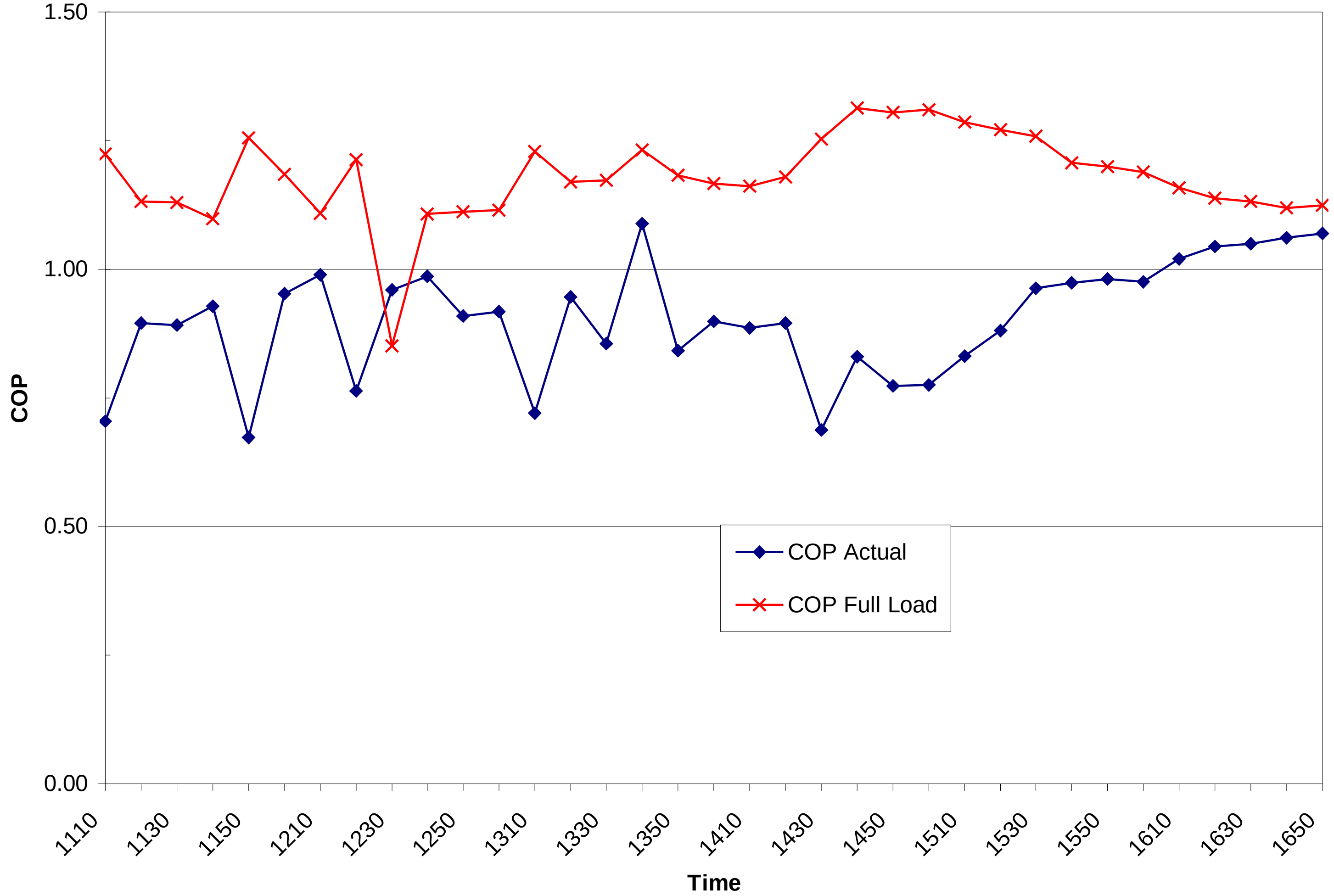


Fig. 27 - McQuay (08/29/01)

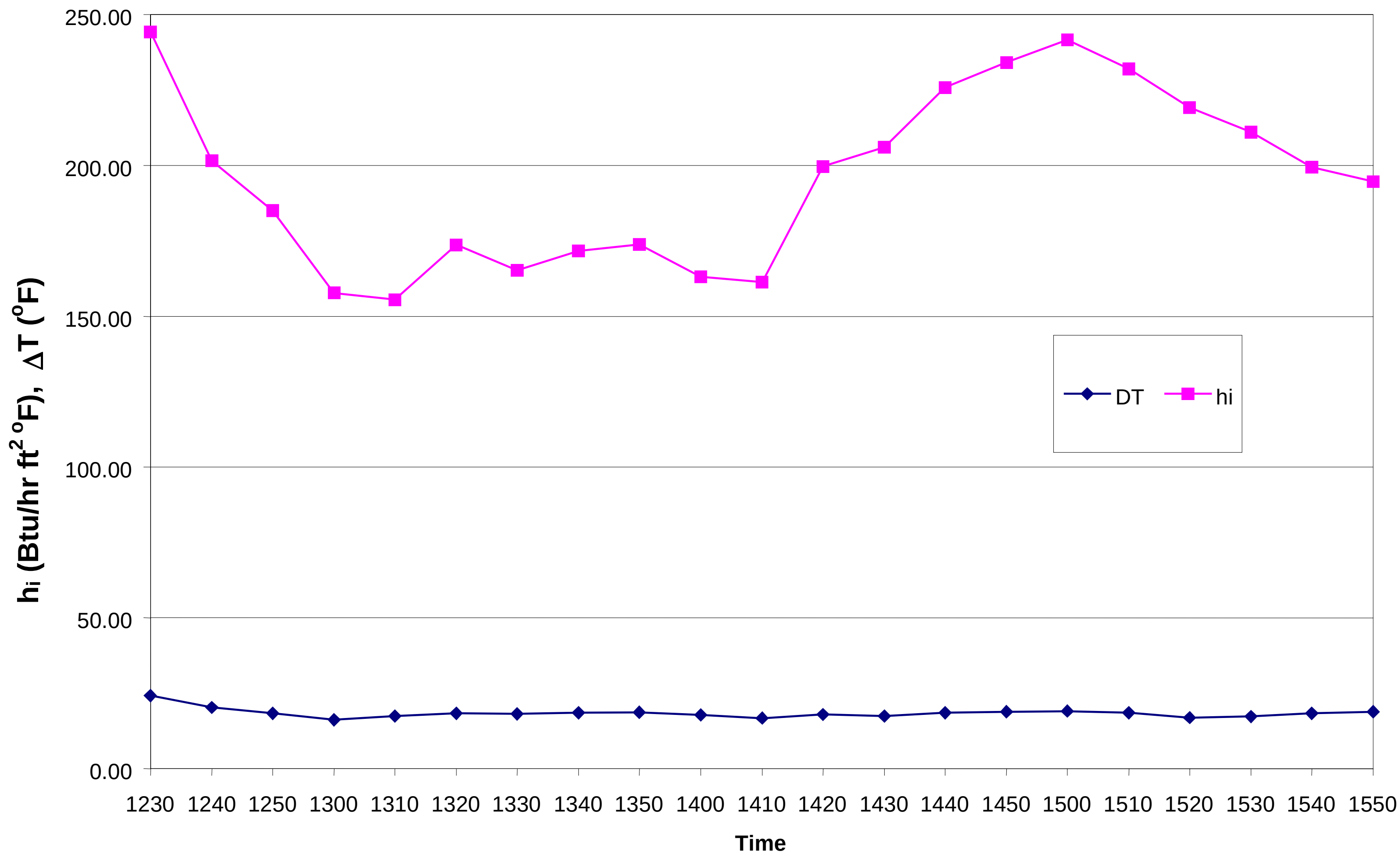


Fig. 28 - McQuay (8/29/01)

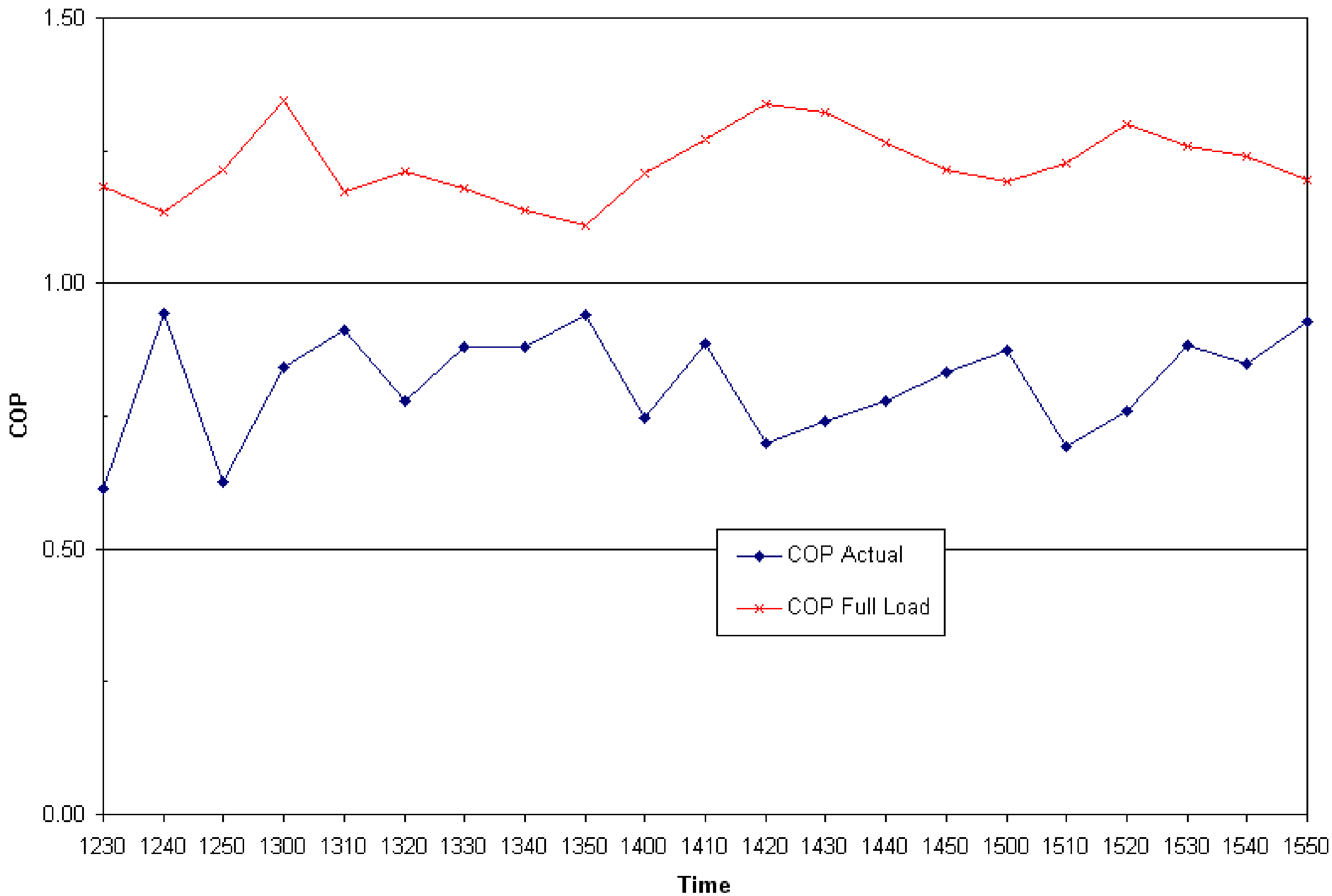
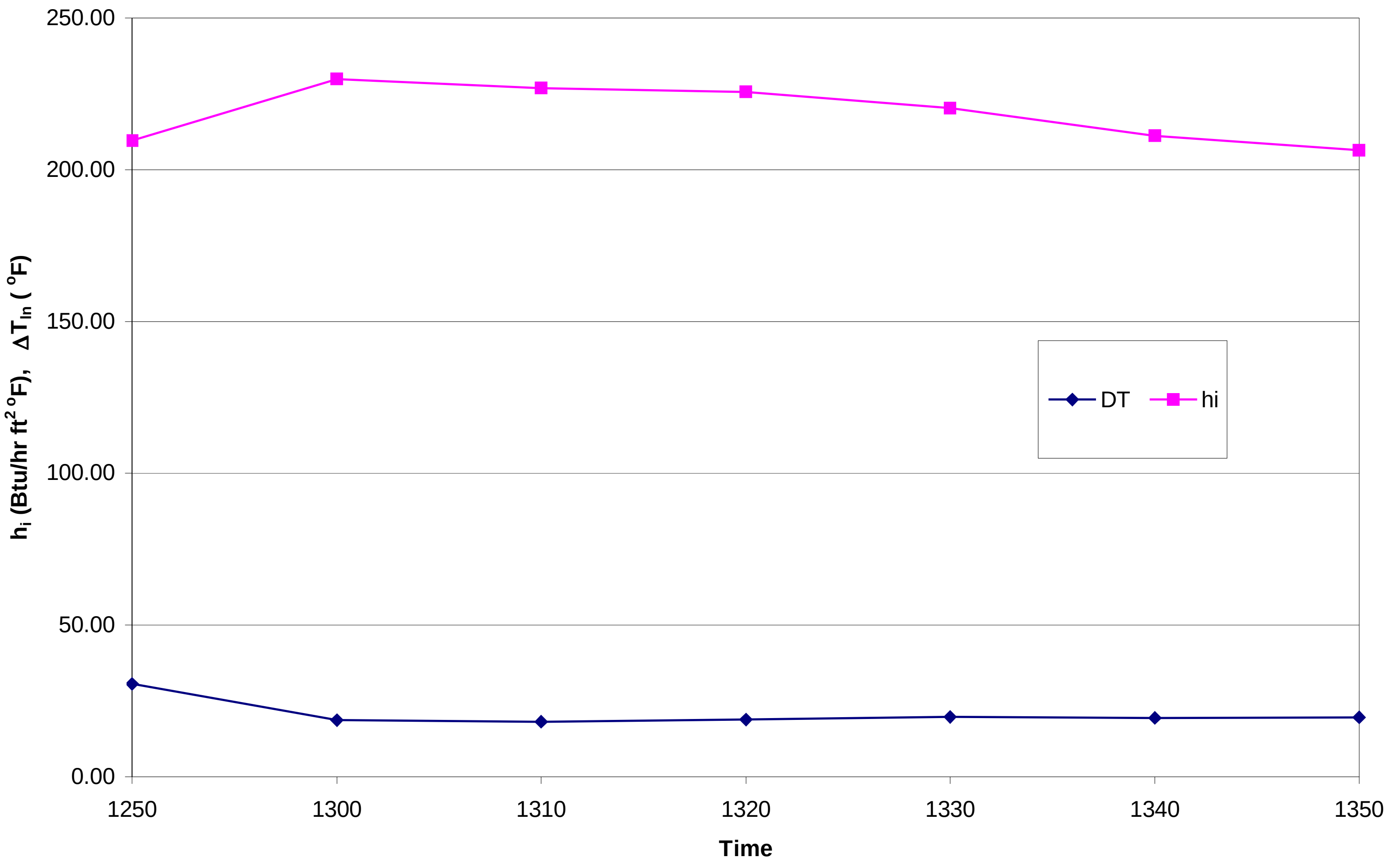


Fig. 29 - McQuay (08/30/01)



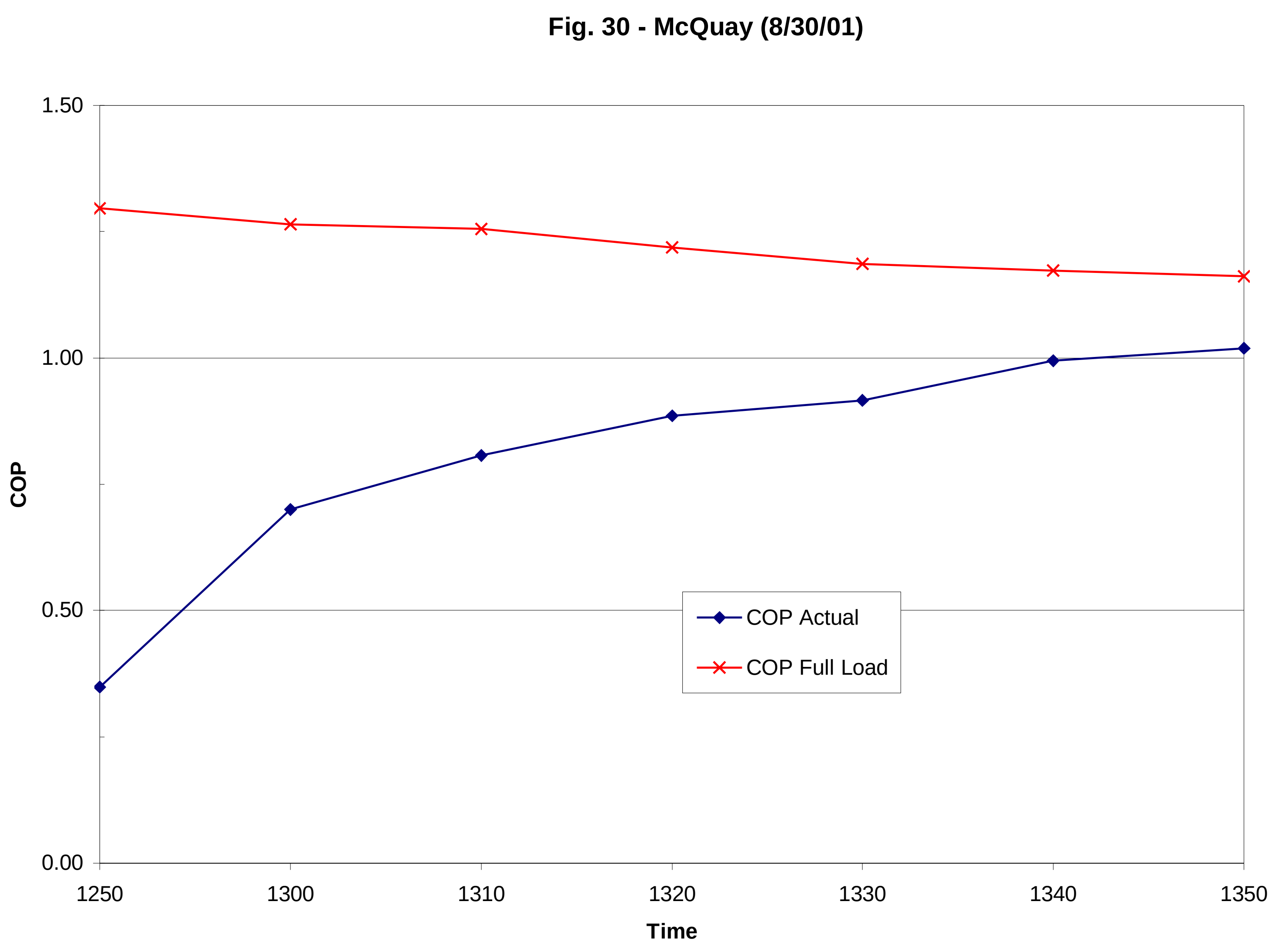


Fig. 30 - McQuay (8/30/01)
1.50
1.00
0.50
0.00
COP
1250
1300
1310
1320
1330
1340
1350
Time
COP Actual
COP Full Load

Fig. 31 - McQuay (08/31/01)

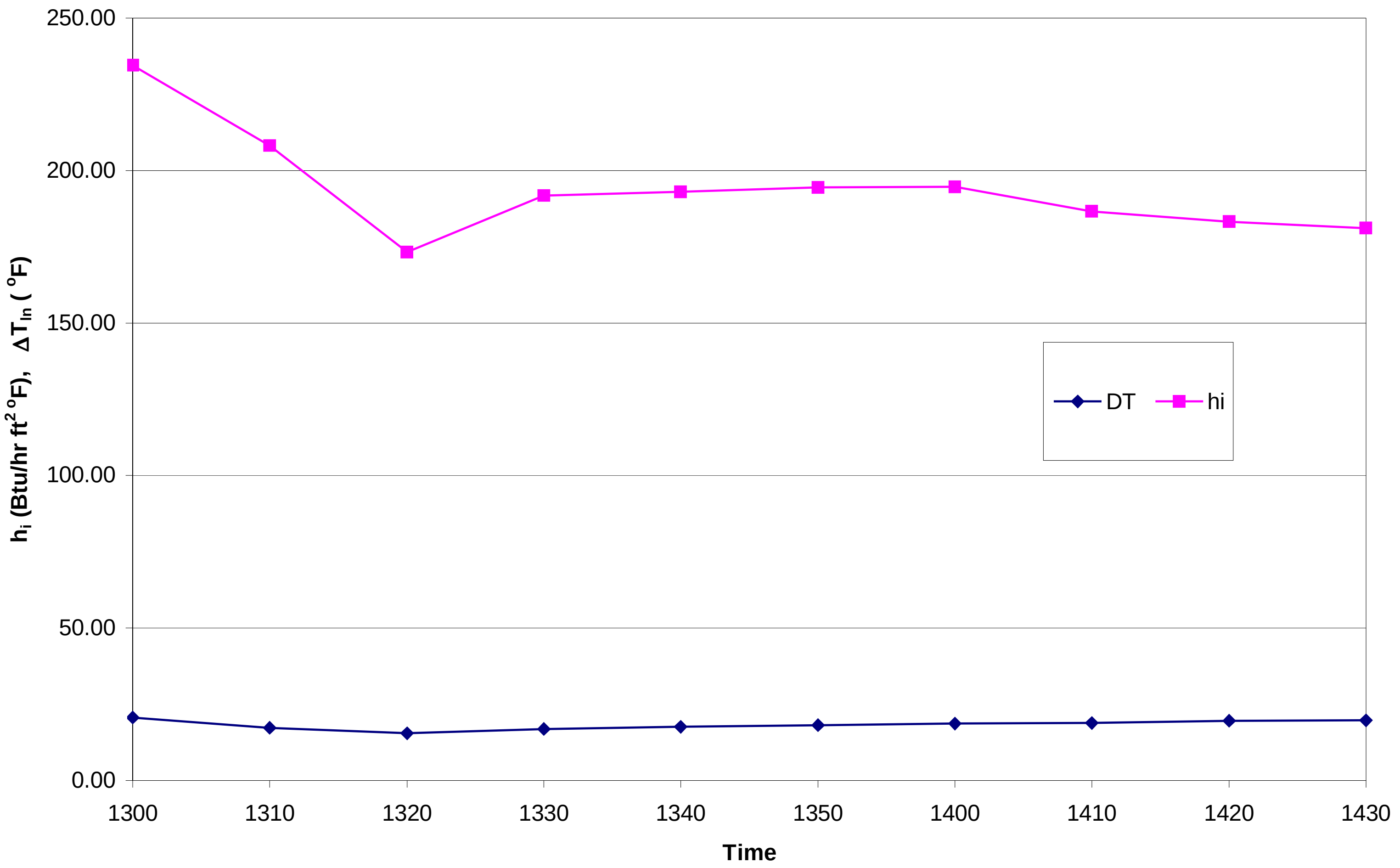


Fig. 32 - McQuay (8/31/01)