PROOF-OF-CONCEPT OF A DUAL-FIRED (SOLAR & NATURAL GAS) GENERATOR FOR USE IN A SPACE-COOLING SYSTEM FOR RESIDENTIAL AND LIGHT COMMERCIAL BUILDINGS (3-15RT

FEASIBILITY ANALYSIS AND FINAL EISG REPORT

Prepared For:
California Energy Commission
Energy Innovations Small Grant Program

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ENERGY INNOVATIONS SMALL GRANT (EISG) PROGRAM

FEASIBILITY ANALYSIS REPORT (FAR)

PROOF-OF-CONCEPT OF A DUAL-FIRED (SOLAR & NATURAL GAS) GENERATOR FOR USE IN A SPACE COOLING SYSTEM FOR RESIDENTIAL AND LIGHT COMMERCIAL BUILDINGS (3-15RT)

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LEGAL NOTICE

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 of which $2.4 million/year is allocated to the Energy Innovation Small Grant (EISG) Program for grants. The EISG Program is administered by the San Diego State University Foundation under contract to the California State University, which is under contract to the Commission.

The EISG Program conducts four solicitations a year and awards grants up to $75,000 for promising proof-of-concept energy research.

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

- Residential and Commercial Building End-Use Energy Efficiency
- Industrial/Agricultural/Water End-Use Energy Efficiency
- Renewable Energy Technologies
- Environmentally-Preferred Advanced Generation
- Energy-Related Environmental Research
- Energy Systems Integration

The EISG Program Administrator is required by contract to generate and deliver to the Commission a Feasibility Analysis Report (FAR) on all completed grant projects. The purpose of the FAR is to provide a concise summary and independent assessment of the grant project using the Stages and Gates methodology in order to provide the Commission and the general public with information that would assist in making follow-on funding decisions (as presented in the Independent Assessment section).

The FAR is organized into the following sections:

- Executive Summary
- Stages and Gates Methodology
- Independent Assessment
- Appendices
  - Appendix A: Final Report (under separate cover)
  - Appendix B: Awardee Rebuttal to Independent Assessment (Awardee option)

For more information on the EISG Program or to download a copy of the FAR, please visit the EISG program page on the Commission's Web site at:
http://www.energy.ca.gov/research/innovations

or contact the EISG Program Administrator at (619) 594-1049 or email eisgp@energy.state.ca.us.

For more information on the overall PIER Program, please visit the Commission's Web site at http://www.energy.ca.gov/research/index.html.

Proof-of-Concept Of A Dual-Fired (Solar & Natural Gas) Generator For Use In A Space Cooling System For Residential And Light Commercial Buildings (3-15RT)

EISG Grant # 01-06

Awardee:	Cooling Technologies, Inc.
Principal Investigator:	Michael Garrabrant
PI Contact Info:	(419) 536-9006; mag@coolingtechnologies.com
Grant Funding:	$75,000
Grant Term:	February 2002—April 2003

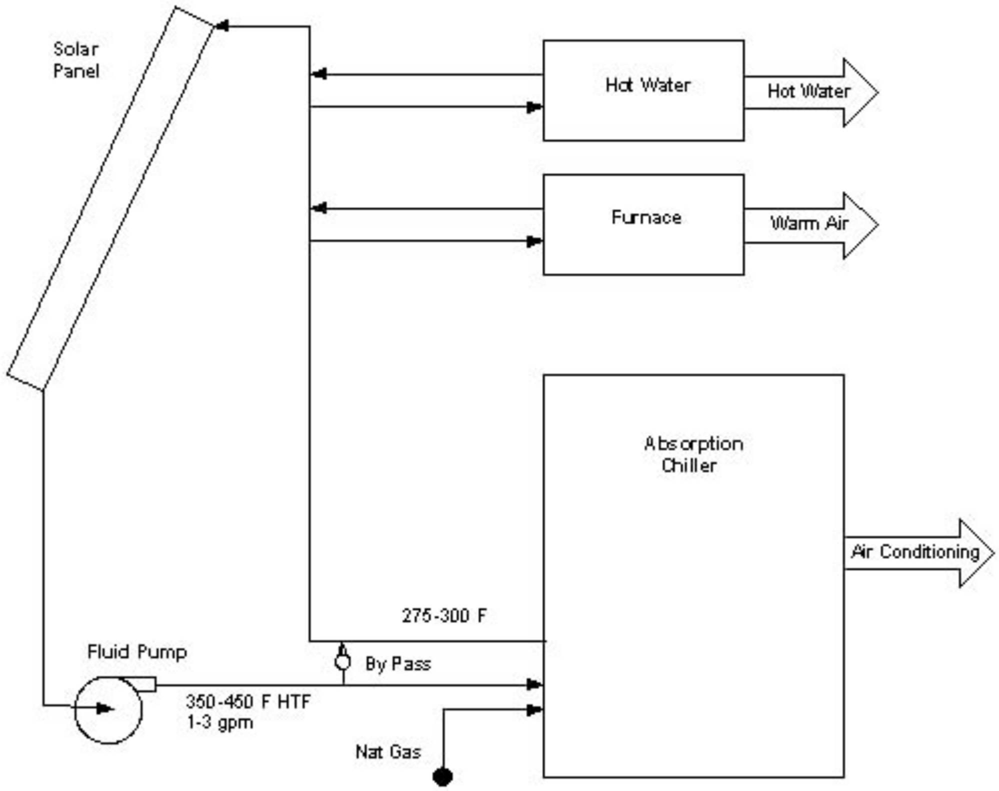
Introduction

Many residential and light commercial buildings utilize vapor compression air conditioning technology for space cooling. These devices require significant power (~1.25 kW/ton depending on size and age of the air conditioning unit) and are large contributors to residential and commercial peak electric demands. According to DOE data, residential buildings in the U.S. consume 1.62 quads for space cooling and virtually all of this energy is supplied by grid-connected electrical equipment. Most metropolitan areas of the U.S. are experiencing significant electric demand and supply imbalance during the summer months due to the increasing use of air conditioning equipment. Solar energy cooling systems would avoid significant amounts of electric energy normally required for space cooling.

Benefits to the California public resulting from the implementation of the proposed dual-fuel solar/gas cooling system include, but are not limited to –

- A potential 75% reduction in electric demand for residential and small commercial space cooling.
- A significant reduction of fossil fuel-fired electric generation required for HVAC electric loads.
- Small air conditioning application integration with combined heat and power systems.

Cooling Technologies, Inc. has developed, adapted and tested a new dual-fired (gas-solar) generator for their 5-ton ammonia-water (NH_3-H_2O) air-cooled absorption chiller. The generator is a simple adaptation of their standard gas-fired chiller generator, which minimizes the extra cost associated with the solar/gas product. With dual-fire capability, the need for thermal storage or back-up systems is eliminated. Two prototype generators were developed and tested as part of a breadboard chiller.



Solar HVAC System w/Dual-Fired Absorption Chiller

The system differs from previous solar-powered cooling systems in many respects:

1. Small Size: the system targets the residential, multi-family, and light commercial markets (3-25 Refrigeration Tons[1]).
2. Air-Cooled: the cooling system is air-cooled and therefore does not require expensive and difficult to maintain cooling towers.
3. Dual-Fired: the cooling system is able to operate on either or both solar energy and natural gas or propane, an important feature for practical, affordable systems.

Objectives

The goal of this project was to prove the feasibility of a dual-fuel (solar/gas) generator for a standard 5-ton NH_3-H_2O air-cooled absorption chiller. The specific research goals were –

1. Design, fabricate and test a dual-fired (solar/gas) generator for a 5-ton ammonia-water absorption chiller. The design is a modification to the geometry of an existing Cooltec generator in order to minimize product cost.
2. Demonstrate that the heat exchanger is capable of heating the working fluid (ammonia-water) up to an operating temperature of 375 °F, given a solar source temperature of 450 °F.

[1] The word "ton" used in this report is understood to mean Refrigeration Tons.

3. Demonstrate that the dual-fired generator, when coupled to the other components of a 5-ton absorption chiller, is capable of producing 5-ton of cooling at the standard ARI conditions (95 °F ambient, 55 – 45 °F chilled water).
4. Operate the breadboard chiller while firing the generator on both simulated solar energy and the combustion of natural gas simultaneously in order to provide guidance for the packaged prototype control system logic specification.

Outcomes

1. The generator, as tested, is a modification of Cooltec's standard gas-fired generator used as part of their 5-ton chiller by the addition of a heat transfer surface to the outside shell of the generator. The model was verified and calibrated based on the experimental data and can be used to fine tune the generator design for the next development stage.
2. The dual-fired generator prototypes, when fired by hydronic fluid, were able to heat the generator bottom temperature up to the target operating temperature with hydronic inlet temperatures of 460 °F and above. Generator temperatures were within 12% of the target at nominal 400 °F hydronic temperatures. The prototypes retained their ability to reach full operating temperatures when gas-fired.
3. At 95 °F ambient conditions and 55 °F return chilled water temperature, the breadboard chiller achieved full cooling capacity with hydronic temperatures greater than 470 °F. The measured capacity reduction was 6 and 20% with hydronic inlet temperatures on the order of 450 °F and 400 °F respectively.
4. The prototype generator was fired using both simulated solar heat and the combustion of natural gas at the same time.

Conclusions

1. The project team successfully designed, fabricated and tested two dual- fired (solar/gas) absorption chiller generators.
2. The dual-fired generator temperatures were 12% below the target 400 °F when "fired" with 460 °F hydronic fluid inlet temperature.
3. The breadboard absorption chiller with the dual-fired generator did not achieve full cooling capacity at ARI standard conditions.
4. Simultaneous dual-firing of the generator with both simulated solar and gas fuel was accomplished and data was collected. The recorded behavior of the cycle under these conditions provides information required to design the control system for future packaged prototypes.

Benefits to California

The research project resulted in successful design, fabrication and testing of a solar-fired absorption chiller generator. Since the research and development are based on modifying a commercially available gas-fired 5-ton absorption chiller, the project would likely result in a commercial solar absorption air conditioning unit for large residential, multi-family and small commercial customers. The estimated incremental production cost for modification of Cooltec's current absorption chiller is $150 to $225 per unit.

The electric energy savings are substantial, reducing space cooling electric load by as much as 75%. Since space cooling is a significant component of electric energy usage in California and as much as 30% of peak demand on a hot, sunny California afternoon, the impact of this system would be great.

Recommendations

This project has resulted in proven feasibility of the technology. However, additional testing and refinement of the dual-fired generator design are necessary to improve its performance with the absorption cycle. Also a significant amount of development work is necessary for co-firing of the generator with solar and gas simultaneously, particularly with respect to generator burner controls and absorption cycle optimization.

Next step development work should focus on generator performance improvement and co-firing operation. Testing should continue on breadboard with simulated time varying (diurnal) solar energy delivered and cooling load cycling to fine tune absorption cycle and controls.

Following successful performance and co-firing operation, the system could then be field tested, in limited non-critical applications to capture real world performance.

Independent Assessment

For the research under evaluation, the Program Administrator assessed the level of development for each activity tracked by the Stages and Gates methodology. This assessment is summarized in the Development Assessment Matrix below. Shaded bars are used to represent the assessed level of development for each activity as related to the development stages. Our assessment is based entirely on the information provided in the course of this project, and the final report. Hence it is only accurate to the extent that all current and past work related to the development activities are reported.

Development Assessment Matrix

Stages / Activity	1 Idea Generation	2 Technical & Market Analysis	3 Research	4 Technology Develop-ment	5 Product Develop-ment	6 Demon-stration	7 Market Transfor-mation	8 Commer-cialization
Marketing								
Engineering / Technical								
Legal/ Contractual								
Risk Assess/ Quality Plans								
Strategic								
Production. Readiness/								
Public Benefits/ Cost								

The Program Administrator's assessment was based on the following supporting details:

Marketing/Connection to the Market

The targeted markets are large residential, multi-family, and small commercial space cooling applications in California, nationwide and international. The researchers should identify potential energy savings for typical class customers for each California weather zone.

Engineering/Technical

This project proved the feasibility of a dual-fired (solar and gas) generator for a 5-ton absorption chiller. Additional testing is needed to improve generator performance with solar energy and optimize the operating parameters for co-firing of the generator. The Program Administrator recommends that further generator design development takes place to improve performance and co-firing capability.

Legal/Contractual

Cooltec's product development began with research performed by the Ohio State University and Battelle Laboratories. The project was funded with assistance from the Gas Research Institute (GRI), the Department of Energy (DOE), and a utility consortium headed by Columbia Gas of Ohio. Through technology integration, Cooltec developed proprietary heat exchangers for enhanced performance with support of GRI, Southern California Gas Company (SOCAL), and Keyspan Energy (Brooklyn Union). Primary technology for the Cooltec products is covered by licenses with Ohio State and GRI. These license agreements cover the technology for absorption chillers, heat pumps, and process chillers smaller than 100RT and apply to all products currently planned for development by Cooltec. In addition, provisional patents including application under the Patent Cooperating Treaty have been applied for in two general areas covering heat transfer control devices and dual-fired absorption or heat pump controls.

Environmental, Safety, Risk Assessments/ Quality Plans

Quality Plans include Reliability Analysis, Failure Mode Analysis, Manufacturability, Cost and Maintainability Analyses, Hazard Analysis, Coordinated Test Plan, and Product Safety and Environmental. Cooling Technologies is directly responsible for product development, marketing, distribution, and cooling product warranty. Partnerships with component suppliers and contract manufacturers support the manufacturing process with the Company performing final assembly and inspection. The modifications required to convert a standard Cooltec chiller to the dual-fire configuration use comparable manufacturing techniques. By design, the dual-fire generator heat exchanger can be applied to the standard Cooltec generator with minor modifications and an overall geometry that fits within existing cabinetry.

Strategic

This product has no known critical dependencies on other projects under development by PIER or elsewhere.

Production Readiness/Commercialization

The researchers have successfully developed and tested a dual-fired (solar and gas) absorption chiller generator design. The technology requires some additional performance improvement and co-firing capability development. Once the researchers meet these new objectives, the technology would be ready for field testing and then commercialization.

Public Benefits

Public benefits derived from PIER research and development are assessed within the following context:

- Reduced environmental impacts of the California electricity supply or transmission or distribution systems
- Increased public safety of the California electricity system
- Increased reliability of the California electricity system
- Increased affordability of electricity in California

The primary benefit to the ratepayer from this research is increased affordability of electricity in California, resulting from the reduction of electric demand for residential and small commercial space cooling. The researchers state a potential reduction in demand for air conditioning of 75%, thus diminishing need for new fossil power generation, transmission and distribution facilities in California. The reduction of electric demand will also enhance the reliability of the California electric delivery system, alleviating stress and congestion in important grid areas.

Program Administrator Assessment

After taking into consideration: (a) research findings in the grant project, (b) overall development status as determined by Stages and Gates and (c) relevance of the technology to California and the PIER program, the Program Administrator has determined that the proposed technology should be considered for follow on funding within the PIER program.

Receiving follow on funding ultimately depends upon: (a) availability of funds, (b) submission of a proposal in response to an invitation or solicitation and (c) successful evaluation of the proposal.

Appendix A: Final Report (under separate cover)

Appendix B: Awardee Rebuttal to Independent Assessment (none submitted)

ENERGY INNOVATIONS SMALL GRANT (EISG) PROGRAM

EISG FINAL REPORT

PROOF-OF-CONCEPT OF A DUAL-FIRED (SOLAR & NATURAL GAS) GENERATOR FOR USE IN A SPACE COOLING SYSTEM FOR RESIDENTIAL AND LIGHT COMMERCIAL BUILDINGS (3-15RT)

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Legal Notice

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Inquires related to this final report should be directed to the Awardee (see contact information on cover page) or the EISG Program Administrator at (619) 594-1049 or email eisgp@energy.state.ca.us.

Acknowledgement

Cooling Technologies, Inc. acknowledges and expresses our gratitude to Dr. Tom Henkel from Duke Solar Energy, LLC, and Dr. Rod Mahoney, Sandia National Laboratory, for providing solar panel, market and heat transfer fluid information assistance.

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ABSTRACT

Solar powered cooling systems for the residential and light commercial markets is closer to reality due to recent technology advances in thermal solar panels and NH_3-H_2O air-cooled absorption chillers. A dual-fired (gas-solar) generator for an ammonia-water absorption chiller was developed and successfully tested. The generator design is fully capable of operating on natural gas, solar energy (via a hydronic fluid), or both at the same time. The generator is a simple adaptation of the standard Cooltec5™ gas-fired chiller generator, which will minimize the extra cost associated with the solar-gas product. With dual-fire capability, the need for thermal storage or back-up systems is eliminated. Two prototype generators were tested as part of a breadboard chiller, with the beta version achieving full capacity when provided a hydronic temperature of 470° F. Capacity was reduced by 6% and 20% with hydronic temperatures of 450° F and 400° F respectively. Full capacity was maintained when fired on natural gas. This technology has additional applications in the recovery of waste heat from micro turbines, engines and fuel cells.

Key Words: solar cooling, absorption, GAX cycle, dual-fired generator, indirect fired generator, solar heating, absorption heat pump.

EXECUTIVE SUMMARY

Introduction: Cooling Technologies, Inc. (Cooltec), has developed a high-efficiency, heat activated absorption chiller (Cooltec5^{tm}) utilizing proprietary heat transfer technology, designed for the light commercial and high-end residential air conditioning markets. The unit converts heat energy to cooling more effectively than competitive chillers and is less expensive to operate than conventional cooling products. Heat sources to operate Cooltec products currently include natural gas and propane. Future products will be able to utilize the waste heat streams from a microturbine, reciprocal or Stirling engines, or the heat generated by solid oxide fuel cells and thermal solar collectors. Thermally activated products based on absorption technology shift the air conditioning and cooling from an electric load to a thermal load.

Objective: The objective of this project is to demonstrate the *feasibility* of a thermally activated space cooling system, for *residential, multi-family, and light commercial* markets, that is fired primarily by a renewable energy source (solar), with the capability of using natural or propane gas as the *back-up* energy source when the solar energy fraction is less than that required to meet the cooling load. The ultimate commercial system consists of a concentrated, evacuated tube solar collector and an air-cooled absorption chiller specially designed to operate on either the thermal energy generated by the solar collectors or by natural or propane gas.

In order to make this system commercially feasible, a refrigerant generator (for the absorption chiller) must be developed that can operate on both solar energy and natural or propane gas. Current small-size generator technology is suitable for direct firing only. During the proposed project, a dual-fired generator was designed, fabricated, and tested on a breadboard chiller. Performance of the generator and the chiller was evaluated using both simulated solar energy and natural gas, using a hydronic boiler to simulate the solar collector energy input.

Outcomes & Conclusions: The generator, as tested, is a modification of the standard gas-fired generator used as part of the Cooltec5^{TM} chiller by the addition of a heat transfer surface to the outside shell of the generator. The model was verified and calibrated based on the experimental data and can be used to fine tune the generator design for the next development stage.

The dual-fired generator prototypes, when fired by hydronic fluid, were able to heat the generator bottom temperature up to the target operating temperature with hydronic inlet temperatures of 460^o F and above. Generator temperatures were within 12% of the target at nominal 400^o F hydronic temperatures. The prototypes retained their ability to reach full operating temperatures when gas fired.

At 95^o ambient conditions and 55^o F return chilled water temperature, the breadboard chiller achieved full cooling capacity with hydronic temperatures greater than 470^o F. A capacity reduction of 6 and 20% was measured with hydronic inlet temperatures on the order of 450^o and 400^o F respectively. The cooling cycle COP when solar fired ranged from 0.7 to 0.77.

As part of a project extension, the prototype generator was fired using both simulated solar heat AND the combustion of natural gas at the same time. The recorded behavior of the cycle under

these conditions provides information required to design the control system for future packaged prototypes.

Based on the results the dual-fired generator will also be able to operate off the waste heat of micro turbines (550^o F), engines ($900-1100^o$ F) and high temperature fuel cells (1000^o F). These applications will increase the sales volume of the proposed product, further reducing the equipment cost for solar fired applications.

Recommendations: The dual-fired generator design is now far enough along in its development to be installed in a packaged chiller for a field test/demonstration in conjunction with an appropriate thermal solar collector. With the dual-fired capability, the future Cooltec5-Solar absorption chiller will be able to operate 24 hours a day, regardless of weather conditions, utilizing the solar energy available at any given time. This feature eliminates the need for redundant cooling systems or expensive thermal energy storage equipment.

The engineering effort required before the field test is initiated primarily involves the hydronic fluid system design (pump, plumbing, expansion tanks, etc), and a control board to logically control our variable speed combustion fan so that the gas input complements the solar energy without over-driving the chiller. Ambient temperatures, cooling load, and other inputs will be used to optimize the overall efficiency of the system by ensuring that supplementary energy (natural gas/propane) is utilized ONLY when the available solar energy is not adequate to satisfy the cooling load.

The dual-fired generator prototype can be improved by adding a parallel hydronic loop path inside the generator. This loop would be a coil of plain tubing, running along side the current weak solution heat recovery coil already part of the standard gas-fired generator. The additional coil will allow the chiller to achieve full capacities at less than 450^o F hydronic temperatures, and reduce the capacity reduction at 400^o F hydronic temperatures to 10% or less. Based on the dual-fired testing, performance can be optimized by re-configuring the generator so that the solar heat and combustion energy enter the generator in series, rather than parallel.

Development of the heat pump version of the Cooltec technology is the ultimate end game, providing solar cooling at cycle COP's approaching 0.8, and heating COP's approaching 1.7. A solar fired heat pump could provide the space heating needs of a building with a third less collector surface area, significantly reducing the installed cost of the solar HVAC system.

Public Benefits to California: Benefits to the California public resulting from the implementation of the proposed dual-fuel solar/gas cooling system include:

1. Monetary savings in the form of reduced utility bills.
2. Reduced reliance on fossil fuels for HVAC system energy needs, leading to lower energy costs and improved environmental conditions.
3. Reduced commercialization time for emerging distributed energy technologies (fuel cells, Stirling engines, micro turbines) by reducing their capital payback period by adding waste energy derived cooling.

1.0 INTRODUCTION

Cooling Technologies, Inc. (Cooltec), has developed a high-efficiency, heat activated absorption chiller (Cooltec5^{tm}) utilizing proprietary heat transfer technology and designed for the light commercial and high-end residential air conditioning markets. The unit converts heat energy to cooling more effectively than competitive chillers and is less expensive to operate than conventional cooling products. Heat sources to operate Cooltec products currently include natural gas and propane. Future products will be able to utilize the waste heat streams from a microturbine, reciprocal or Stirling engines, or the heat generated by solid oxide fuel cells and thermal solar collectors.

Cooltec products fit perfectly with Department of Energy sponsored Integrated Energy Systems and Distributed Generation projects. These programs package cooling, heating, and power for on-site applications that reduce the dependency on centralized power plants. In short, the product addresses the current call for both alternative energy sources and distributed generation. Cooltec's product line strategy is based on current and on-going developments in thermally activated technologies. Thermally activated products based on absorption technology, can change a building's thermal and electric profile, shifting air conditioning and cooling from an electric load to a thermal load.

1.1 Program Objective

The objective of this project is to demonstrate the *feasibility* of a thermally activated space cooling system, for *residential, multi-family, and light commercial* markets, that is fired primarily by a renewable energy source (solar), with the capability of using natural or propane gas as the *back-up* energy source when the solar energy fraction is less than that required to meet the cooling load. The ultimate commercial system consists of a concentrated, evacuated tube solar collector and an air-cooled absorption chiller specially designed to operate on either the thermal energy generated by the solar collectors or by natural or propane gas. A sketch of the proposed system is shown in Figure 1, along with approximate design temperatures and flow rates.

In order to make this system commercially feasible, a refrigerant generator (for the absorption chiller) must be developed that can operate on both solar energy and natural or propane gas. Current small-size generator technology is suitable for direct firing only. During the proposed project, a dual-fired generator will be designed, fabricated, and tested on a breadboard chiller. Performance of the generator and the chiller will be evaluated using both simulated solar energy and natural gas, using a hydronic boiler to simulate the solar collector energy input.

At the conclusion of the project, the design of an affordable, commercially manufacturable dual-fired ammonia-water *generator* will be completed, in addition to proof-of-concept testing on the generator and chiller.

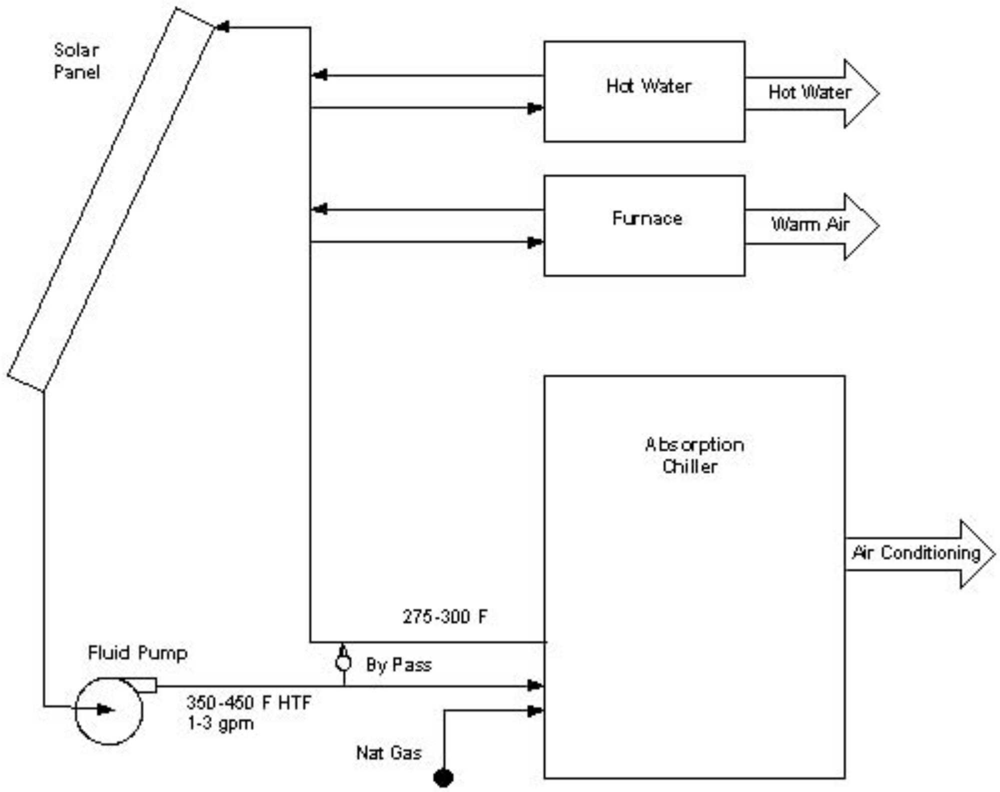

Figure 1: Proposed Solar HVAC System w/Dual-Fired Absorption Chiller

1.2 Energy Problem Targeted

There exists today an imbalance in the supply and demand for electrical energy throughout the U.S. California is acutely aware of this problem, experiencing rolling blackouts and sharply rising prices in early 2001, with expectations for continuing problems through-out the summer. While the state is hoping to resolve these problems by purchasing and managing transmission lines, increasing prices to consumers to reduce consumption, and fast-tracking the construction of natural gas fired electric generating plants, it is a short term solution with a significant negative impact on natural gas supplies and prices.

With reserves of fossil fuels projected to diminish beginning in 2010, and possibly exhausted by the end of the century, it is imperative that utilization of alternative renewable energy sources be expanded immediately. Presently less than 10% of our energy supply is provided by renewable energy, including hydroelectric, geothermal, biomass, wind and solar. With energy costs in the U.S. relatively inexpensive and the cost of equipment that converts or utilizes renewable energy relatively expensive, it has been difficult to convince the consumer to use renewable energy in sufficient volume to drive equipment costs down. Rapidly increasing costs for electric power and natural gas, inadequate and unreliable supply, and new emerging technologies to use renewable energy should change the way consumers consider this alternative.

According to DOE data, residential buildings in the U.S. consume 1.62 quads for space cooling and virtually all of this energy is supplied by grid-connected electrical equipment. An additional 7.52 quads is used for space heating and 2.66 quads for hot water heating. Most metropolitan areas of the U.S. are experiencing significant electric demand and supply imbalance during the summer months due to the increasing use of air conditioning equipment. This project will feature two new technologies to provide space cooling for residential and light commercial buildings. One is the medium temperature (90^0C to 205^0C) solar thermal energy collector technology developed by Duke Solar Energy, LLC (Duke Solar), and the other is the efficient 5RT gas-fired GAX chiller developed by Cooling Technologies, Inc. (Cooltec).

Air-cooled absorption chillers operating on natural or propane gas have been commercially available for over 35 years, and have proven to be reliable and durable, requiring minimal maintenance. Recent technology advances, partially supported by DOE and GRI, have increased efficiencies by over 40% when fired with natural or propane gas. The ability to effectively use solar energy with natural/propane supplemental energy will increase the efficiency 18-20 times, without compromising comfort, reliability or durability.

The concept of solar arrays to provide space heating and domestic hot water for residential and commercial structures has been available for a number of years. The cost to benefit (lower energy cost) ratio has been marginal at best. The dramatically higher costs for electricity and natural gas, and the additional reduction in air-conditioning electricity use with the proposed solar powered chillers, should increase commercial acceptance. The addition of a small gas-fired generator would also allow continued use during power outages at a reasonable first cost.

This concept for a dual-fuel absorption chiller generator has been proven for large tonnage, water-cooled, Lithium-Bromide chillers in a joint project of Duke Solar and the Mechanical Engineering Department of The Ohio State University, partially funded by NREL, that culminated in successful tests in October, 1999. A gas-fired 50-ton Yazaki 2E absorption chiller/heater was modified with a dual-fuel (hot fluid and gas) generator, and the chiller achieved design operating parameters.

The proposed system differs from previous solar powered cooling systems in many respects:

1. **Small Size:** the proposed system targets the residential, multi-family, and light commercial markets (3-25 Refrigeration Tons).
2. **Air-Cooled:** the proposed cooling system is air-cooled and therefore does not require expensive and difficult to maintain cooling towers.
3. **Dual-Fired:** the proposed cooling system will be able to operate on either or both solar energy and natural gas or propane, an important feature for practical, affordable systems.

In order to make this system commercially feasible, a refrigerant generator (for the absorption chiller) must be developed that can operate on both solar energy and natural or propane gas. Current small-size generator technology is suitable for direct firing only. During the proposed project, a dual-fired generator will be designed, fabricated, and tested on a breadboard chiller.

1.3The Cooltec5^{TM} GAX Chiller

The Cooltec5^{TM} is a 5RT gas-fired chiller developed in response to a market assessment commissioned by the Gas Research Institute (GRI) that indicated the need for higher efficiency air-cooled gas-fired cooling equipment. Small chillers have been on the market for almost 40 years, but their low COP of 0.5 or less have rendered them less economical than electric vapor compression chillers. Cooltec has significantly improved this cycle through redesigned heat exchangers that incorporate state-of-the-art and proprietary heat transfer technologies. The Cooltec5^{TM} achieves a gas-fired COP of 0.68, the highest GAX performance to date for commercially viable equipment. When the combustion losses are eliminated, the system thermal efficiency increases to 0.83. The increased efficiency reduces the size (and cost) of the solar panel required to provide the thermal energy for the cooling load. As the solar energy is renewable, the cooling COP when operating on solar energy is between 17 and 19 (cooling capacity divided by the electrical energy required to drive pumps and fans). Assuming the solar availability of 70%, the average system cooling COP will be approximately 12.5.

The ammonia-water absorption cycle is well known and has been used in commercial refrigeration for the past 100 years. The cycle can be summarized in the following steps:

- A strong solution of ammonia and water is heated in the generator so that ammonia vapor is boiled off leaving a weak solution of primarily water.
- The ammonia vapor is condensed and expands through a restrictor to a lower pressure and is cooled. The total heat of the cycle is rejected through the air-cooled absorber and condenser.
- The cold ammonia liquid passes through an evaporator heat exchanger where it absorbs heat from the external chill water that is circulated to it and evaporates.
- The cool ammonia vapor next goes to the absorber where the weak solution absorbs the vapor creating a strong solution.
- The strong solution is pumped to the generator where the cycle repeats.

Cooltec has scheduled modifications to the current chiller that will enable it to operate as a heat pump with a heating cycle COP of approximately 1.7, an efficiency improvement of 90% for the residential and light commercial markets when compared to 90% efficient gas furnaces.

The Cooltec5^{TM} specification sheet is provided in Appendix III.

1.4 The ICPC Evacuated Tube Solar Energy Collector

Duke Solar markets a solar collector array made up of medium temperature collectors that take advantage of Duke Solar's patented non-imaging optics technology together with proprietary coating and manufacturing techniques. The collector is a revolutionary integrated compound parabolic concentrating (ICPC) evacuated tube solar energy collector called the *VAC-2000.* It features a highly reflective metal compound parabolic concentrating (CPC) mirror placed inside an evacuated glass tube along with a stainless steel absorber tube with a state-of-the-art selective coating. This coating maintains high solar radiation absorptivity and low IR emissivity over a wide range of operating temperatures. Eight collectors are assembled with a manifold into a module identified as the VAC-2008. Technical data is included in Appendix IV. A heat transfer fluid, such as water, thermal oil, or other high temperature heat transfer fluid, is pumped through the collectors to and from a thermal storage tank. This heated fluid is then pumped between the storage tank and the heat-driven equipment.

The *VAC-2000* collects solar energy more efficiently than typical glass evacuated tube collectors as well as any available tracking low to mid-range solar thermal collector. It is a simple and reliable unit that collects solar energy efficiently even in low radiation areas, during the winter months, and with operating temperatures higher than those required for double-effect absorption water chillers and the GAX chillers. The expected solar-to-thermal efficiency for this project location operating at 200°C exceeds 50% on an annual basis. No other non-tracking solar thermal energy collector exceeds this performance. This performance means that the roof area of a typical low-rise building up to 4-stories in some locations is large enough for a VAC-2008 collector array that can drive a chiller to meet the peak cooling load of the building. The first such system, that uses an earlier ICPC design collector and a 20-ton Sanyo-McQuay 2E chiller that was modified to run on hot water, has been operating successfully since 1998 in Sacramento, CA. This project has proven that the ICPC non-tracking collector is capable of continually providing 165^0C hot water to drive the chiller.

Fourteen VAC2008 collector modules will be required to power the GAX chiller at peak capacity, and these will require around 660 sqft. of surface area on which to be mounted. This area is available in a typical upscale residential and light commercial building.

1.5 The Cooltec5 Direct Fired Generator

The current Cooltec5 generator design is internally fired, the flue gas travels up through the generator inside a 3.5” OD internally finned fire tube. The ammonia-water solution is boiled inside the annulus between the fire tube and a 6” OD outer shell. Baffles in the annulus assist the heat and mass transfer of the ammonia-water solution flowing downward and the generated ammonia vapor flowing upward. All other commercially available ammonia-water generators are externally fired (the flue gas passes over the OD of the 6” shell). These other designs have numerous disadvantages, primarily excessive energy loss to the environment and no dual-fired capability.

2.0 PIER PROJECT OBJECTIVES

The technical objectives for this initial phase PIER development effort focused on the dual-fired generator component. The specific objectives were to:

1. Design, fabricate and test a dual-fired (solar-gas) generator for a 5RT ammonia-water absorption chiller. Preferably the design is a modification to the current Cooltec5^{TM} generator geometry in order to minimize product cost.

2. Demonstrate that the heat exchanger is capable of heating the working fluid (ammonia-water) up to an operating temperature of 375^{o} F, given a solar source temperature of 450^{o} F.

3. Demonstrate that the dual fired generator, when coupled to the other components of a 5RT absorption chiller, is capable of producing 5RT of cooling at the standard ARI conditions (95^{o} F ambient, 55 - 45^{o} F chilled water).

4. Project Extension: Operate the breadboard chiller while firing the generator on both simulated solar energy and the combustion of natural gas at the same time in order to provide guidance for the packaged prototype control system logic specification.

3.0 PROJECT APPROACH

The research project was accomplished in four phases, three originally planned plus an extension phase. The proposed dual-fired generator was designed, along with a system to simulate a thermal solar panel (hydronic source system) in the first phase. An alpha version of the generator was fabricated and tested in a laboratory breadboard chiller in Phase 2. A beta version of the generator, with changes based on the results of Phase 2, was fabricated and tested in Phase 3. In both Phase 2 and 3, the performance of the generator, and that of the overall chiller system were evaluated. The beta generator was fired on both solar energy and natural gas in Phase 4 in order to provide guidance for future controls development.

The project task titles are listed below, followed by a discussion of the work completed under these tasks.

Phase 1.0: Design/Fabricate Dual-Fuel Generator and Hydronic Source System

T1.1: Re-design the current Cooltec5^{TM} gas-fired generator to incorporate a hot fluid heat exchanger that will transfer heat to the ammonia-water solution at the design heat transfer rate and temperature. The design will enable the generator to be energized by either natural gas or by the hot fluid.

T1.2: The alpha generator designed in Task 1.1 will be fabricated at Cooltec's Johnstown manufacturing facility.

T1.3: A hydronic fluid heating system (solar panel simulator) will be designed and fabricated in order to test the generator using a high temperature heat transfer fluid (such as Dow500) in the Cooltec laboratory. A gas-fired heat exchanger will be designed to heat the hydronic fluid up to 450^{o} F.

Phase 2.0 Install/Test Alpha Generator in Breadboard Chiller

T2.1: The alpha generator will be installed on a full scale breadboard chiller in the Cooltec engineering laboratory.

T2.2: The breadboard chiller will be operated using both the hot hydronic and natural gas to fire the generator. A test matrix will be constructed prior to the testing phase that covers several different hydronic inlet temperatures, flow rates, ambient temperatures and other operating conditions of interest.

T2.3: Collected data will be analyzed and comparisons to the design model will be completed. Differences between measured and predicted performance will be evaluated. If necessary, calibration constants will be added to the design model.

T2.4: Additional testing will be completed in order to repeat tests where the data is suspect or additional data is desired.

Phase 3.0 Generator Design Modifications; Fabricate/Test Beta Generator

T3.1: Based upon the alpha generator testing and model calibration, a beta dual-fuel generator will be designed.

T3.2: The beta generator will be fabricated at Cooltec's manufacturing facility.

T3.3: Any physical changes to the solar simulator system, breadboard, or instrumentation will be completed.

T3.4: The beta generator will be installed in the breadboard chiller.

T3.5: Testing similar to that completed in Task 2.2 and 2.4 will be completed.

T3.6: Test data will be reduced, analyzed, and compared to the design model. The design model will be further refined if necessary.

Phase 4.0 Simultaneous Solar and Natural Gas Firing (Project Extension)

T4.1: Operate the breadboard chiller while firing on both natural gas and solar heat simultaneously, using a variety of solar hydronic temperatures and co-firing logic.

T4.2: Plot and evaluate the operation of the breadboard chiller, focusing on future controls development requirements.

Task 1.1: Dual-Fired Generator Design

To minimize the cost of the ultimate solar-gas chiller product, the preferred dual-fired generator design was based on the existing gas-fired Cooltec5™ generator. More preferably was a design that simply "added parts" to the base generator and did not require an increase in height.

The base Cooltec5™ generator is shown in Figure 1 of Appendix VI. It is an annular design where flue gas flows upward through an inner 3.5" tube and ammonia is boiled from an ammonia-water solution in the annulus between the inner and outer (6" diameter) tubes. Stainless steel fins enhance the heat transfer between the flue gas and the inner tube. Baffles and packing materials located in the annulus help maintain temperature and concentration gradients on the ammonia-water side, an important factor for optimum cycle efficiency.

Prior art for this component consisted of externally fired chambers where the flue gas passed over the outside of a 6" tube (there is no inner tube). Since the high temperature flue gas is on the outside, this design is subject to high ambient energy losses. **Cooltec's internally fired design eliminates this energy loss, while leaving the external surface of the outer shell available for a second heat (energy source) to be integrated.**

The heat transfer surface added to the outer shell of the generator must provide good heat transfer and surface area enhancement enhancement, but must not impose a high pressure drop penalty to minimize parasitic pumping power. High temperature heat transfer fluids have poor heat transfer characteristics, and the flow rate and temperature differences in this application are low. Ammonia-water generator design is governed by the heat transfer resistance of the heating fluid since the ammonia side is two phase boiling heat transfer (very high heat transfer coefficients). Cooltec utilizes offset-strip-fin (OSF) in two other components, the SCAGAX absorber and evaporator and was chosen for these applications because it provides excellent heat transfer enhancement without the high pressure loss penalties. From an economic point of view, it is preferential to use the same OSF the dual-fired generator.

A model of the generator was completed using EES (Engineering Equation Solver, see Appendix I). The alpha design consists of our standard gas-fired generator, with an extended heat transfer surface (OSF) wrapped around the outside diameter. The model calculates the heat transfer resistance on the heat transfer fluid side (OSF) using equations by Manglik & Bergles in Shah[12]. The ammonia-water annulus geometry is quite complex and very difficult to model. It consists of both falling film and pool boiling, with lots of turbulence created by the baffles and vapor counter-flow. Prior gas-fired development work and modeling have also shown that the surface in contact with the boiling solution consists of both wet (surface in contact with solution) and dry areas (surface in contact with vapor). For the purpose of this effort, the ammonia side heat transfer coefficient and wetted surface percentage was set at 800 btu/hr sqft F and 50% based on previous work during the development of the Cooltec5™ (references 4, 8, and 11). The wetted wall percentage is the primary unknown for a new geometry, and will be used to calibrate the model to the experimental data.

Target state points for the generator are shown in Figure 2, while the resulting alpha generator design is shown in Figures 2-4 of Appendix VI. The hydronic fluid enters an annular header near the bottom of the generator before entering a narrow annulus created by the outer generator shell, and a sheet metal shell wrapped around the OSF. All internal flue gas fin surfaces and pathways are retained so that the design could be fired by both a fluid or combustion of a fossil fuel.

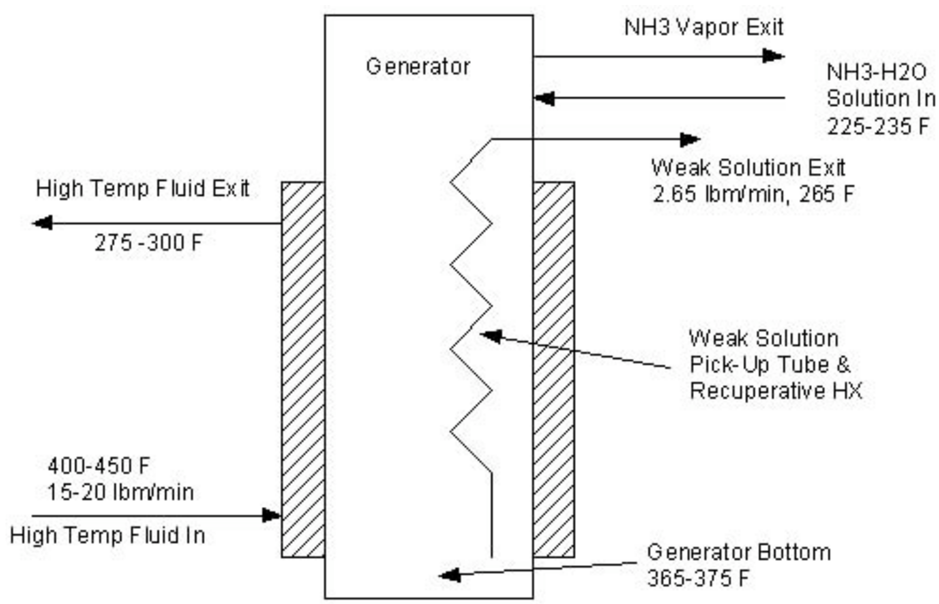

Figure 2: Dual-Fired Generator Target State Points

Task 1.2 Fabrication of Alpha Generator

Fabrication of the alpha generator consisted of the following steps:

1. Fabrication of a standard Cooltec5^{TM} generator.
2. Outer shell preparation (cleaning/sanding).
3. Application of nickel brazing alloy (spray)
4. Application of OSF. The 5” wide by 10” long sections of fin were carefully spot welded to the outer shell so that it would stay in place during brazing.
5. The entire assembly was brazed in a vacuum furnace at approximately 2000^{o} F.
6. Header rings were welded to the outer shell above and below the finned area.
7. A thin sheet metal outer shell was carefully wrapped around the OSF, pulled tight and seam welded.
8. The entire assembly was checked for leaks.

In production, it is estimated that the additional steps will add approximately $150 - $225 to the cost of the generator component. The largest cost is brazing furnace time, which will be dependent upon the production volume (number of generators brazed at a time).

Task 1.3: Design and Fabrication of Solar Simulator System

The “solar simulator system” consisted of a gas-fired heat exchanger, variable speed pump, expansion tank, and flow meter (Figure 3). The heat exchanger was designed and fabricated in-house. EES was again used to design the heat exchanger (Figures 4) which consisted of a length of copper-nickel finned tube coiled around a ceramic plug. The coil was then wrapped with a sheet metal shell, with a 1” layer of ceramic insulation between the coil and the shell.

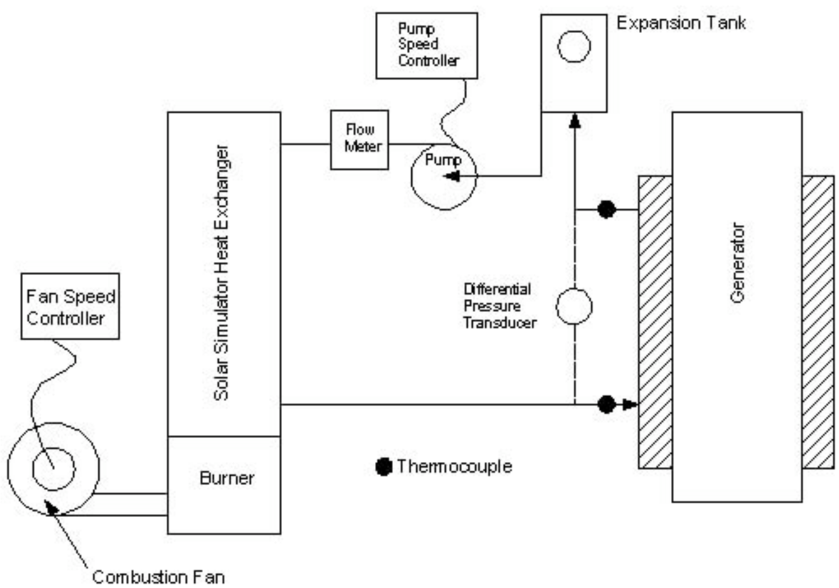

Figure 3: Solar Simulator System

Figure 4: Solar Simulator HX

A standard Cooltec5™ combustion system was installed below the heat exchanger. The combustion system is a 100,000 btu/hr mesh type flame holder and a variable speed, constant fuel-air ratio, combustion fan/gas valve assembly. The variable speed combustion fan and fluid pump permitted both the flow rate and temperature of the heat transfer fluid to be easily controlled during testing.

A review of commercially available high temperature heat transfer fluids was completed. The desired characteristics of the fluid are:

1. Boiling temperature at 1 atmosphere of at least 600° F.
2. Allowable working temperature of at least 600° F.
3. Good heat transfer properties (conductivity, density).
4. Low viscosity (to minimize parasitic pumping power).
5. Non-toxic, environmentally friendly.
6. Low cost.

The results of the review are provided in table form in Appendix V. Generally, high temperature fluids are categorized as organic, aromatic oils, and synthetic oils. Based on this review, Paratherm NF was selected as the working fluid. Paratherm NF has good heat transfer characteristics, has a boiling/working temperature of 600° F, non-toxic, recyclable and reasonably priced. Paratherm NF is basically a highly refined colorless oil that can be recycled with standard engine lubrication oils.

Task 2.1: Assembly of Alpha Generator in Breadboard Test Stand

The alpha generator and solar simulator were assembled into the breadboard test stand located in our engineering laboratory (Figure 5). The breadboard is comprised of all the components necessary for a Cooltec5™ chiller mounted on a rack (so that individual components can be swapped in/out for testing), and an extensive array of instrumentation and controls. The condenser/absorber coil, not shown in Figure 5, is located inside the environmentally controlled "outdoor" room directly behind the breadboard. The ambient temperature inside the outdoor room can be maintained plus/minus 1° F using chillers, heaters and appropriate controls (Figure 6). Instrumentation on the breadboard includes:

- Ammonia Flow (Micro Motion coreolis mass flow meter)
- Weak Solution Flow (Micro Motion coreolis mass flow meter)
- Strong Solution Flow (Micro Motion coreolis mass flow meter)
- Chilled Water Flow (Omega Magnetic Flow Meter)
- High Side Pressure (Transducer + gauge)
- Low Side Pressure (Transducer + gauge)
- Chilled Water Temperature (6 RTDs, 3 inlet, 3 outlet)
- State Point Temperatures (Type T thermocouples)
- Gas Flow Meter (Diaphragm Type, gas temperature and pressure compensation)
- Exhaust Gas Analyzer

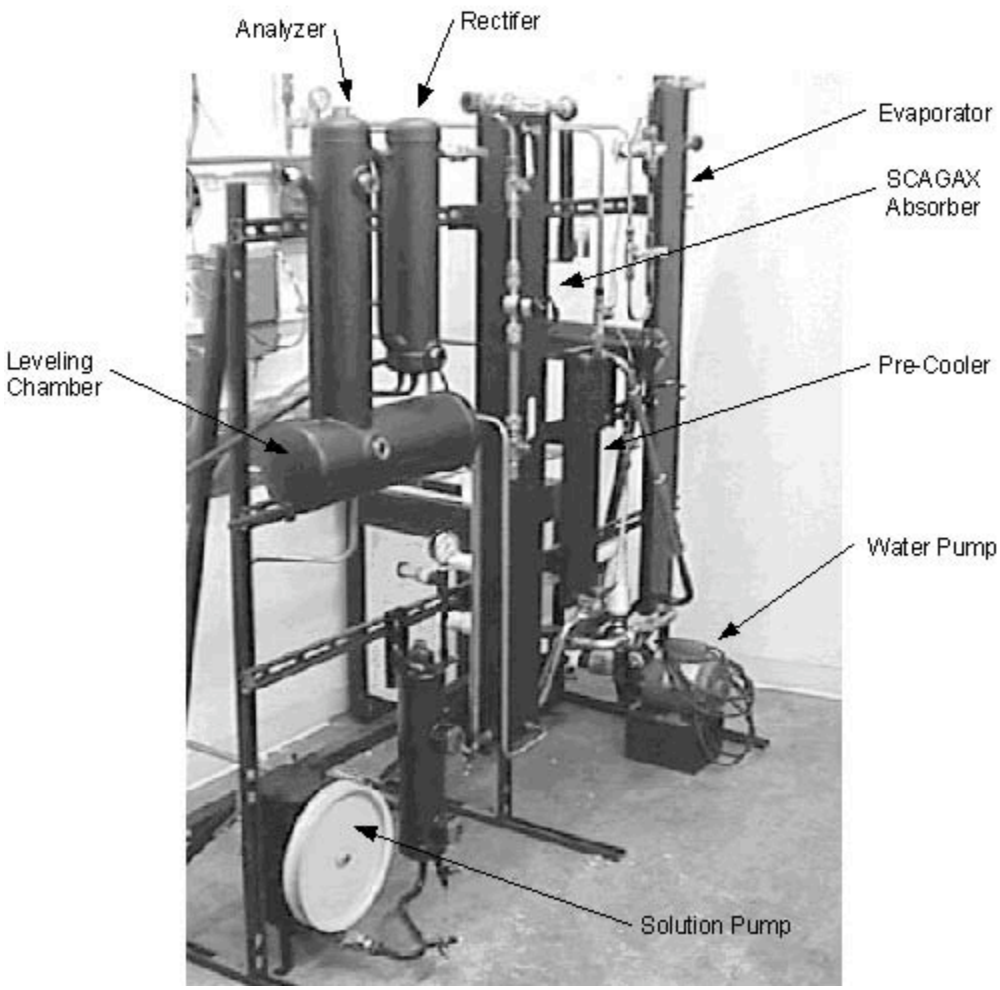

Figure 5: Breadboard Test Stand (before generator was installed)

Before testing was initiated, the breadboard instrumentation was calibrated using industry standard methods. After the alpha generator was connected to the breadboard, it was tested for leaks and charged with an approximate amount of ammonia-water. The generator was then fired conventionally with natural gas in order to confirm proper operation of the breadboard chiller and trim the ammonia-water charge. After operation of the breadboard was confirmed and the charge trimmed, testing using the high temperature fluid was initiated.

Task 2.2 Alpha Generator Testing

With the breadboard complete and the charged trimmed, testing using the solar simulator was then initiated. After operating the solar simulator for a day, it was removed to add insulation in a few “hot” spots. The solar simulator was re-installed in the breadboard and testing initiated. Testing consisted of:

1. Initiation of the solar heat exchanger system and the breadboard chiller.
2. System Warm-Up. Since the breadboard is so spread out and consists of about 600 lb of heat exchangers, it takes some time for the system to reach stable operating temperatures and pressures.
3. State Point Tweaking. The solar simulator burner and pump set points were changed until the hydronic fluid and flow were at the desired points. Since many of the loads were below 5 ton, the solution pump speed was slowed to match the system flows and condenser coil exposed area (using paper towels) was reduced to simulate a 95° F ambient day (denoted by the high and low side pressures).
4. Steady Operation: Once the inputs and flow rates were at the test condition, the operator monitored the cycle state point temperatures and pressures (displayed on the data acquisition (DA) system screen) until they were constant or cycling on a steady average. Data is read and displayed on 4 second intervals, although a screen displaying a rolling 1 minute average is most often used by the operator to confirm steady operation.
5. Data Collection: Data was recorded by the DA system every 4 seconds for a period of 30 minutes. Generator outer skin temperatures were recorded, using a hand-help thermocouple reader, four times during the 30 minute period.

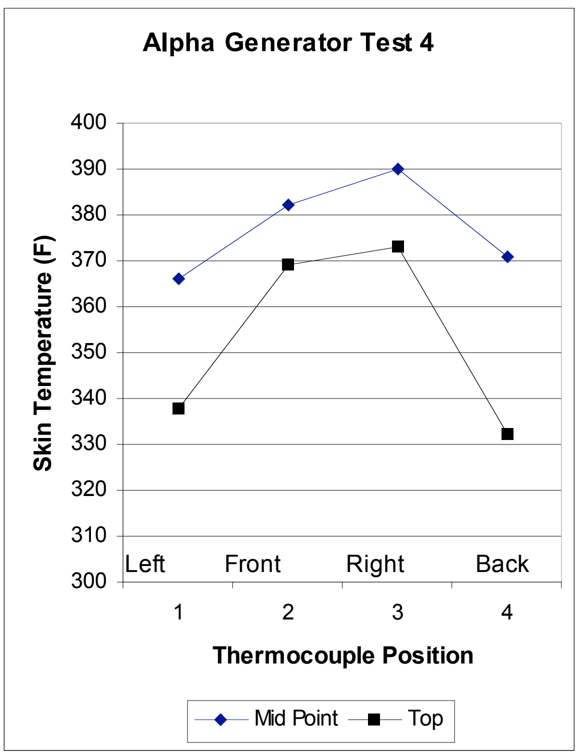

Figure 6: Generator Shell Temperature Distribution

Six data sets (test number 1 – 6) were collected at hydronic fluid flow rates ranging from 15-20 lbm/min and inlet temperatures (solar simulator exit) ranging from 380 – 430° F. An initial look at the temperatures (thermocouples spot welded) collected from the outer surface of the generator showed poor cylindrical distribution of the hydronic fluid (Figure 6). Since the hydronic fluid inlet/outlet tubes were located on the same side of the generator, the outlet tube was moved 180° to the opposite side in an effort to help the distribution. Five more data sets were collected (tests 7 – 12) at hydronic fluid flow rates ranging from 17 – 19.5 lbm/min and inlet temperatures ranging from 400 – 490° F.

Task 2.3 Data Analysis

Data from the first eleven data sets were averaged and summarized in Table 1 in Appendix II. Overall performance (chilled water production) ranged from 2.75 – 4.5 tons, depending on the temperature and flow rate of the hydronic fluid. Performance of the generator (based on the quantity of heat transferred from the hydronic fluid to the absorption cycle, 73,000 btu/hr required to achieve 5 tons of cooling) was about 80% of the desired performance at hydronic fluid inlet temperatures of 400° F (test 7), and about 92% at oil temperatures of 460° F (test 10). Hydronic fluid pressure drop through the generator was very good (less than 2 psi). Moving the oil outlet location increased the heat input to the generator from the hydronic fluid only about 3-5% (Figure 7) and did not significantly improve the poor flow distribution based on the exterior temperature profile (Figure 8).

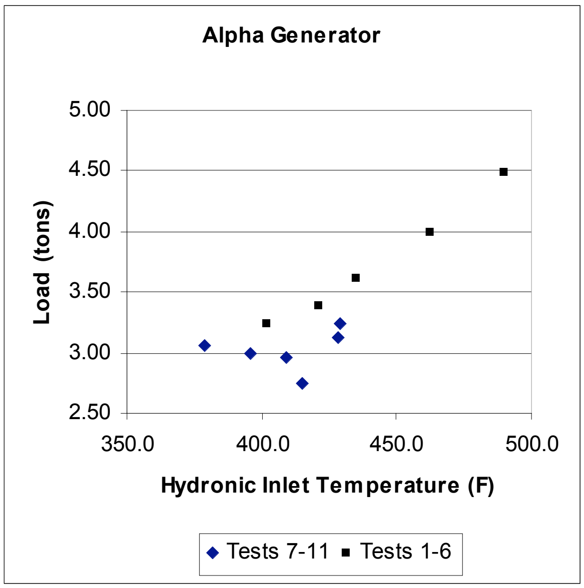

Figure 7: Cycle Capacity vs. Hydronic Temperature

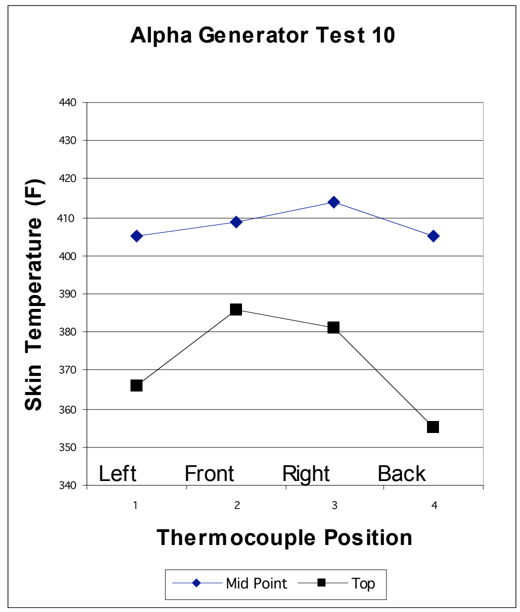

Figure 8: Generator Shell Temp. Dist.

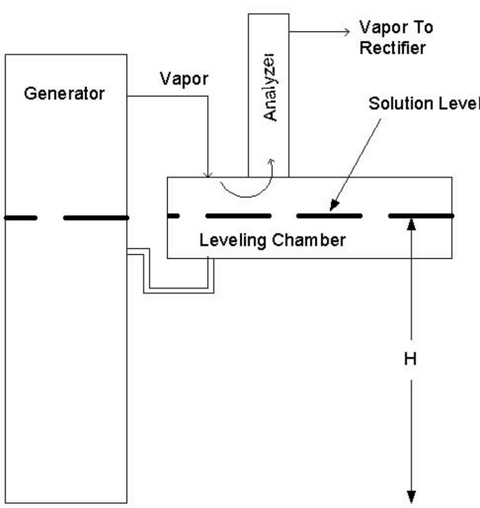

Figure 9: Leveling Chamber Position

Outer generator surface temperatures indicate poor cylindrical distribution, as well as evidence of vertical flow by-pass at the seam weld in the outer skin covering the hydronic fluid heat transfer fin. Since these conditions could not be corrected in the Alpha generator, it was decided to do additional testing with the generator lowered with respect to the solution storage chamber. The elevation of the solution storage chamber determines what portion of the generator operates in a flooded mode vs. falling film. By lowering the generator with respect to the solution chamber (distance H), the flooded portion of the generator is increased, presumably increasing the active boiling surface area (Figure 9).

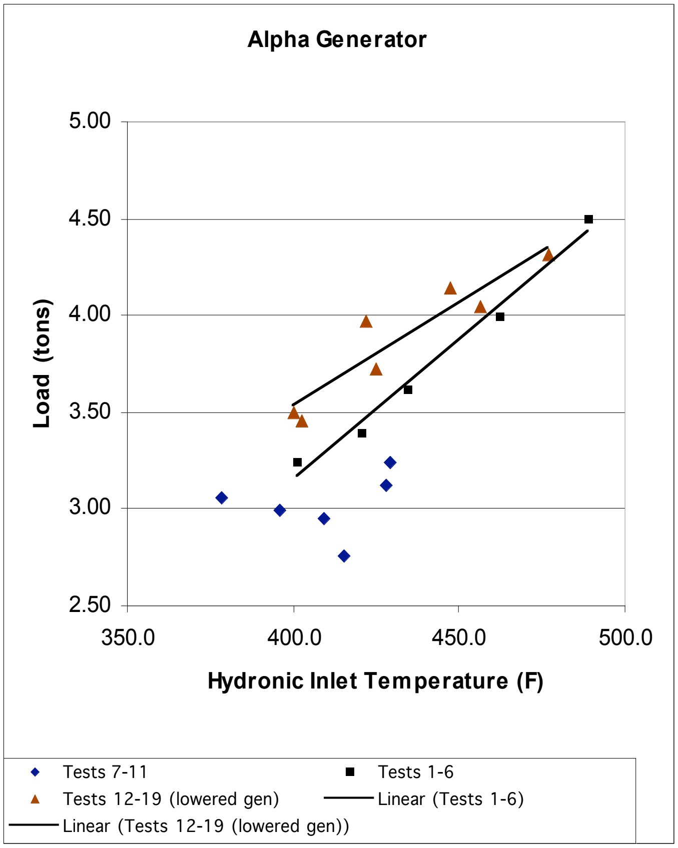

Figure 10: Cycle Capacity vs. Hydronic Inlet Temperature

Task 2.4 Alpha Generator Re-Test

The alpha generator was lowered 10" with respect to the leveling chamber and seven more data sets were collected. Averaged and summarized data from the second set of tests are shown in Table 2 of Appendix II.

Cycle performance (in terms of load) with the generator lowered with respect to the leveling chamber improved approximately 8% at lower hydronic fluid inlet temperatures and approximately 2% at higher fluid inlet temperatures (Figure 10). By raising the solution level in the generator, the upper portion of the generator was more active, causing more ammonia to be boiled and higher measured loads.

Task 3.1: Design Beta Duel-Fuel Generator

Based on the alpha generator performance, the following changes were made for the beta generator:

a. The bottom five inches of heat transfer fin was rotated 90 degrees (easy flow direction perpendicular to flow direction) to improve the flow distribution. This will increase the hydronic fluid pressure drop somewhat, but it was very low in the alpha generator (2" w.c.).

b. Five inches of OSF was added (maximum that will fit on our standard gas-fired generator).

c. The outer skin (thin sheet metal) that is wrapped around the outside of the heat transfer fin was butt welded at the joint instead of using a lap joint. Executing a lap joint with no by-pass flow area proved very difficult on the alpha generator.

The beta generator is shown in Figures 5-7 in Appendix VI.

Task 3.2: Fabricate Beta Generator

Fabrication of the Beta generator followed the same procedure as the Alpha.

Task 3.3/3.4: Physical Changes to Breadboard & Install Beta Generator

No physical changes to the breadboard were required. The Alpha generator was removed and the Beta generator installed in its place with no significant modifications.

Task 3.5: Beta Generator Testing

Breadboard testing with the Beta generator proceeded similarly to the Alpha generator testing. The Beta generator was tested initially in the "lowered" position and then in the normal position. In both positions, data was collected when fired on natural gas, which provided a breadboard performance normalization point (breadboard systems typically operate 2-5% less efficiently than packaged systems due to extra heat losses to the ambient and extra pressure losses. Averaged data for the Beta generator in the "lowered" and "normal" positions are provided in Tables 3 and 4 in Appendix VI.

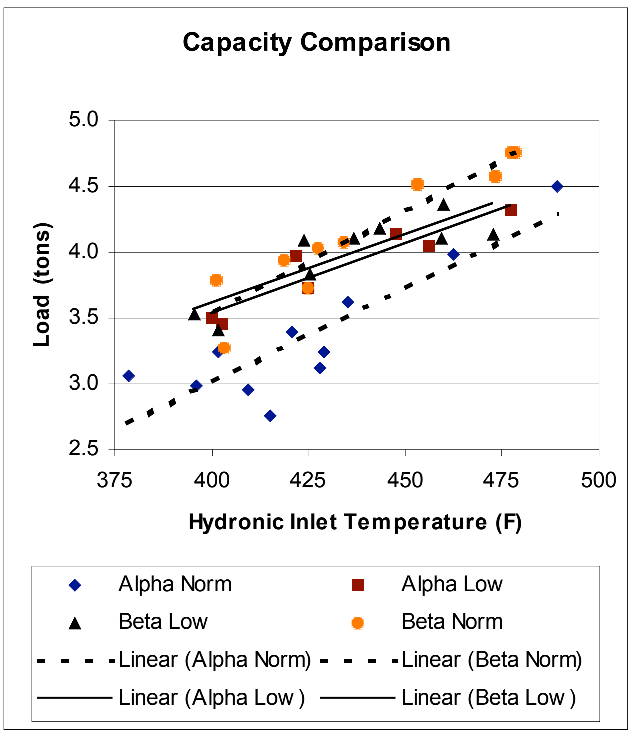

Figure 11: Cycle Capacity vs. Hydronic Temperature

A comparison between the four test cases (Alpha Low/Norm, Beta Low/Norm) in terms of the measured load vs. inlet hydronic temperature is shown in Figure 11. With the extra amount of OSF, the Beta generator out-performed the Alpha generator when the leveling chamber was in its normal "stock" position. However, unlike the Alpha generator results, the performance of the Beta generator decreased when the leveling chamber was lowered 10 inches. This finding is explored more in the Outcomes section below.

Hydronic fluid pressure drop increased dramatically due to the first five inches of OSF being installed sideways in order to improve fluid distribution. The pressure loss ranged from 6-18 psi depending upon the flow rate. As shown in Figure 12, the the hydronic fluid temperature distribution was not significantly different than that found with the Alpha generator once the inlet/outlet points were moved 180° from each other. Since the hydronic pressure drop with the Beta generator exceeded the range of the differential pressure transducer, the transducer was replaced with a calibrated 0-30 psi gauge. The operator manually recorded the inlet and outlet pressures (by opening/closing isolation valves) to determine the pressure drop.

Task 3.6: Test data reduction, analysis, and model calibration.

After all of the testing was completed, the remaining data was averaged and reduced to a single spread sheet. Performance parameters were calculated for each test including:

Cycle Cooling Capacity (tons)
Coefficient of Performance (COP)
Energy Into Generator (btu/hr)
Generator LMTD (^{o}F)
Generator UA (btu/hr ^{o}F)

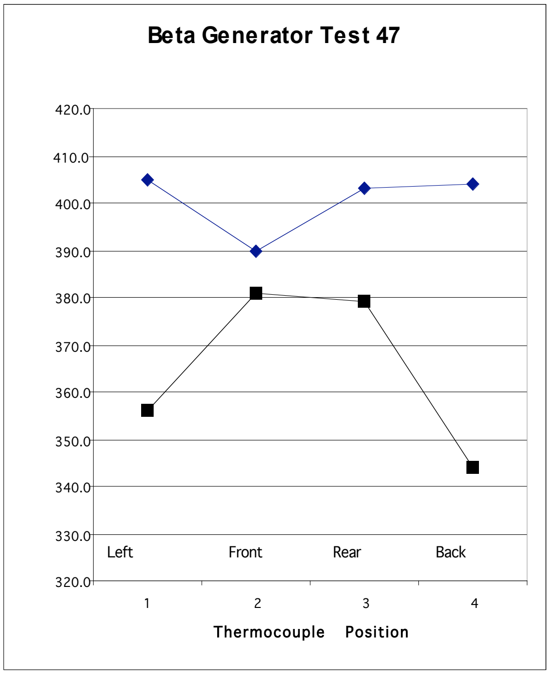

Figure 12: Generator Shell Temp. Dist.

Coefficient of Performance (COP) for absorption machines is calculated by dividing the cooling capacity by the energy input:

COP = Cooling Load / Energy Input

Generator LMTD is the log mean temperature difference between the fluids exchanging heat given by the equation:

$$\mathbf{LMTD = (\Delta T1 - \Delta T2) / (\ln \Delta T1/\Delta T2)}$$

where: $\Delta T1 = T_{hydronicin} - T_{generatorbottom}$
$\Delta T2 = T_{hydronicout} - T_{generatorin}$

Generator in temperature is sometimes referred to as T_{gax}, as it is the solution leaving the GAX absorber. Generator UA is the overall heat transfer coefficient from the hydronic fluid to the ammonia solution times the effective surface area over which that heat transfer take place. It is derived from the equation:

$$\mathbf{Q = UA(LMTD)}$$

Where Q is the energy transferred from the hydronic fluid to the ammonia solution in btu/hr. Averaged data with calculated performance parameters is listed in Tables 3-4 in Appendix II.

After the experimental data was averaged and organized, it was compared to the design model (Appendix I). Inputs to the design model were:

1. Hydronic Inlet Temperature (F)
2. Hydronic Flow Rate (lbm/min)
3. Generator Bottom Temperature (F)
4. Weak Solution Flow Rate (lbm/min)
5. Weak Solution Exit Temperature (F)
6. High Side Pressure (psia)

With this information, the model calculated the hydronic and ammonia side heat transfer coefficients and surface area, and using energy and mass balances, calculated the energy input transferred from the hydronic fluid to the ammonia solution, hydronic outlet temperature, generator UA, and generator LMTD. The calculated values (energy transferred, UA) were then compared to the experimental results. The modeling parameter which was the least understood was the wetted wall area (see Task 1.1). Therefore, the wetted wall area in the model was adjusted until the average deviation between experimental and calculated for the energy transferred (Q) and UA was less than 2% on average for a data set group. There were four data set groups, Alpha Normal, Alpha Low, Beta Normal and Beta Low.

Task 4.1: Operate the Breadboard Chiller Firing on Both Natural Gas and Solar Heat

In Task 4.1, both the solar simulator heat exchanger and the beta generator were outfitted with our standard gas burner and variable speed combustion fan. The breadboard chiller was started and brought up to temperature using the solar hydronic heat energy. Once the breadboard was running at a fairly consistent capacity, the generator gas burner was initiated. The gas burner was manually turned on and off in relation to the ammonia-water solution temperature at the bottom of the generator. When the generator temperature reached the standard operating temperature of 365-375^{o} F, the burner was turned off. The gas burner was turned back on when the generator bottom temperature fell to a pre-determined point. Cycle performance data was recorded continuously during these tests.

These tests simulated the case when the available solar energy is not high enough to satisfy the cooling demands of the load. Therefore, most of the testing utilized low solar hydronic temperatures (300–400^{o} F). The ammonia flow controller (TEV), weak solution fixed restrictor pack, and solution pump were not modified from the standard configuration.

Nine test cases were completed. The full set of recorded data is given in Table 5 of Appendix II. Nominal test conditions for the nine test cases is given below.

Nominal Parameters For Dual-Fired Test Cases

Test#	Nominal Hyd. Temperature (^oF)	Gas Input (btu/hr)	Burner On/Off Temperature (^oF)*
54	400	32,000	330/365
55	425	32,000	340/365
57	450	30,000	350/375
58	475	33,000	350/375
59	300	25-45,000	250/365
60	300	25-45,000	250/365
61	350	31,000	250/365
62	350	30,000	280/365
63	350	30,000	300/365

*** Generator Bottom Temperature**

T4.2: Plot and evaluate the operation of the breadboard chiller, focusing on future controls development requirements

Figures 1– 9 of Appendix II are plots of the key state points and capacity vs. elapsed time. Plots for each test case include:

1. Hydronic Inlet Temperature
2. Generator Bottom Temperature
3. Weak Solution Generator Exit Temperature
4. Normalized Cycle Capacity
5. Low Side Pressure

All of these state points cycled as the gas burner was turned on and off. The amount of time the gas was on or off depended on the hydronic temperature and the target control point (generator temperature). Capacity varied over a much wider range for the low hydronic temperature cases as the load developed by the hydronic energy alone was much lower than when supplemented by natural gas.

The hydronic inlet temperature cycled with the gas burner due to a decrease in the amount of energy transferred from the hydronic to the ammonia side decreased as the generator temperatures increased when the gas burner was on. Generator bottom and weak solution exit temperatures also cycled (as expected) as the gas burner turned on/off. Somewhat unexpected was the magnitude of the low side pressure cycling due to the varying ammonia flow created in the generator.

4.0 Project Outcomes/Results

The summary results of the PIER project are (with respect to the project objectives):

1. A dual-fired (solar-gas) generator was modeled, designed, fabricated and tested for a nominal 5RT ammonia-water absorption chiller. The generator, as tested, is a modification of the standard gas-fired generator used as part of the Cooltec5™ gas-fired chiller. The modification is the addition of a heat transfer surface to the outside shell of the standard generator. The model was verified and calibrated based on the experimental data and can be used to fine tune the generator design for the next development stage.

2. The dual-fired generator prototypes, when fired by hydronic fluid, were able to get the generator bottom temperature up to the following values (365 – 375° F required to achieve full capacity):

Test#	Generator	Hydronic (°F)	Generator Bottom (°F)	Percent From Target
37	Beta Low	396	331	12%
13	Alpha Low	400	331	12%
47	Beta Normal	453	359	4%
40	Beta Low	460	367	2%
50	Beta Normal	478	380	+1%

When fired on gas, the generator bottom temperature ran at 371-373° F (beta generator, test numbers 31 and 52.

3. At 95° ambient conditions and 55° F return chilled water temperature, the breadboard chiller achieved the following capacities (normalized to capacity achieved gas-fired to account for breadboard inefficiencies, see Task 3.5) and efficiencies (ARI performance certification requires measured capacity to be within 5% of nameplate rating):

Test#	Generator	Hydronic (°F)	Cooling Capacity tons	Cycle COP	Percent of Target
44	Alpha Low	401	4.0	0.75	80%
47	Beta Normal	453	4.7	0.77	94%
50	Beta Normal	478	4.9	0.70	98%

When gas-fired, the breadboard chiller achieved cooling capacity of 4.8 tons at a COP of 0.66 (the gas-fired COP includes a combustion efficiency of 82%, which equates to a cycle COP of 0.80).

4. The breadboard chiller operated successfully while being fired simultaneously on simulated solar and gas combustion sources, using generator bottom temperature as the control point for determining whether the gas burner was on or off. Capacity and several of the key state points

cycled as the gas burner was turned on and off, becoming more pronounced (cycle magnitude) as the hydronic temperature dropped.

These findings provide invaluable information concerning what control features must be developed for the packaged prototype so that the chiller can operate optimally over a wide range of solar energy availability and load requirements.

4.1 Model Verification & Calibration

As described under Task 3.6, the generator model was verified and calibrated to the experimental data, specifically the determination of the wetted wall area. Initially, the model used a wetted wall area of 50%, a typical value found during previous modeling/testing efforts for the gas-fired Cooltec5^{TM}. The four test groups (Alpha Generator Normal/Low, Beta Generator Normal/Low) were evaluated independently since it was anticipated that the geometrical differences would result in different wetted areas. The wetted wall areas that achieved on average, a less than 2% difference between measured and actual energy transferred and UA were:

Generator	**Wetted Wall %**
Alpha Normal	48
Alpha Low	65
Beta Normal	37
Beta Low	40

These results illustrate important information regarding what was going on inside the generator. For the Alpha Normal case, the hydronically heated section is very close to the same length as the gas-fired section, as well as the solution level. The wetted percentage for the Alpha Normal (48%) is very close to what was found during gas-fired product development (50%). By raising the solution level inside the generator (by lowering the generator with respect to the leveling chamber (Alpha Low)), the wetted percentage increased dramatically from 48 to 65%

For the Beta Normal case, the five extra inches of OSF raised the hydronically heated section above the normal solution level. While this extra length increased the overall performance of the generator and cycle capacity, the effectiveness of this extra "fired area" was not very good, decreasing the average wetted area compared to the Alpha Normal case. Increasing the solution level in the Beta generator (Beta Low) did not significantly increase the wetted area as it did for the Alpha generator. In fact, other performance parameters (discussed below) indicate the combination of extra OSF and raised solution level created a fundamental fluid dynamics change inside the generator that actually decreased overall performance.

Measured generator UA vs. energy transferred from the hydronic to the ammonia solution is shown in Figure 13. It is anticipated that the UA should increase slightly with increased energy input due to increased solution/vapor flow rates and the associated turbulence. This was the case for the cases in which the leveling chamber was in the normal position, and as anticipated, the UA's obtained with the Beta generator were greater than the Alpha. However, for the cases were the generator was lowered, the UA decreased (at a very fast rate) with increasing energy

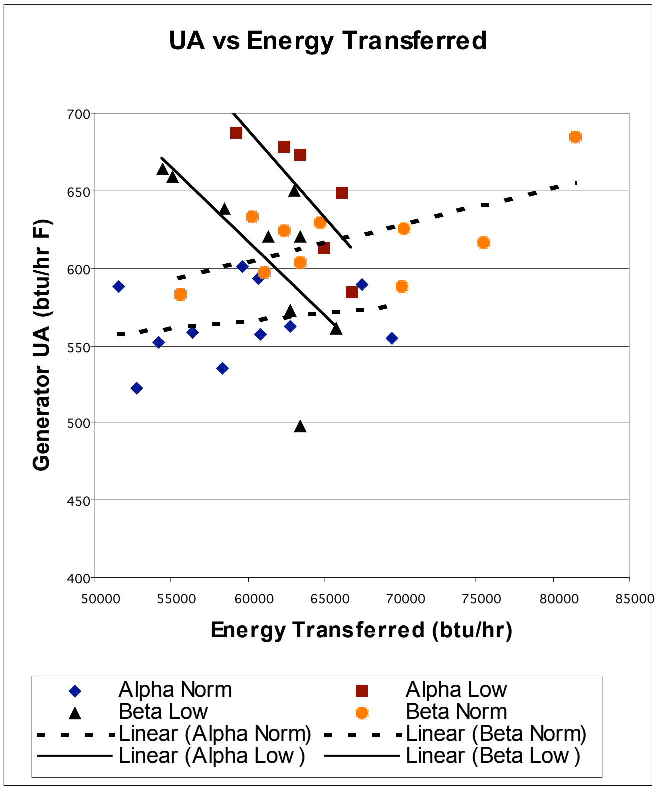

Figure 13: Generator UA vs. Energy Input

input, and the UA's obtained by the Alpha generator exceeded those of the Beta. These results indicate that raising the solution level and putting more heat energy into the upper part of the generator creates a fundamental change in generator performance and flow dynamics.

The probable explanation for this phenomena is flooding or recirculation. Ammonia-water absorption components must be carefully designed to prevent flooding since vapor flows are always going up and solution flows down. If the vapor velocity is too high, it will stop and/or reverse the flow of solution. Since the top portion of our generator was not specifically designed to contain a solution level (it is falling film) or heat input, it is likely that the combination of these two changes created a flooding problem, especially with higher energy inputs. The extra OSF fin length in the Beta generator caused the problem to be more significant than that found for the Alpha.

Note that as long as the leveling chamber position is "normal" the extra OSF installed in the Beta generator increased performance (in terms of cycle capacity) to within 6% of the target at nominal 450^{o} F fluid temperatures and did not appear to cause flooding.

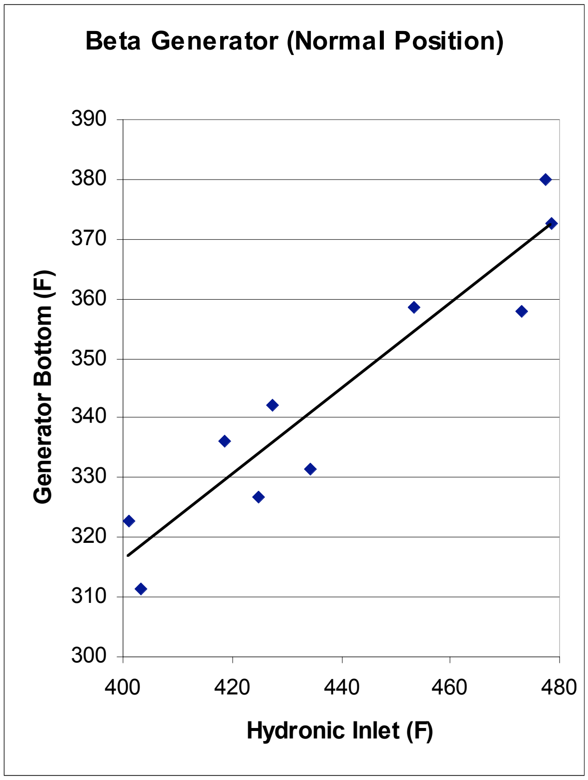

Figure 14: Effect of Hydronic Temperature On Generator Bottom Temperature

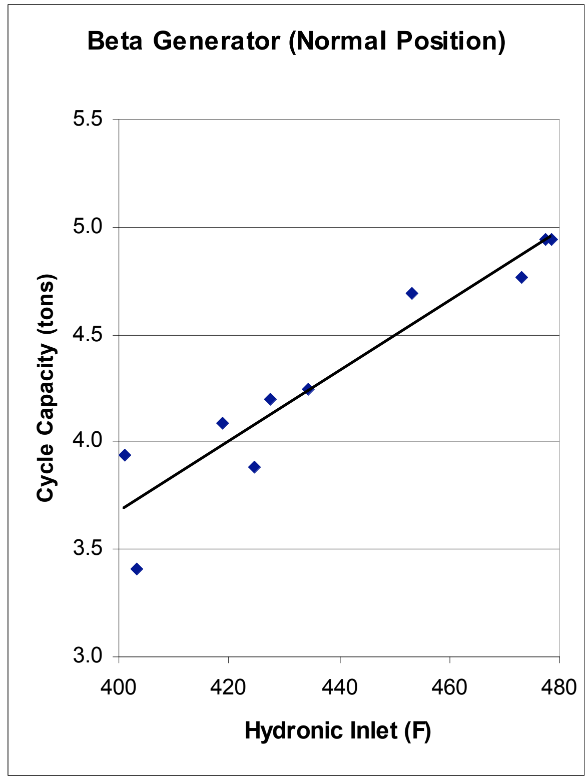

Figure 15: Effect of Hydronic Temperature On Cycle Capacity

4.2 Generator Bottom Temperature

The generator bottom temperature is an important state point related to the performance of an ammonia-water absorption cycle. The temperature determines the weak solution concentration and saturation temperature when the weak solution enters the SCAGAX absorber. The lower the concentration and higher the temperature, the more energy can be internally recovered in the SCAGAX, increasing cycle efficiency and capacity. The standard gas-fired chiller is designed to operate with a generator bottom temperature of 365 – 375^o F, which results in a gas-fired COP of 0.68 (cycle COP of 0.83).

The generator bottom temperature obtained by the Beta generator (normal position) as a function of hydronic inlet temperature is shown in Figure 14. Hydronic temperatures above 460^o F result in generator bottom temperatures of about 365^o F, the minimum required to achieve target cycle efficiencies. The scatter in the Figure 14 data is due to varying hydronic flow rates.

4.3 Cycle Cooling Capacity

Measured cooling capacities (normalized to the gas fired capacity to account for breadboard losses) as a function of hydronic inlet temperatures is shown in Figure 15. Cooling capacity ranged from just under four tons (1 ton of cooling = 12,000 btu/hr) at hydronic temperatures of 400^{o} F, to just under five tons at hydronic temperatures above 470^{o} F. Cycle COP's (btu's of cooling obtained for every btu of solar energy captured) ranged from 0.7 to 0.78.

4.4 Dual-Firing

The breadboard chiller operated successfully while being fired simultaneously on simulated solar and gas combustion sources, using generator bottom temperature as the control point for determining whether the gas burner was on or off. Capacity and several of the key state points cycled as the gas burner was turned on and off, becoming more pronounced (magnitude of cycle) as the hydronic temperature dropped. The cycling was exacerbated by the natural function of the ammonia TEV valve, which was continuously hunting for the optimum ammonia flow rate under continuously changing conditions, and the fixed weak solution flow orifice, which was never able to deliver the optimum weak solution flow rate for the current operating condition. Additionally, the amount of energy transferred from the hydronic to the generator decreased when the gas burner was operating, due to the lower LMTD between the ammonia side and the hydronic.

These findings provide valuable information concerning control specifications that must be developed for the packaged prototype so that the chiller can operate optimally over a wide range of solar energy availability and load requirements:

a. Maximize the turn-down ratio of the gas burner so that the required additional energy from the gas burner can be "trickled" into the generator very slowly so that the gross on/off cycling can be minimized.

b. Improve the reaction time and dampening characteristics of the TEV valve so that it can react more quickly to changing ammonia flow rates.

c. Add the ability to control the flow of weak solution exiting the generator so that it does not exit too hot or too cold.

d. Possibly add solution pump speed control so that it does not run dry during low solar availability with corresponding low capacity requirements.

The reduction in energy transferred from the hydronic to the ammonia solution when the burner is operating indicates the optimum arrangement would be to have these two energy inputs be configured in series, rather than parallel. The solar heat would be added in the upper (cooler) portion of the generator, while the combustion heat would be added in the bottom (hotter) portion. This will require additional height be added to the generator so a cost/benefit analysis will be required to see if the additional cost is worthwhile.

5.0 Conclusions

1. A dual-fired (gas-solar) generator for an ammonia-water absorption chiller was developed and successfully tested. The generator design is fully capable of operating on natural gas, solar energy (via a hydronic fluid), or both at the same time. The generator is a simple adaptation of the standard Cooltec5™ gas-fired chiller generator, which will minimize the extra cost associated with the solar-gas product The additional cost for the dual-fired generator is anticipated to be approximately $150 - $225 in production quantities.

2. Capacity (cooling load) of the solar-fired chiller is a function of the hydronic inlet temperature. Full chiller capacity (nominal 5 tons) was achieved at hydronic temperatures above 470^o F. Capacity is reduced by 6% at 450^o F hydronic temperatures and by 20% at 400^o F hydronic temperatures.

3. The dual-fired generator retained its ability to achieve full capacity when gas-fired.

4. The cooling cycle COP measured when solar fired ranged from 0.7 to 0.77.

5. The dual-fired generator design is far enough along in its development to be installed in a packaged chiller for a field test/demonstration in conjunction with an appropriate thermal solar collector.

6. With the dual-fired capability, the future Cooltec5-Solar absorption chiller will be able to operate 24 hours a day, regardless of weather conditions, utilizing the solar energy available at any given time. This feature eliminates the need for redundant cooling systems or expensive thermal energy storage equipment.

7. Based on the results the dual-fired generator will be able to operate off the waste heat of micro turbines (550^o F), engines (900-1100^o F) and high temperature fuel cells (1000^o F). These applications will increase the sales volume of the proposed product, further reducing the equipment cost for solar fired applications.

6.0 Recommendations

1. Based on the testing results, the dual-fired generator is ready to be incorporated into the packaged Cooltec5™ chiller and installed in a controlled field test with a suitable thermal solar collector. We recommend the initial field test be located at a facility with engineering or research capabilities (university, national or private lab, public utility building, etc.) so that the test equipment can be constantly monitored and evaluated.

2. The engineering effort required before the field test is initiated primarily involves the hydronic fluid system design (pump, plumbing, expansion tanks, etc), and a control board to logically control the variable speed combustion fan so that the gas input complements the solar energy without over-driving the chiller and causing cycling. The combustion fan speed controller will use one or more absorption cycle state points (generator bottom temperature) as the control input signal. Adding additional control capabilities to the ammonia TEV and perhaps weak solution restriction may also be necessary to allow the chiller to operate at peak efficiencies when dual-fired. Ambient temperatures, cooling load, and other inputs could be used to optimize the overall efficiency of the system by ensuring that supplementary energy (natural gas/propane) is utilized ONLY when the available solar energy is not adequate to satisfy the cooling load. Most cooling units need operate at full capacity for only a small portion of their lifetime (10-15%).

3. The dual-fired generator prototype can be improved by adding a parallel hydronic loop path inside the generator. This loop would be a coil of plain tubing, running along side the current weak solution heat recovery coil already part of the standard gas-fired generator. The additional coil would add less than $15 to the cost of the generator and should allow the chiller to achieve full capacities at less than 450^{o} F hydronic temperatures, and reduce the capacity reduction at 400^{o} F hydronic temperatures to 10% or less.

4. An exhaust gas to hydronic fluid recuperator should be developed to convert the gas phase waste heat from micro turbines and engines to a liquid form (hydronic) for use in this design. A single recuperator could collect the waste heat from several turbines/engines and distribute the energy to multiple dual-fired chillers.

5. Development of the heat pump version of the Cooltec technology is the ultimate end game. A dual-fired heat pump would provide solar cooling at cycle COP's approaching 0.8, and heating COP's approaching 1.7. Therefore a solar fired heat pump could provide the space heating needs of a building with a third less collector surface area, significantly reducing the installed cost of the solar HVAC system.

7.0 Public Benefits to California

Benefits to the California public resulting from the implementation of the proposed dual-fuel solar/gas cooling system include:

4. Monetary savings in the form of reduced utility bills.
5. Reduced reliance on fossil fuels for HVAC system energy needs, leading to lower energy costs and improved environmental conditions.
6. Reduced commercialization time for emerging distributed energy technologies (fuel cells, Stirling engines, micro turbines) by reducing their capital payback period by adding waste energy derived cooling.

7.1 Monetary and Energy Savings: Standard electric air conditioners consume about 1.25 kW per ton while Cooltec's solar/gas units will use about .3 kW per ton. Therefore, each ton of heat activated absorption cooling replaces about 0.95 kW demand for air conditioning, a 75% reduction. Assuming an annual average of 2500 cooling hours, 10,000 units installed over the next 10 years will reduce air conditioning power demand by 120,000 MWhr annually.

Table 1 gives the monthly energy loads for space heating, cooling and domestic hot water for a typical 2000 sqft residence in Raleigh, NC with a 3 ton cooling load. This analysis was completed by Duke Solar Energy, LLC and is based on a building load simulation provided by Insite Software. The analysis shows that the energy saved by utilizing a solar heating and cooling system with a Cooltec5™ dual-fired chiller is 71%, with a corresponding utility cost reduction of 78%.

Comparative estimated installed cost for the conventional and solar HVAC systems is shown in Table 2. Based on the current solar panel installed cost of $165/sqm, the resulting payback for the solar system is 7.1 years. With a rebate of $500 or $1000 per ton, the simple payback drops to 5.8 and 4.2 years respectively.

With volume and improved manufacturing techniques, the installed cost of the solar panels is expected to drop to $100/sqm, resulting in a simple payback of 4.5 years. With a rebate of $500 or $1000 per ton, the simple payback drops to 3.2 and 1.6 years respectively.

This economic analysis is dependent upon the building location, type and structure, as well as the local utility rates. However, the analysis shows that a solar HVAC system utilizing a GAX chiller can change a typical building HVAC energy profile by shifting the cooling from an electric load to a thermal load, reducing energy usage by 60 – 80%, and have a simple payback in 1.6 – 7.1 years dependent upon rebate programs and reduced solar panel costs.

Table 1: Conventional HVAC to Solar HVAC with Cooltec5 Dual-Fired Chiller

Month	Solar Energy Collected kWh	Domestic Hot Water kWh	Heating Load kWh	Cooling Load kWh	Total Load kWh	Solar Energy Utilized kWh	Auxiliary Energy Required kWh
Jan	2722	1052	3195	0	4247	2722	1524
Feb	2912	1052	2681	0	3733	2912	820
March	3727	1052	2100	132	3284	3283	0
April	3953	1052	755	176	1983	1983	0
May	3024	1052	196	1494	2742	2742	0
June	2246	1052	0	3428	4480	2246	2233
July	2280	1052	0	4746	5798	2280	3518
Aug	2237	1052	0	4373	5425	2237	3188
Sept	3184	1052	0	2219	3271	3184	87
Oct	3510	1052	778	439	2269	2269	0
Nov	2842	1052	1888	0	2940	2842	98
Dec	2572	1052	3104	0	4156	2578	1577
Annual	**35209**	**12624**	**14697**	**17007**	**44328**	**31278**	**13045**

Installation Type	Residence
Location	Raleigh, NC
Size	2000 sqft
Occupancy	4
Walls	R-11
Roof	R-19
Windows	200 sqf Double Glazed
Equipment	0.75 watts/sf
Lighting	0.75 watts/sf
Gas Cost	$1.00/therm
Electric Cost	$0.12/kWh
Annual Cost Conventional	$1,995
Annual Cost Solar HVAC	$444
Annual Savings	**$1,551**

Table 2: Simple Payback, Solar/GAX HVAC to Conventional HVAC
Residential Installation with 5 ton Cooling Load and Three Zones

	Solar HVAC $165/sqm	**Solar HVAC $100/sqm**	**Conventional HVAC**
Dual-Fired Chiller, Installed	$8,000	$8,000	
Solar Panel Size (sqft)	660	660	
Solar Panel Installed Cost ($/sqm)	165	100	
Total Solar Panel Installed Cost	$10,065	$6,100	
5 ton, 3 zone Electric DX Air Conditioner, Installed			$5,000
Gas High Efficiency Furnace, Installed			$2,000
Domestic Hot Water Heater, Installed			$900
Total Installed Cost	$18,890	$14,860	$7,900
Utility Savings, Annual	$1,551	$1,551	
Simple Payback, Years	**7.1**	**4.5**	
Simple Payback with $500/ton Rebate (years)	**5.8**	**3.2**	
Simple Payback with $1000/ton Rebate (years)	**4.2**	**1.6**	

7.2 Reduced Commercialization Time for Emerging BCHP Technologies: Over the last few years, substantial interest has developed for products utilizing alternative/renewable energy sources and distributed energy technology. Factors influencing the interest include:

- The need to reduce ozone depleting emissions supported by the current energy policy.
- The deregulation of electric utilities, offering new opportunities to generate clean, economical, reliable power on site.
- DOE commitment to doubling the number of installed alternative energy and distributed generation cooling heating and power systems by 2010, an incremental 50,000 megawatts of capacity.

A variety of new products are being developed to target one or more of these factors, e.g., thermal solar panels, micro turbines, fuel cells, and Stirling engines. Cooling Technologies dual-fired chiller enables these alternative energy systems to both expand market applications and accelerate market penetration by improving the economic story for these systems. Utilizing the waste heat during the summer cooling months in addition to winter heating and domestic hot water reduces the payback period to an acceptable number. Cooltec's air-cooled GAX chillers are especially well suited to smaller BCHP installations (30-60 kW power generation) where the customer does not want the maintenance requirements of cooling towers required for lithium-bromide absorption systems.

8.0 Development Stage Assessment

8.1 Project Summary: Cooling Technologies has completed the technical feasibility and the planning for demonstrating commercial feasibility. Cooltec has completed PIER Stage 3/Gate 3, the research and prototype testing phase, and some of the elements of Stage 4/Gate 4 and beyond.

The full scale Cooltec solar powered generator has been successfully tested on a laboratory breadboard, and proven to be completely operational on natural gas, solar energy, or some combination thereof. The generator is a cost effective adaptation of the standard Cooltec5 generator. This dual fired generator design is ready to be installed in a packaged chiller for field test in conjunction with an appropriate thermal solar collector. Cooltec intends to seek PIER follow-on funding in conjunction with the funding/demonstration resources of DOE/NREL.

8.2 Sales & Marketing: Anticipating the evolution of alternative energy technology and emerging energy systems becoming more economically viable, Cooltec has begun taking important steps in designing units that will integrate with future power production and energy systems. In the current plan, the thermal fluid powered unit integrated with thermal solar power panels could be commercially ready for market by mid-2005. In the US, Cooltec chillers will be distributed primarily to the light commercial HVAC market. The Company's products are sold through a network of both stocking and non-stocking distributors to the small commercial market and high-end residential applications requiring 5 to 80 tons of cooling capacity. The initial distribution targets for the TSP products are the Southwest and Northeast US, followed by the Midwest. Each region was chosen because of a suitable climate, a supportive utility environment, a favorable differential between gas and electric rates, and an established service infrastructure.

International growth is a key element of Cooltec's strategy. The Company has already penetrated the Mexican market, and has standard units being tested or to be tested in conjunction with local utilities in France, Korea and Japan. When export sales are fully developed, they will represent more than 50% of the chiller sales.

Nationally and internationally the sales effort will be built on a foundation of distributors, manufacturer's representatives, and trading companies, with the US supported by a national account team to handle assisted living, fast food, and convenience stores.

8.3 Engineering/Technical: Thermal fluid cooling units integrate with some of the latest alternative energy technologies, primarily thermal solar systems and fuel cells. With this grant, the Company has developed the thermal fluid activated proof-of-concept unit demonstrating solar thermal feasibility. Cooltec's proprietary control logic for dual fuel sources supplements the thermal solar heat with natural gas or propane, providing full capacity cooling when the solar heat fraction is reduced by weather or darkness. Concept testing has been successfully completed.

Integrated products for distributed generation will take advantage of the Company's competitive advantages in designing, building, and marketing thermally activated absorption systems. In the near future, the Cooltec5^{tm} will have the capability to be fired by heat from turbines, microturbines, and industrial processes. Proprietary control logic and Cooltec's unique design permit dual fuel sources, using natural gas or propane to supplement the heat stream when necessary to maintain full cooling capacity. Second generation microturbine-integrated units are in beta testing and will be commercially available by June 2004.

8.4 Legal/Contractual: Cooltec's product development began with research performed by the Ohio State University and Battelle Laboratories. The project was funded with assistance from the Gas Research Institute (GRI), the Department of Energy (DOE), and a utility consortium headed by Columbia Gas of Ohio. Through technology integration, Cooltec developed proprietary heat exchangers for enhanced performance with support of GRI, Southern California Gas Company (SOCAL), and Keyspan Energy (Brooklyn Union). Primary technology for the Cooltec products are covered by licenses with Ohio State and GRI. These license agreements cover the technology for absorption chillers, heat pumps, and process chillers smaller than 100RT and apply to all products currently planned for development by Cooltec. In addition, provisional patents including application under the Patent Cooperating Treaty have been applied for in two general areas covering heat transfer control devices and dual fired absorption or heat pump controls.

8.5 Quality/Production/Readiness: Cooling Technologies is directly responsible for product development, marketing, distribution, and cooling product warranty. Partnerships with component suppliers and contract manufacturers support the manufacturing process with the Company performing final assembly and inspection. This business structure allows the Company to concentrate on the successful commercialization of its products, use the best manufacturing resources available, save significant investment in capital equipment by utilizing incremental manufacturing capacity, and maintain maximum development and production flexibility. The modifications required to convert a standard Cooltec chiller to the TSP configuration use comparable manufacturing techniques. By design, the TSP generator heat exchanger can be applied to the standard Cooltec generator with minor modifications and an overall geometry that fits within existing cabinetry.

8.6 Strategic: With dual fired capability, the future Coltec5/Solar absorption chiller will be able to operate at full capacity, 24 hours per day, regardless of the weather conditions, utilizing solar energy when available. This feature eliminates the need for redundant cooling systems or expensive thermal energy storage equipment, optimizing energy utilization. Cooling Technologies' business plan for heat activated cooling includes the demonstration and commercialization of thermal solar panel (TSP) powered air conditioning.

8.7 Public Benefit/Cost: For the stakeholders and the public, the benefits and returns are substantial. Replacing 1 kW of electric cooling demand with every installed ton of heat activated cooling, can eliminate or reduce demand charges. The cooling product uses ammonia as the refrigerant, eliminating the potential harmful ozone layer effects attributed to standard CFC and HCFC refrigerant air conditioning systems. This air cooled system requires no costly water towers or water treatment maintenance. The equipment maintains constant cooling capacity over the life of the system, and with a minimal number of moving parts in the sealed refrigeration system, has a projected lifespan considerably longer than conventional electric air conditioning equipment.

Table 3: Development Assessment Matrix

Stages **Gates**	1 Idea Generation	2 Technical & Market Anal.	3 Research	4 Technology Development	5 Product Development	6 Demonstration	7 Market Transformation	8 Commer- ization
Marketing								
Eng/Tech.								
Legal/Contract.								
Risk/Quality								
Strategic								
Production								
Public Benefits/Cost								

7.0 Glossary

COP:	Coefficient of Performance, Cooling Out divided by Energy In
CPC:	Compound Parabolic Concentrating (Solar Collector)
DA:	Data Acquisition (System)
EES:	Engineering Equation Solver (Software)
GAX:	Generator-Absorber Exchange Cycle
HVAC:	Heating, Ventilating and Air Conditioning
ICPC:	Integrated Compound Parabolic Concentrating (Solar Collector)
LMTD:	Log Mean Temperature Difference
NH3:	Ammonia
OD:	Outside Diameter
OSF:	Offset-Strip-Fin
RT:	Refrigerant Ton = 12,000 btu/hr
SCAGAX:	Solution Cooled Absorber, Generator Absorber Exchanger. Recuperative component in an ammonia-water absorption machine.
SS:	Strong Solution. Solution with an ammonia percentage greater than 30%.
UA:	Overall Heat Transfer Coefficient times Surface Area
WS:	Weak Solution. Solution with a large percentage of water and a low percentage of ammonia.

8.0 References

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APPENDIX I

Generator Model

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{Modeling of Externally Fired (Heat Transfer Fluid / Solar) Generator}
{OSF Used For Fin Surface}
{5RT GAX Chiller}
{M. A. Garrabrant, Cooling Technologies, Final Version January, 2003}


"Ammonia-Water Property Routines"

PROCEDURE TPQ(T,P,Q:X,H,v)
{known:  T(F), p(psia), q(quality - fraction)}
     TK:=(T-32)*5/9+273.15
     PBAR:=CONVERT(PSI,BAR)*P
     CALL NH3H2O(128,TK,PBAR,Q: TK, PBAR, X, hSI, sSI, uSI, vSI, Q)
     H:=hSI*CONVERT(KJ/KG,bTU/LBM)
     v:=vSI*CONVERT(m^3/kg,ft^3/lbm)
END
PROCEDURE TPX(T,p,x:q,h,v)
{known:  T(F), p(psia), x(concentration - fraction)}
     TK:=(T-32)*5/9+273.15
     PBAR:=CONVERT(PSI,BAR)*p
     CALL NH3H2O(123,TK,PBAR,x: TK, PBAR, X, hSI, sSI, uSI, vSI, q)
     h:=hSI*CONVERT(KJ/KG,bTU/LBM)
     v:=vSI*CONVERT(m^3/kg,ft^3/lbm)
END
PROCEDURE PXQ(P,X,Q:T,H,v)
{known:  p(psia), x(concentration - fraction), q(quality)}
     PBAR:=CONVERT(PSI,BAR)*P
     CALL NH3H2O(238, PBAR,X, Q: TK, PBAR, X, hSI, sSI, uSI, vSI, Q)
     H:=hSI*CONVERT(KJ/KG,bTU/LBM)
     T:=TK*9/5-459.67
     v:=vSI*CONVERT(m^3/kg,ft^3/lbm)
END
PROCEDURE PHX(p,h,x:T,q,v)
{known:  p(psia), h(Btu/lbm), x(concentration)}
     PBAR:=CONVERT(PSI,BAR)*p
     hsi:=h*CONVERT(Btu/lbm,kJ/kg)
     CALL NH3H2O(234, PBAR,x,hsi: TK, PBAR, X, hSI, sSI, uSI, vSI, q)
     T:=TK*9/5-459.67
     v:=vSI*CONVERT(m^3/kg,ft^3/lbm)
END
PROCEDURE TXQ(T,x,q:p,h,v)
{known:  T(F),x(concentration - fraction), q(quality - fraction)}
     TK:=(T-32)*5/9+273.15
     CALL NH3H2O(138, TK,x,q: TK, PBAR, X, hSI, sSI, uSI, vSI, q)
     p:=PBAR*CONVERT(bar,psi)
     h:=hSI*CONVERT(KJ/KG,bTU/LBM)
     v:=vSI*CONVERT(m^3/kg,ft^3/lbm)
END

"Proceedure To Calculate Minor Pressue Losses, Hydronic Side"
PROCEDURE K(AR:kc,ke)
Kc=.6597*AR^3-.9673*AR^2-0.19365*AR+0.5053
Ke=0.09259*AR^3+0.745*AR^2-1.836*AR+0.9971
END

PROCEDURE minorloss(flow,rho:Dpm)

A1=0.22                          {sqin}  "flow area 1"
```

```
A2=1                            {sqin}  "flow area,2”
A2p=5                           {sqin} “flow area 2p”
A3=0.5                          {sqin} “flow area 3”
A4=1                            {sqin} “flow area 4”
V1=Flow/(Rho*A2/144)            {ft/hr}
V1_sec=V1/3600                  {ft/sec}
V2=Flow/(Rho*A2/144)            {ft/hr}
V2_sec=V2/3600                  {ft/sec}
V2p=Flow/(Rho*A2/144)           {ft/hr}
V2p_sec=V2p/3600                {ft/sec}
V3=Flow/(Rho*A2/144)            {ft/hr}
V3_sec=V3/3600                  {ft/sec}
V4=Flow/(Rho*A2/144)            {ft/hr}
V4_sec=V4/3600                  {ft/sec}
AR12=A2/A1                      "area ratio"
AR22p=A2/A2p                    “area ratio”
AR2p3=A2p/A3                    “area ratio”
AR34=A3/A4                      “area ratio”
g=32.2                          "gravity constant"

Call k(AR12:kc12,ke12)
Call k(AR22p:kc22p,ke22p)
Call k(AR2p3:kc2p3,ke2p3)
Call k(AR34:kc34,ke34)

"Calculate Expansion and Contraction Losses"

dPe12=((Ke12*rho*V1_sec^2)/(2*g*144))*27.673        {in water}
dPe22p=((Ke12*rho*V2_sec^2)/(2*g*144))*27.673       {in water}
dPc2p3=((Kc22p*rho*V2_sec^2)/(2*g*144))*27.673      {in water}
dPe34=((Ke34*rho*V3_sec^2)/(2*g*144))*27.673        {in water}
dPc34=((Kc34*rho*V3_sec^2)/(2*g*144))*27.673        {in water}
dPe2p3=((Ke22p*rho*V2_sec^2)/(2*g*144))*27.673      {in water}
dPc22p=((Kc12*rho*V2_sec^2)/(2*g*144))*27.673       {in water}
dPc12=((Kc12*rho*V1_sec^2)/(2*g*144))*27.673        {in water}

Dpm=dPe12+dPe22p+dPc2p3+dPe34+dPc12+dPc22p+dPe2p3+dPc34
END

"Model Inputs"

{datafile=1
m_htf_min=19.5                  {lbm/min} "mass flow rate hydronic fluid, given"
m_ws_min=2.56                   {lbm/min} "mass flow rate weak solution, given"
Thtf_in=421                     {deg F} "hydronic inlet temperature, given"
Tgen_bot=335                    {deg F} "generator bottom temperature, given"
Tws=260                         {deg F} "weak solution exit temperature"
P_gen=279                       {psia} "high side pressure, given"
UA_exp=600                      {UA calculated from data, for comparison purposes}
Q_exp=60000                     {Energy In calculated from data, for comparison purposes}
}
m_htf=m_htf_min*60                      {lbm/hr} "convert from lbm/min to lbm/hr"
m_ws=m_ws_min*60                    {lbm/hr} "convert from lbm/min to lbm/hr"

"Heat Transfer Fluid Properties"

T_avg=(Thtf_in+Thtf_out)/2      {deg F}
T=T_avg                         {deg F} "average hydronic temperature for property calcs"
```

```
Cp=0.00048*T+0.415                {btu/lbm F} "Specific Heat"
rho=55.31167-0.01415*T      {lbm/cuft} "density"
k=0.0785-0.000023*T               {btu/hr ft F} "conductivity"
mu=33.909-0.2035*T+0.000446*T^2-0.00000034*T^3       {lb/hr ft} "viscosity"
Pr=Cp*mu/k                                                       "Prandtl number"

"Calculate NH3 Side Properties"

CALL TPQ(Tgen_bot,P_gen,q_ws:x_ws,h_ws,v_ws)
q_ws=0.0
CALL TXQ(T_gax,x_gax,q_gax:p_gen,h_gax,v_gax)
q_gax=0.0
CALL TPQ(T_vap,P_gen,q_vap:x_vap,h_vap,v_vap)
q_vap=1.0
T_vap=T_gax
CALL TPX(Tws,p_gen,x_ws:q_wsex,h_wsex,v_wsex)

"Generator Shell Dims"

OD_gen=6/12              {ft}       "outer diameter of generator shell"
CIR_gen=PI*OD_gen           {ft}       "circumference of outer generator shell"
CIR_fin=PI*(OD_gen+h)  {ft}       "avg. circumference of OSF fin"

"Enter Overall HX Dimensions and Coolant State Points"

LL=25/12                    {ft}       "hx length"
W=CIR_gen                   {ft}       "hx width, surface area"
W_fin=CIR_fin               {ft}      "hx width, centerline fin"

"Enter OSF Dimensions and Conductivity"

pitch_in=16                 {fin pitch, fins per inch}
h_in=0.1                    {in}   "fin hieght"
l_in=0.1                    {in}   "lance length"
tt_in=0.008                 {in}   "OSF material thickness"
k_osf=28                    {Btu/hr ft F, Thermal Conductivity, alum=128,
copper=225,steel=28,SS=10}

"Convert OSF dimensions to feet and calculate dimensionless parameters"

pitch=pitch_in*12               {convert pitch to ft}
h=h_in/12                       {convert to ft}
l=l_in/12                   {convert to ft}
tt=tt_in/12                 {convert to ft}
s=(1-pitch*tt)/(pitch-1)    {ft}
alpha=s/h
delta=tt/l
gamma=tt/s

"Calculate Hydraulic Diameter and Total Flow Cross Sectional Area"

Dh=(4*s*h*l)/(2*(s*l+h*l+tt*l)+tt*s)  {ft}{Manglik & Bergles, (Compact Heat Exchangers, 1990)}
Ac=(W_fin*h)-(h*tt*pitch*W_fin)        {sqft} "Cross Sectional Flow Area, OSF"

"Calculate  Reynolds Number"

V_hr=m_htf/(rho*Ac)                            {Velocity, ft/hr}
V_sec=V_hr/3600                                {Velocity, ft/sec}
```

```
Re=rho*V_hr*Dh/mu                                {Reynolds Number, dimensionless}

"Calculate Heat Transfer Coefficient, j = dimensionless Colburn Factor"
{Manglik & Bergles, (Compact Heat Exchangers, 1990)}
{Single Phase Flow, Based on Experimental Data Collected Using Air}

FF1 = 0.5                              {Prandtl number correction for liquids}
j=0.6522*Re^(-.5403)*alpha^(-.1541)*delta^(0.1499)*gamma^(-.0678)*C1*FF1
C1=(1+0.00005269*Re^(1.340)*alpha^(0.504)*delta^(0.456)*gamma^(-1.055))^0.1
j=Nuu/(Re*Pr^(1/3))                  {Nu = dimensionless Nusselt Number}
ho=Nuu*k/Dh                            {heat transfer coefficient, btu/hr sqft F}

"Calculate Fin Efficiency and Area"

Af=(4*(l*h+tt*h)+2*l*s)*(W_fin/(2*s))*(LL/l)    {sqft}  "total finned area"
Ab=2*s*l*(W_fin/(2*s))*(LL/l)                    {sqft}  "total un-fined area"
Ao=Af+Ab                                               {sqft} "total heat transfer area"

eta_o=1-(Af/Ao)*(1-eta_f)                        {fin efficiency calculation}
eta_f=tanh(phi)/phi
phi=(h/2)*((2*ho)/(k_osf*tt))^0.5

"Calculate hA OSF Side"

hA_htf=(eta_o*ho*Ao)*FF1                               {btu/hr F}

"Calculate Ammonia Side hA"

hnh3 = 800                             {btu/hr sqft F}
Anh3 = CIR_gen * LL * WET              {sqft} "total nh3 side surface area"
hA_nh3 = hnh3 * Anh3                   {btu/hr F}
WET = 0.57                             "wetted wall percentage, model input"

"Calculate UA"

UA = (1/hA_htf + 1/hA_nh3)^(-1)                        {btu/hr F} "overall heat exchanger UA"
UAcomp=UA/UA_exp                                       "compare measured to calculated"

"Energy, Mass and Species Balance"

Q=m_htf*Cp*(Thtf_in-Thtf_out)                          {btu/hr} "energy transferred from hydronic"
Q=m_ws*h_wsex+m_vap*h_vap-m_gax*h_gax       {btu/hr} "energy received by ammonia-
water"
m_gax*x_gax=m_ws*x_ws+m_vap*x_vap              "ammonia-water side species balance"
m_gax=m_ws+m_vap                                   "ammonia-water side mass balance"
LMTD_gen=lmtd(Thtf_in,Thtf_out,T_gax,Tgen_bot)     {deg F} "generator LMTD"
Q=UA*LMTD_gen                       {btu/hr} "energy transferred, calculated from UA, LMTD"
Qcomp=Q/Q_exp                       "compare calculated to measured"

"Estimated Load"

RT=Q*COP/12000                                         {tons} "estimated cooling capacity"
COP=0.01615896*Tgen_bot-0.00002159*Tgen_bot^2-2.24705        "estimate of cycle COP
based on generator bottom temp."

"Calculate Pressure Loss through OSF"
{Manglik & Bergles, (Compact Heat Exchangers, 1990)}
{Single Phase Flow, Based on Experimental Data Collected Using Air}
```

```
{f = fanning friction factor}
f=(9.6243*Re^(-.7422)*alpha^(-.1856)*delta^(0.3053)*gamma^(-.2659))*C2
C2=(1+0.00000007669*Re^(4.429)*alpha^(0.920)*delta^(3.767)*gamma^(0.236))^0.1

FF2=1.0
dP=((f*4*(1/12)*rho*V_sec^2)/(2*32.2*Dh*144))*FF2          {pressure loss (psi), through 1 inch
of fin}
dP_h2operft=dP*27.7*12                                     {pressure loss (inches H2O), per ft
of flow length}
dP_tosf=dP_h2operft*LL                                     {pressure loss osf, inches water}

"Calculate Minor Losses and Total Pressure Drop"
CALL minorloss(m_htf,rho:Dpm)                              {in water} "minor loss
dP_total = dPm + dP_tosf                                   {in water} "total calcualted
hydronic pressure loss"

"END OF MODEL"
```

APPENDIX II

Experimental Test Data

Table 1: Alpha, Generator at Normal Elevation

File		1	2	3	4	5	6	7	8	9	10	11
Date		9/24/02	9/24/02	9/24/02	9/24/02	9/24/02	9/24/02	9/26/02	9/30/02	9/30/02	10/1/02	10/1/02
CW avg IN	F	56.3	55.4	55.5	55.8	56.0	56.0	55.7	55.7	55.9	55.8	55.8
CW avg OUT	F	50.8	49.3	49.6	49.4	50.0	49.8	49.3	48.6	49.1	47.8	46.8
Load, RT	tons	2.8	3.1	3.0	3.2	3.0	3.1	3.2	3.6	3.4	4.0	4.5
Energy Input	btu/hr	52808	51559	56349	60798	54119	58385	60751	62751	59639	67454	69441
Cycle COP		0.63	0.71	0.63	0.64	0.66	0.64	0.64	0.69	0.68	0.71	0.78
Gen LMTD	F	101	88	101	109	98	109	102	112	99	115	125
Gen UA	btu/hr-F	522	588	558	558	552	535	593	562	601	589	555
CW flow rate	lbm/min	100.4	100.2	100.3	100.2	100.3	100.2	100.3	100.6	100.5	100.4	100.2
WS flow rate	lbm/min	2.4	2.5	2.5	2.5	2.5	2.5	2.6	2.5	2.6	2.7	2.6
NH3 flow rate	lbm/min	1.15	1.28	1.23	1.36	1.27	1.32	1.43	1.51	1.44	1.68	1.92
SS flow (calc)	lbm/min	3.55	3.75	3.69	3.86	3.80	3.83	4.04	4.05	4.00	4.36	4.48
Hot Oil flow	lbm/min	15.7	20.5	20.2	20.0	15.6	15.3	17.4	18.4	19.5	18.4	18.7
Pressure HI	psia	276	265	273	287	271	279	287	278	279	289	285
Pressure LOW	psia	71	70	70	73	74	74	77	73	74	72	72
dP Oil	psi	1.3	1.7	1.6	1.6	1.4	1.3	1.5	1.6	1.7	1.6	1.6
Oil In (1)	F	415	379	409	429	396	428	402	435	421	462	489
Oil Out (10)	F	321	306	331	345	296	321	302	340	335	363	391
Gen Bot (2)	F	321	306	323	334	300	319	303	334	336	360	382
WS Out (14)	F	271	249	277	289	241	268	239	279	277	297	333
GAX (13)	F	213	201	214	220	196	211	195	217	221	236	246
Evap In (8)	F	40	40	40	41	42	42	45	42	42	42	42
Evap Out (17)	F	49	49	48	47	48	48	45	48	48	48	48
Evap S-heat	F	9	9	8	6	5	6	1	6	6	6	6
Pump In (15)	F	101	95	95	95	96	95	100	99	103	102	101
SS Rect Out (6)	F	121	113	116	117	114	115	118	117	120	121	124
VPR Rect In (16)	F	180	170	176	177	169	174	170	173	177	181	196
Cond In (12)	F	139	127	131	132	127	130	132	135	141	144	158
Cond Out (4)	F	106	108	105	111	111	110	117	113	115	117	116
HP Hex out (9)	F	74	75	72	71	71	71	63	72	71	72	66
LP RHX Out (18)	F	96	99	96	97	98	98	74	98	96	100	91
CW IN 1 BB	F	56.3	55.4	55.5	55.8	56.0	56.0	55.7	55.7	55.9	55.8	55.8
CW IN 2 BB	F	56.3	55.4	55.5	55.8	56.0	56.0	55.7	55.7	55.9	55.8	55.8
CW IN 3 BB	F	56.4	55.5	55.6	56.0	56.1	56.1	55.8	55.8	56.0	55.9	55.9
CW OUT 1 BB	F	50.8	49.3	49.7	49.4	50.1	49.8	49.3	48.6	49.2	47.9	46.8
CW OUT 2 BB	F	50.8	49.3	49.6	49.4	50.0	49.8	49.3	48.6	49.1	47.8	46.8
CW OUT 3 BB	F	50.8	49.3	49.6	49.3	50.0	49.7	49.3	48.5	49.1	47.8	46.8

Table 2: Alpha Generator, Lowered Elevation

File		12	13	14	15	16	18	19
Date		11/5/02	11/5/02	11/5/02	11/5/02	11/7/02	11/7/02	11/8/02
CW avg IN	F	56.0	56.0	55.9	55.7	56.0	55.9	55.9
CW avg OUT	F	49.0	49.0	48.4	47.7	47.9	47.5	47.3
Load, RT	tons	3.5	3.5	3.7	4.0	4.0	4.1	4.3
Energy Input	btu/hr	58646	59288	62412	63472	64982	66181	66907
Cycle COP		0.71	0.71	0.72	0.75	0.75	0.75	0.77
Gen LMTD	F	84	86	92	94	106	102	115
Gen UA	btu/hr-F	701	687	678	673	613	649	584
CW flow rate	lbm/min	97.8	100.0	100.0	99.9	99.4	99.5	99.4
WS flow rate	lbm/min							
NH3 flow rate	lbm/min							
SS flow (calc)	lbm/min							
Hot Oil flow	lbm/min	15.2	20.1	15.1	20.3	15.2	20.2	14.6
Pressure HI	psia	279	284	291	284	283	287	283
Pressure LOW	psia	73	73	72	71	71	71	70
dP Oil	psi	1.6	2.2	1.6	2.2	1.6	2.2	1.5
Oil In (1)	F	403	400	425	422	456	447	477
Oil Out (10)	F	292	316	308	335	339	358	353
Gen Bot (2)	F	322	331	337	346	356	365	367
WS Out (14)	F	264	273	273	283	294	310	312
GAX (13)	F	205	210	212	219	227	234	234
Evap In (8)	F	43	42	42	42	42	42	42
Evap Out (17)	F	48	49	49	49	50	49	48
Evap S-heat	F	6	7	7	7	8	7	7
Pump In (15)	F	98	99	99	98	101	101	99
SS Rect Out (6)	F	117	118	119	118	120	121	120
VPR Rect In (16)	F	172	176	178	177	180	182	182
Cond In (12)	F	132	135	137	137	142	144	143
Cond Out (4)	F	114	115	117	115	115	112	113
HP Hex out (9)	F	73	75	77	75	74	72	71
LP RHX Out (18)	F	101	103	106	104	102	98	99
CW IN 1 BB	F	56.0	56.0	55.9	55.7	56.0	55.9	56.0
CW IN 2 BB	F	56.0	55.9	55.8	55.6	56.0	55.8	55.9
CW IN 3 BB	F	56.1	56.0	55.9	55.7	56.0	55.9	56.0
CW OUT 1 BB	F	49.0	49.0	48.4	47.8	47.9	47.6	47.3
CW OUT 2 BB	F	49.0	49.0	48.4	47.7	47.9	47.6	47.3
CW OUT 3 BB	F	48.9	48.9	48.4	47.7	47.8	47.5	47.2

Table 3: Beta Generator, Lowered Position

		gas									
File		**31**	**32**	**33**	**34**	**35**	**36**	**37**	**38**	**39**	**40**
Date		**12/3/03**	**12/4/02**	**12/5/02**	**12/6/02**	**12/6/02**	**12/6/02**	**12/9/02**	**12/9/02**	**12/9/02**	**12/9/02**
CW avg IN	**F**	55.3	55.3	55.3	55.4	55.3	55.4	55.3	55.3	55.3	55.2
CW avg OUT	**F**	45.7	47.2	47.0	48.6	47.6	47.2	48.2	47.1	46.6	46.6
Load, RT	**tons**	4.8	4.1	4.2	3.4	3.8	4.1	3.5	4.1	4.4	4.3
Energy Input	**btu/hr**	87370	63013	63508	54386	58466	62786	55148	61302	65794	65814
Cycle COP		0.66	0.78	0.79	0.75	0.79	0.79	0.77	0.80	0.80	0.79
Gen LMTD	**F**		96.89	102.50	81.88	91.70	109.77	83.72	98.90	117.38	121.44
Gen UA	**btu/hr-F**		650.37	619.58	664.18	637.61	571.96	658.73	619.82	560.51	541.97
CW flow rate	**lbm/min**	100.5	100.6	100.5	100.6	100.5	100.6	100.3	100.3	100.3	100.3
WS flow rate	**lbm/min**	2.54	2.36	2.27	2.13	1.93	2.24	2.19	2.29	2.26	2.12
NH3 flow rate	**lbm/min**	2.03	1.72	1.76	1.44	1.60	1.72	1.49	1.70	1.83	1.81
SS flow (calc)	**lbm/min**	4.6	4.1	4.0	3.6	3.5	4.0	3.7	4.0	4.1	3.9
Hot Oil flow	**lbm/min**		15.8	17.6	15.1	14.9	14.9	20.4	20.5	20.4	26.0
Pressure HI	**psia**	282.2	286.9	285.2	276.3	278.9	282.9	269.6	276.4	281.2	279.4
Pressure LOW	**psia**	67.9	70.2	71.3	72.4	70.4	69.9	71.8	69.7	70.8	70.1
dP Oil	**psi**										
Oil In (1)	**F**		436	443	402	425	459	396	424	460	460
Oil Out (10)	**F**		325	343	299	315	344	319	341	372	391
Gen Bot (2)	**F**	371	350	358	329	343	359	331	346	364	367
WS Out (14)	**F**	260	288	296	273	282	296	278	285	318	330
GAX (13)	**F**	219	217	222	207	214	224	212	217	231	236
Evap In (8)	**F**	41	41	42	42	41	41	41	41	42	41
Evap Out (17)	**F**	46	49	49	51	50	49	50	50	50	50
Evap S-heat	**F**	5	7	8	9	9	9	8	9	9	8
Pump In (15)	**F**	102	101	102	103	100	100	101	98	100	100
SS Rect Out (6)	**F**	124	123	124	121	120	122	119	119	123	121
VPR Rect In (16)	**F**	191	190	194	180	183	190	179	180	198	192
Cond In (12)	**F**	153	150	155	144	145	151	142	142	157	152
Cond Out (4)	**F**	112	98	97	108	102	98	109	104	96	95
HP Hex out (9)	**F**	67	63	62	71	68	64	71	70	62	63
LP RHX Out (18)	**F**	92	85	83	96	92	86	97	95	83	85
CW IN 1 BB	**F**	55.3	55.3	55.3	55.4	55.3	55.4	55.2	55.2	55.3	55.2
CW IN 2 BB	**F**	55.3	55.3	55.3	55.3	55.3	55.4	55.2	55.2	55.3	55.2
CW IN 3 BB	**F**	55.3	55.3	55.3	55.4	55.3	55.4	55.3	55.3	55.3	55.2
CW OUT 1 BB	**F**	45.8	47.2	47.0	48.6	47.7	47.3	48.3	47.2	46.6	46.6
CW OUT 2 BB	**F**	45.8	47.2	47.0	48.6	47.6	47.2	48.2	47.1	46.6	46.6
CW OUT 3 BB	**F**	45.7	47.1	46.9	48.5	47.6	47.2	48.2	47.1	46.5	46.5

Table 4: Beta Generator, Normal Position

												gas
File		**41**	**42**	**43**	**44**	**45**	**46**	**47**	**49**	**50**	**51**	**52**
Date		**1213/02**	**1216/02**	**1216/02**	**1216/02**	**1216/02**	**12/17/02**	**12/17/02**	**12/17/02**	**12/17/02**	**12/18/02**	**12/18/02**
CW avg IN	**F**	55.5	55.5	55.5	55.5	55.6	55.5	55.5	55.5	55.5	55.5	55.5
CW avg OUT	**F**	49.0	48.1	47.3	48.0	47.7	47.4	46.5	46.0	46.0	46.3	46.0
Load, RT	**tons**	3.3	3.7	4.1	3.8	3.9	4.0	4.5	4.8	4.8	4.6	4.7
Energy Input	**btu/hr**	55642	61047	63453	60347	62372	64807	70243	75439	81527	70055	89523
Cycle COP		0.71	0.73	0.77	0.75	0.76	0.75	0.77	0.76	0.70	0.78	0.64
Gen LMTD	**F**	95	102	105	95	100	103	112	122	119	119	
Gen UA	**btu/hr-F**	583	597	604	633	624	629	625	617	685	588	
CW flow rate	**lbm/min**	100.45	100.08	100.07	100.20	100.19	100.10	100.15	100.01	100.08	100.02	100.07
WS flow rate	**lbm/min**	2.52	2.55	2.59	2.51	2.59	2.55	2.63	2.67	2.39	2.59	2.56
NH3 flow rate	**lbm/min**	1.4	1.6	1.8	1.6	1.7	1.7	1.9	2.0	2.0	1.9	2.0
SS flow (calc)	**lbm/min**	3.9	4.1	4.3	4.1	4.3	4.3	4.5	4.7	4.4	4.5	4.6
Hot Oil flow	**lbm/min**	15.089	15.392	14.733	20.217	20.081	20.3431	20.0924	20.0863	26.0675	15.1738	
Pressure HI psia	**psia**	273.5	279.2	277.6	270.9	280.2	285.1	287.1	293.9	288.2	283.3	285.7
Pressure LOW psia	**psia**	72.09	71.15	71.82	71.17	70.42	69.83	68.48	68.58	68.22	69.07	67.95
dP Oil	**psi**	6	6	6	10	10.5	10.5	10.5	10.5	18	6	
Oil In (1)	**F**	403	425	434	401	419	427	453	478	478	473	
Oil Out (10)	**F**	298	313	314	317	332	339	358	378	394	347	
Gen Bot (2)	**F**	311	327	331	323	336	342	359	373	380	358	373
WS Out (14)	**F**	257	268	272	269	280	286	303	321	339	299	267
GAX (13)	**F**	199	206	206	202	212	216	225	237	251	224	218
Evap In (8)	**F**	42	42	42	42	41	42	41	41	41	41	41
Evap Out (17)	**F**	52	50	50	50	50	50	50	48	49	49	48
Evap S-heat	**F**	10	9	7	8	8	9	9	8	8	8	7
Pump In (15)	**F**	103	102	100	99	101	102	99	102	102	101	100
SS Rect Out (6)	**F**	119	120	122	121	121	122	121	125	124	123	125
VPR Rect In (16)	**F**	177	179	193	190	184	189	192	196	193	194	198
Cond In (12)	**F**	140	142	153	150	146	149	150	157	155	155	158
Cond Out (4)	**F**	113	115	115	113	115	114	101	97	94	102	117
HP Hex out (9)	**F**	76	75	71	70	74	73	67	63	62	65	72
LP RHX Out (18)	**F**	103	104	100	98	103	101	91	85	83	89	101
CW IN 1 BB	**F**	55.5	55.5	55.5	55.5	55.5	55.5	55.5	55.5	55.5	55.5	55.5
CW IN 2 BB	**F**	55.5	55.5	55.5	55.5	55.5	55.5	55.5	55.5	55.4	55.5	55.5
CW IN 3 BB	**F**	55.5	55.5	55.5	55.6	55.6	55.5	55.5	55.5	55.5	55.5	55.5
CW OUT 1 BB	**F**	49.0	48.1	47.4	48.0	47.8	47.5	46.5	46.0	46.0	46.4	46.1
CW OUT 2 BB	**F**	49.0	48.1	47.4	48.0	47.7	47.5	46.5	46.0	46.0	46.4	46.0
CW OUT 3 BB	**F**	48.9	48.0	47.3	47.9	47.7	47.4	46.4	45.9	45.9	46.3	46.0

Table 5: Dual-Fired Beta Generator, Normal Elevation (Proprietary)

File		54	55	57	58	59	60	61	62	63
Date		3/21/03	3/21/03	3/24/03	3/24/03	3/25/03	3/25/03	3/25/03	3/31/03	3/31/03
CW avg IN	F	56.7	56.6	56.6	56.6	59.4	57.6	56.7	57.0	57.1
CW avg OUT	F	49.2	49.1	49.3	48.7	55.0	52.7	51.3	51.4	51.5
Load, RT	tons	3.8	3.8	3.7	4.0	2.2	2.5	2.8	2.8	2.8
Energy Input (hyd)	btu/hr	54897	62945	62943	68448	36300	36264	44570	46576	46582
Cycle COP (hyd only)		0.82	0.72	0.70	0.70	0.74	0.81	0.74	0.73	0.73
Cycle COP (gas+hyd)		0.63	0.63	0.66	0.65	0.44	0.49	0.59	0.55	0.55
Gen LMTD	F	78	94	101	108	56	55	70	71	71
Gen UA	btu/hr-F	916	763	661	677	1069	1092	805	873	873
CW flow rate	lbm/min	100.9	100.9	100.5	100.7	101.0	100.7	100.7	100.4	100.5
WS flow rate	lbm/min	2.2	2.2	1.8	2.0	1.5	1.5	1.6	1.8	1.8
NH3 flow rate	lbm/min	1.6	1.6	1.6	1.7	1.0	1.1	1.2	1.2	1.2
SS flow (calc)	lbm/min	3.8	3.8	3.4	3.7	2.5	2.6	2.8	3.1	3.1
Hot Oil flow	lbm/min	18.2	18.4	18.1	18.0	18.6	18.4	18.2	18.2	18.2
Pressure HI psia	psia	287.65	290.02	286.97	293.44	229.11	239.184	247.178	267.747	264.189
Pressure LOW psia	psia	75	74	74	73	73	76	75	75	74
Oil In (1)	F	404	428	450	460	323	337	367	383	382
Oil Out (10)	F	319	333	354	357	264	279	296	309	309
Gen Bot (2)	F	347	351	366	367	286	303	315	332	332
WS Out (14)	F	281	290	311	311	234	249	262	273	274
GAX (13)	F	216	220	232	233	183	196	203	213	213
Evap In (8)	F	44	44	44	43	43	44	43	44	44
Evap Out (17)	F	50	51	52	51	56	53	52	52	52
Evap S-heat	F	6	7	9	8	14	8	8	8	9
Pump In (15)	F	102	102	102	101	97	97	98	104	104
SS Rect Out (6)	F	123	124	126	125	112	116	118	125	125
VPR Rect In (16)	F	187	190	194	192	159	169	173	185	185
Cond In (12)	F	148	150	154	151	128	135	137	147	147
Cond Out (4)	F	117	117	102	105	104	105	106	109	109
HP Hex out (9)	F	70	72	66	67	93	77	72	72	72
LP RHX Out (18)	F	97	99	88	91	98	95	95	95	95
CW IN 1 BB	F	56.7	56.6	56.6	56.6	59.5	57.6	56.8	57.0	57.2
CW IN 2 BB	F	56.6	56.6	56.6	56.6	59.4	57.6	56.7	57.0	57.1
CW IN 3 BB	F	56.7	56.6	56.6	56.6	59.4	57.6	56.8	57.0	57.1
CW OUT 1 BB	F	49.3	49.2	49.4	48.8	55.1	52.8	51.4	51.4	51.6
CW OUT 2 BB	F	49.2	49.1	49.3	48.7	55.0	52.7	51.3	51.4	51.5
CW OUT 3 BB	F	49.2	49.1	49.2	48.7	54.9	52.7	51.2	51.3	51.4
Avg. Gas Input Rate	btu/hr	32300	32000	29000	33000	38500	38500	30789	30248	30248
Avg. Gas On Time	min	3.0	2.0	1.7	1.3	12.8	12.8	11.5	7.5	7.5
Total Gas Input	btu	16300	8708	4132	4416	23985	23985	11800	15000	15000

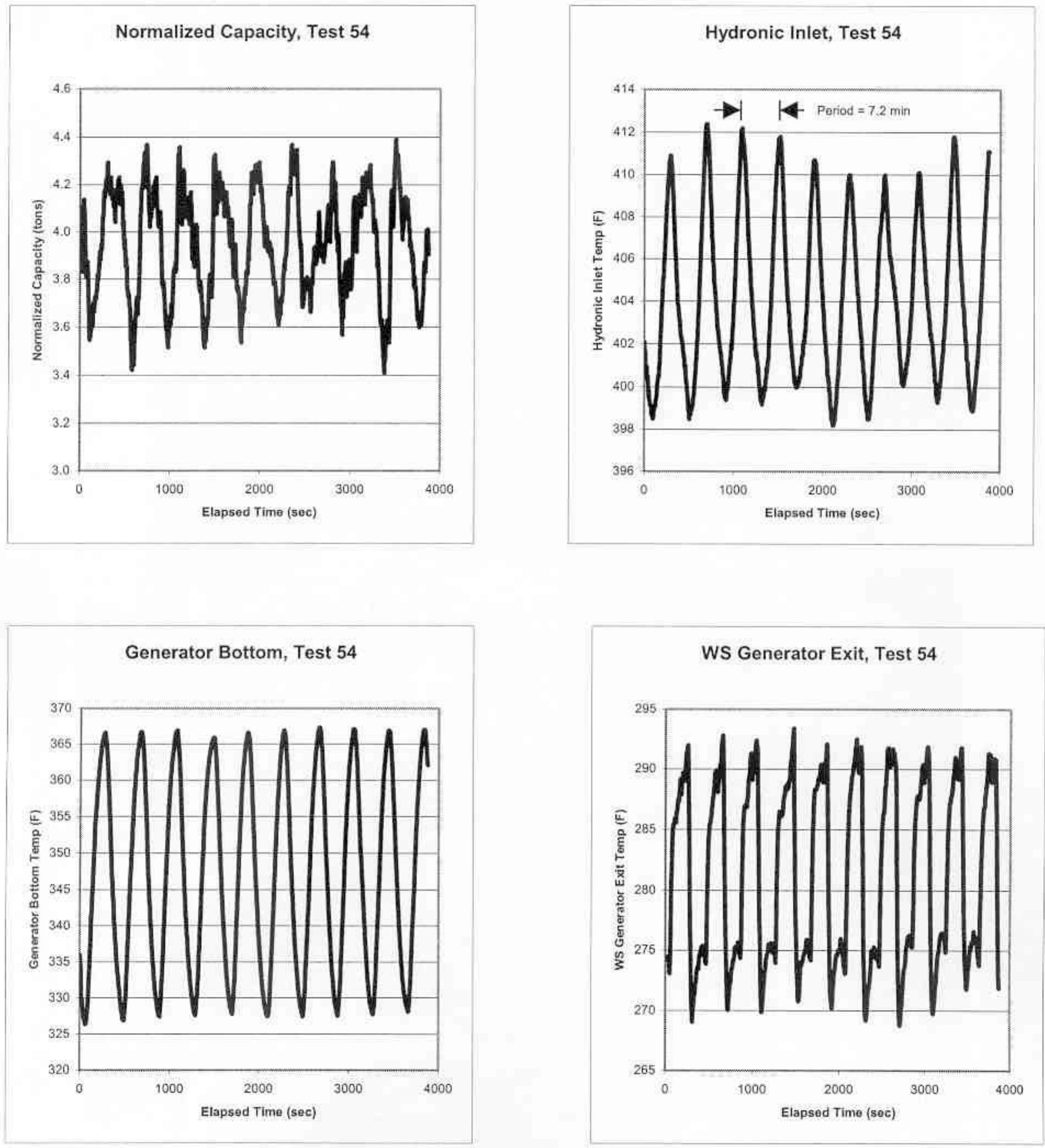


Figure 1: Parameter Plots, Test 54

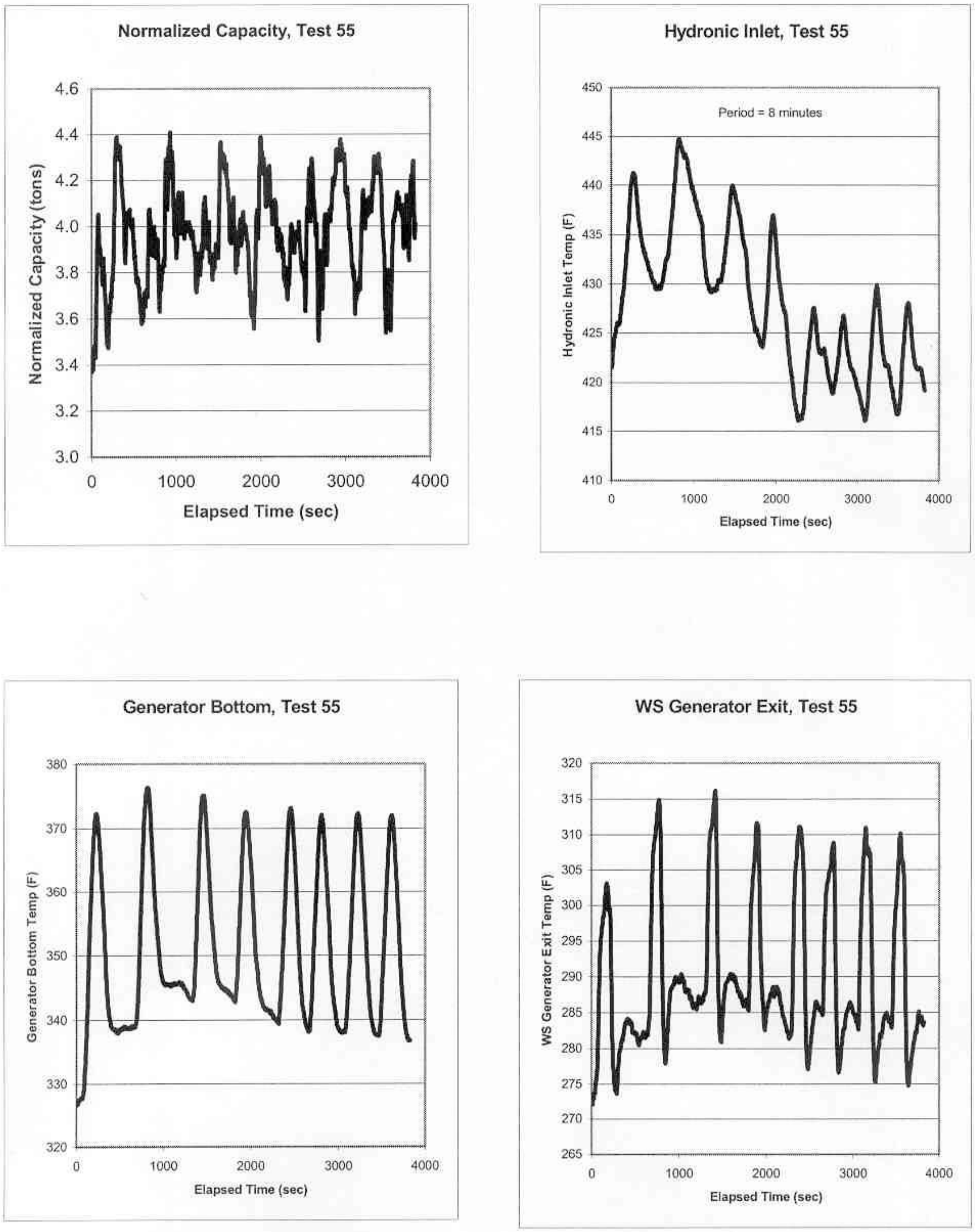


Figure 2: Parameter Plots, Test 55

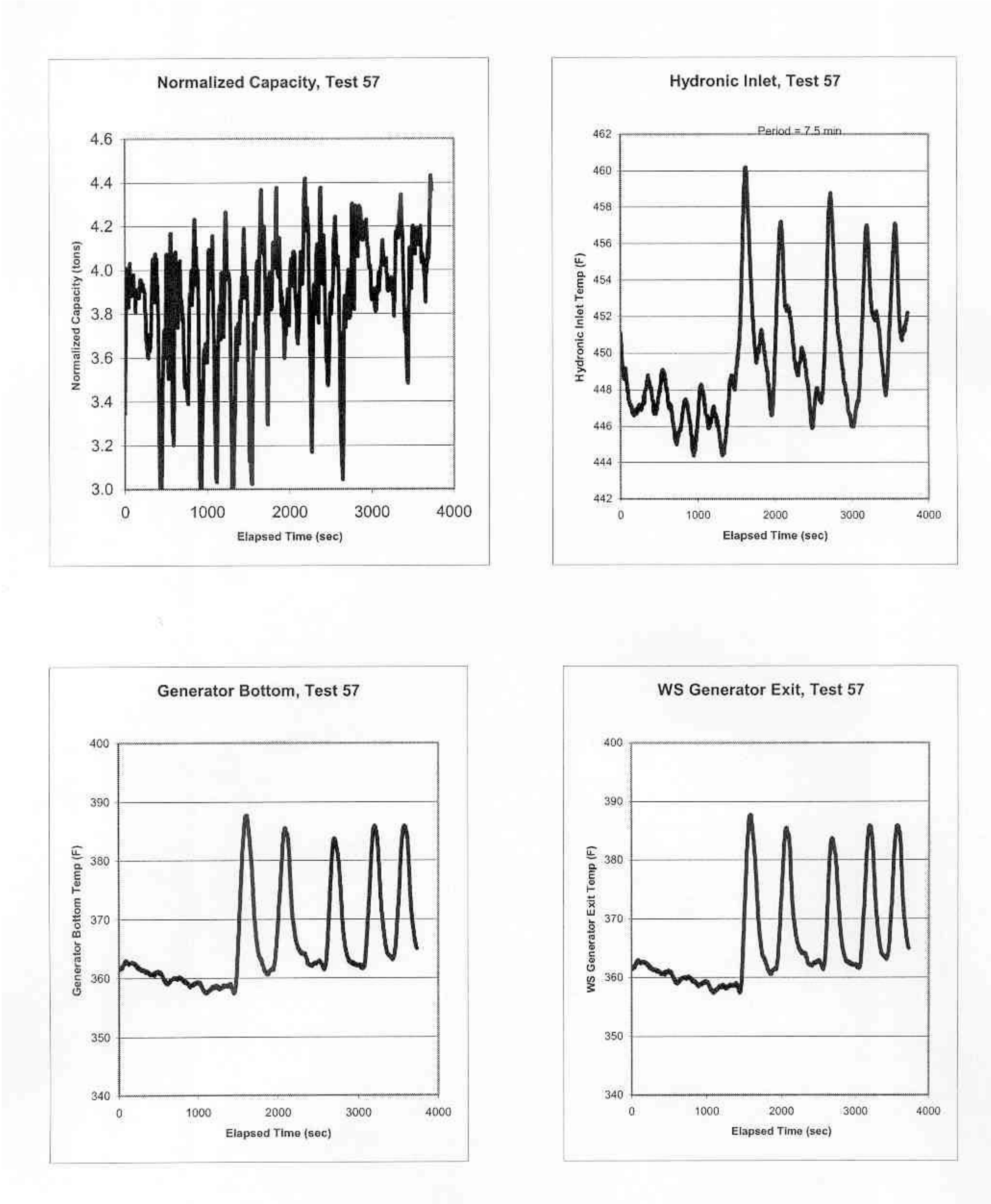


Figure 3: Parameter Plots, Test 57

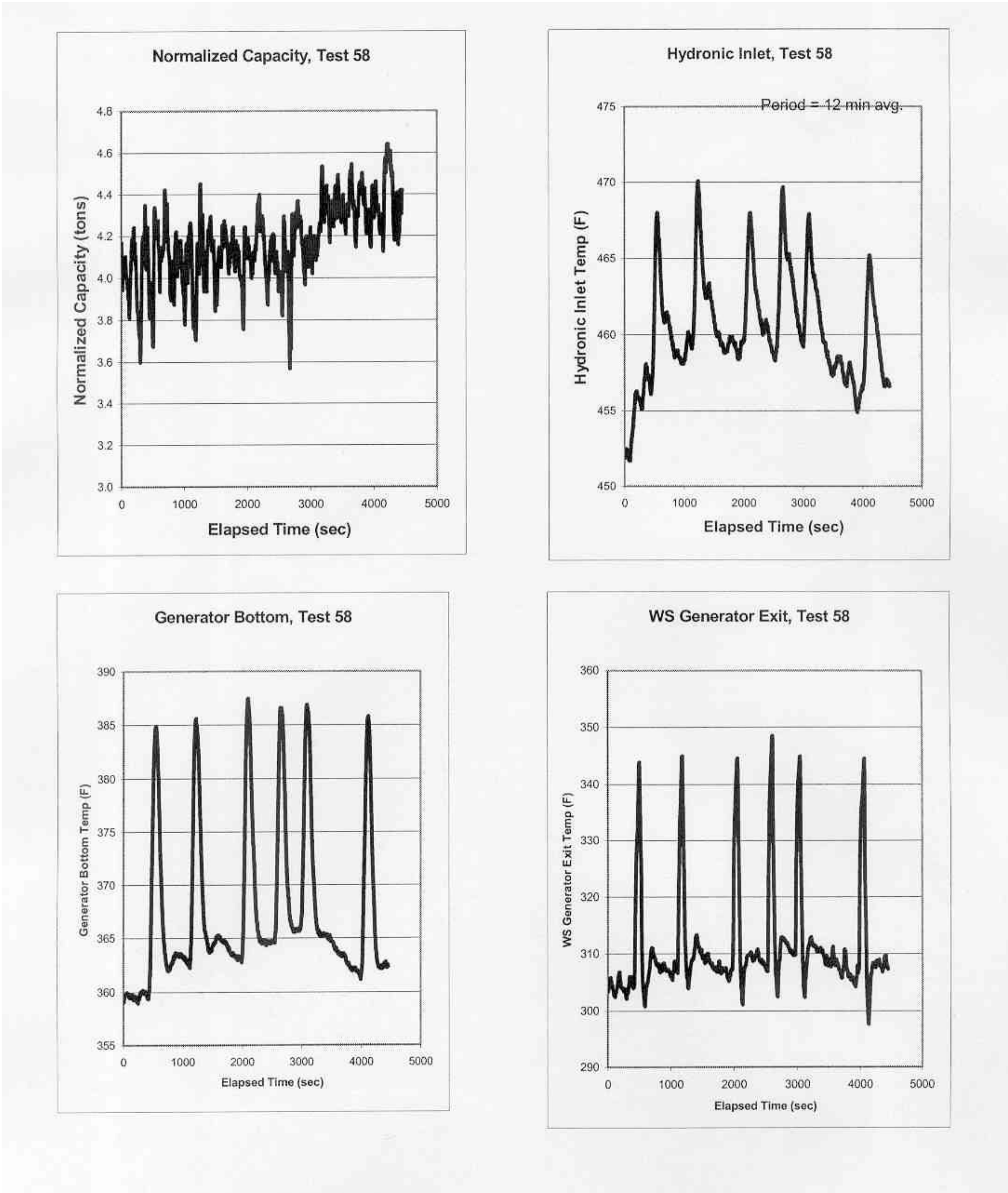


Figure 4: Parameter Plots, Test 58

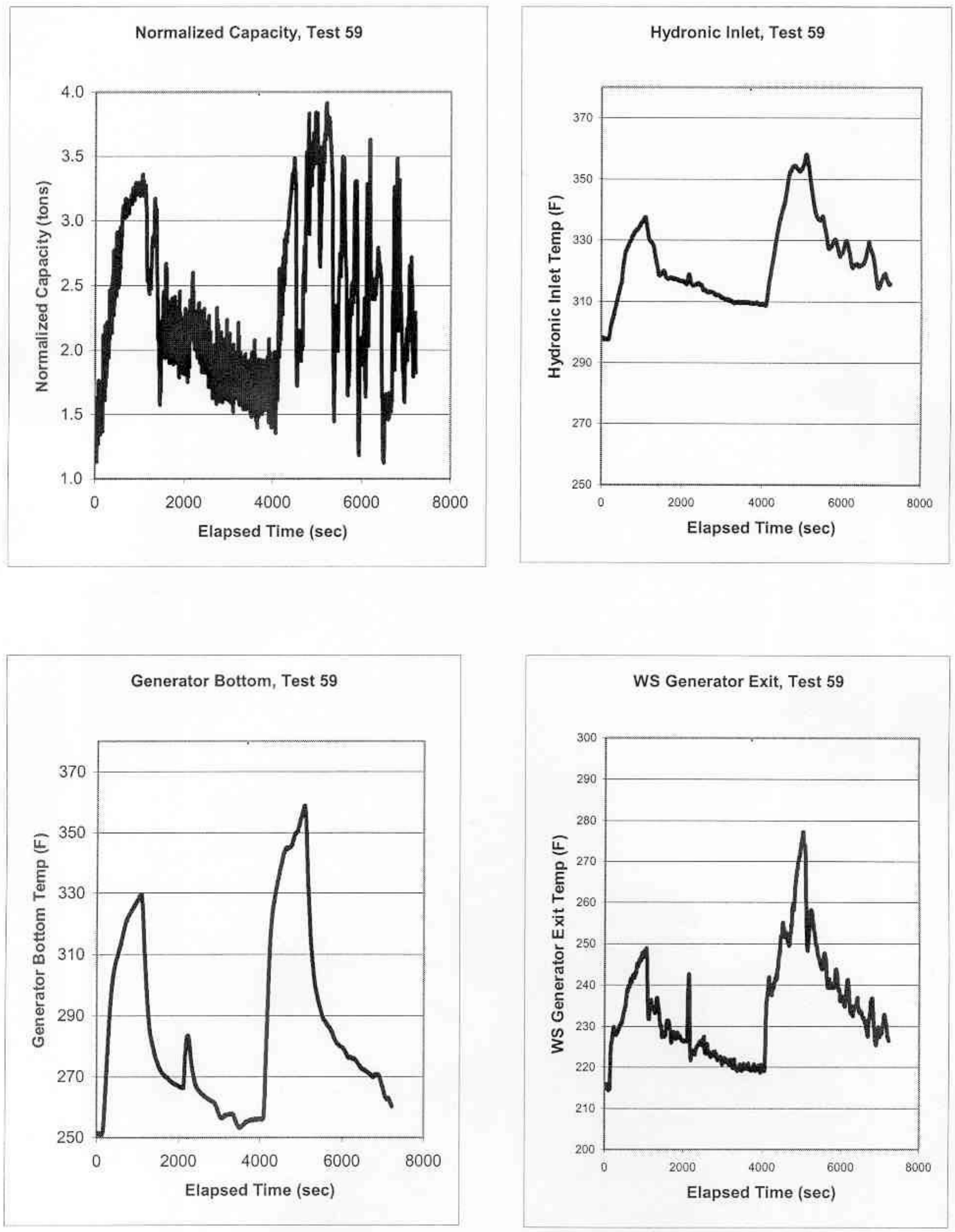


Figure 5: Parameter Plots, Test 59

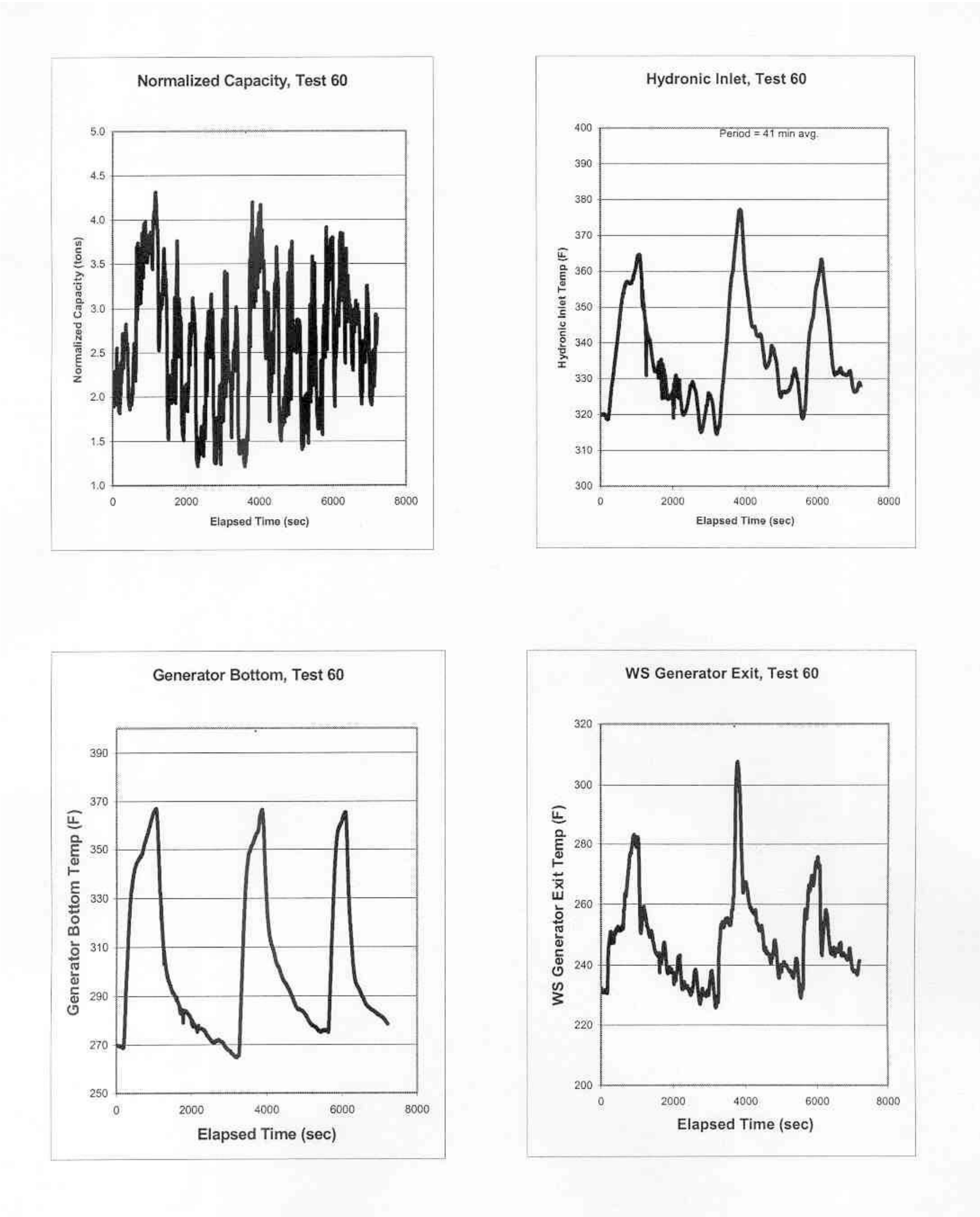


Figure 6: Parameter Plots, Test 60

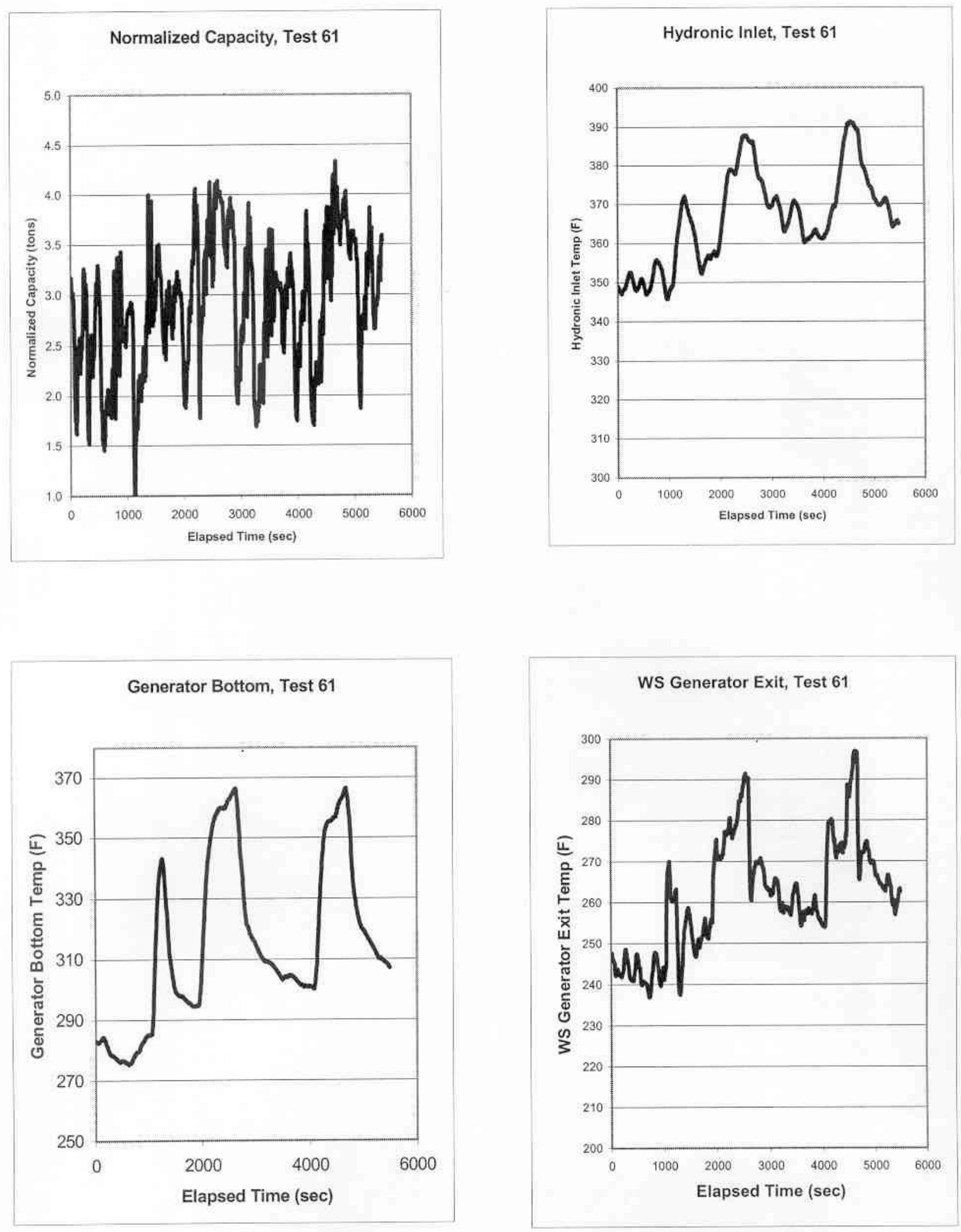


Figure 7: Parameter Plots, Test 61

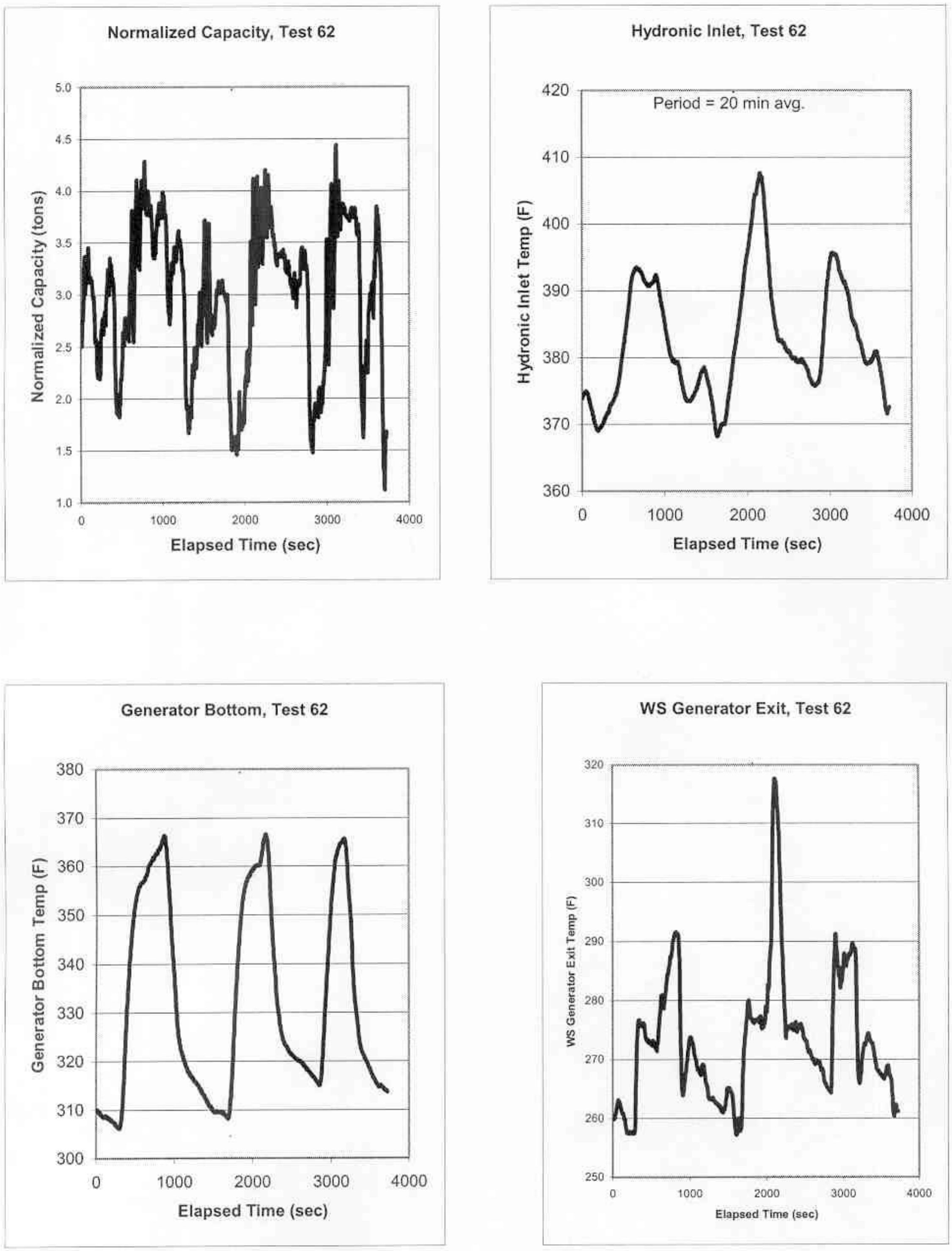


Figure 8: Parameter Plots, Test 62

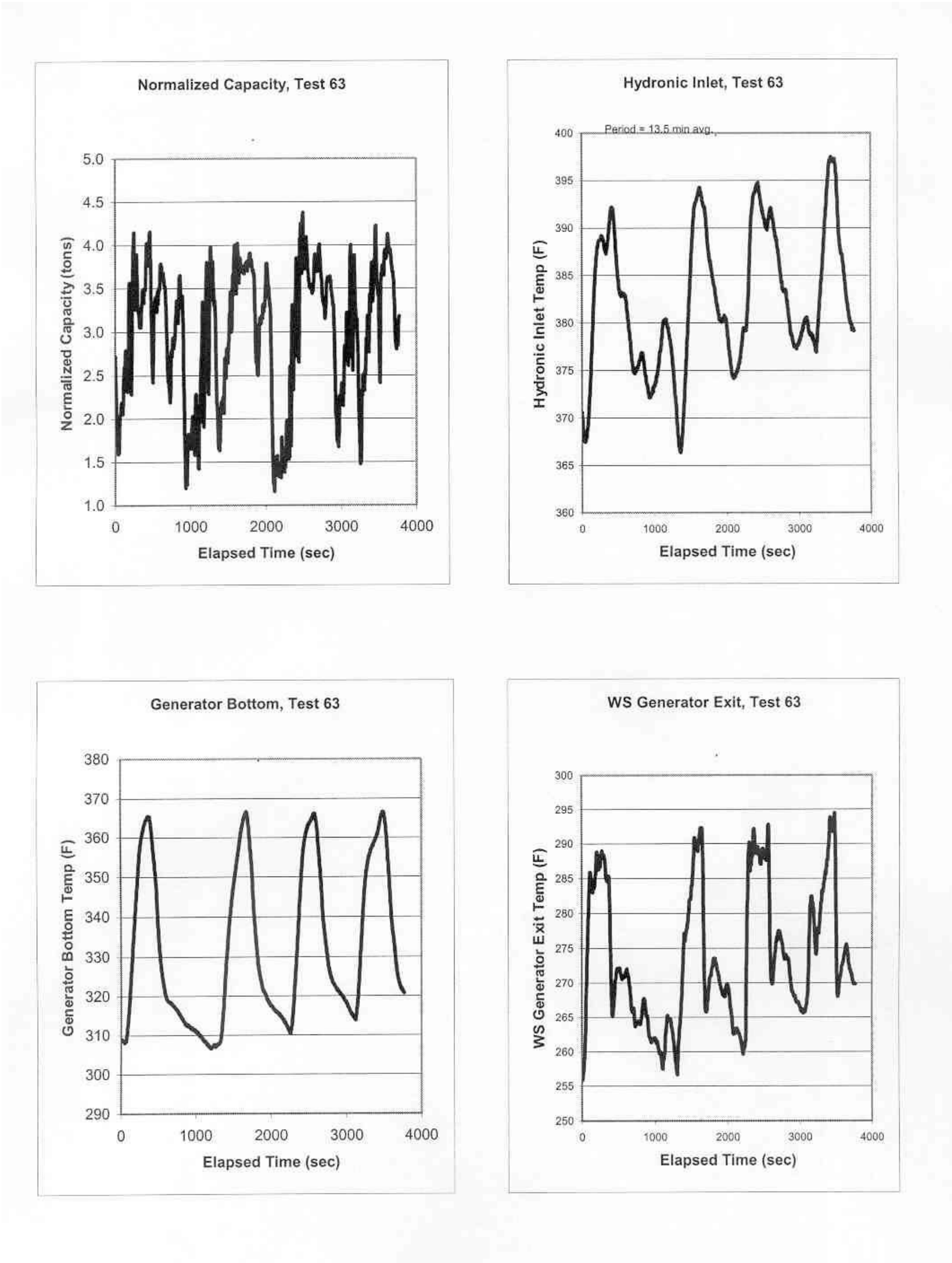


Figure 9: Parameter Plots, Test 63

APPENDIX III

Cooltec5^{TM} Gas-Fired Chiller

product data

Cooltec5™ GAS FIRED COOLING

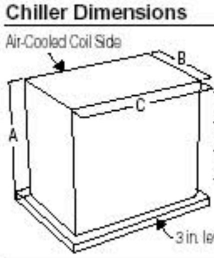
General Description

The Cooltec5™ is designed to provide comfort cooling in residential and commercial buildings, as well as process cooling for commercial and industrial applications. It is a self-contained, air-cooled, natural or propane gas-fired chiller approved for outdoor installation. The Cooling Technologies family of products utilize proprietary heat recovery technology to provide the highest operating efficiency in this size range. The Cooltec5™ is the first product of its type to earn the UL Listing and represents the future of absorption cooling.

Cooltec5 Features

- Highest GAX efficiency
- 0.68+ Coefficient of performance (COP)
- Proprietary heat exchangers
- TEV for higher efficiency at all ambient conditions
- Multi-speed condenser fan for variable loads
- Spark ignition
- Unique Lo-NOx power burner (<25 PPM)
- Closed water loop (pump optional)

Chiller Dimensions



A - Height	57 in.	1.44 m
B - Depth	34 in.	0.86 m
C - Width	55 in.	1.39 m

Performance Specifications

Description		
Delivered cooling capacity in BTU/Hour at standard ARI condition of 95°F ambient	60,100	(17.6 Kw)
Gas input at BTU/Hour	89,000	(26.1 Kw)
Condenser air flow in CFM	7,000	(200 m³/min)

Chilled Water Data

Description		
Return water temperature	55 °F	(12.8 °C)
Supply water temperature	45 °F	(7.2 °C)
Chilled water flow (GPM)	12.0 Gpm	(45.4 Lpm)
Internal pressure drop in feet of water	20.0 ft	(6.1 m)
Unit chilled water volume in gallons	2.5 gal	(9.5 Liters)
Maximum external pressure drop for unit mounted water pump	25 ft	(7.6 m)

Electrical Specifications

Description		
Electrical power requirements, 60 Hz, single phase	208/230V	
Condenser fan motor HP	1/3 (2)	(.223 KW (1))
Refrigerant circuit pump motor HP	1/2	(.372 KW)
Internal water pump HP (optional)	1/3	(.223 KW)
Total cooling operating KW consumption	1.3	
Minimum circuit ampacity (MCA)	15	
Electrical connections, utility plate diameter	7/8 in	(2.22 cm)
Size of time delay fuses (field supplied) /quantity	15 Amp/2 fuses	

Piping Connections/Physical Data

Description		
Chilled water supply/return, FPT	1 1/4 in	(3.2 cm)
Gas inlet, FPT	1/2 in	(1.27 cm)
Electrical knockouts	7/8 in	(2.22 cm)
Shipping weight	1100 lbs	(500 Kg)
Operating weight	1050 lbs	(476 Kg)
Overall height	57 in	(1.44 m)
Overall depth	34 in	(.67 m)
Overall width	55 in	(1.40 m)
Refrigerant type	R-717	

* Specifications are subject to change and periodic updates

Cooling Technologies, Inc.

Cooltec5™ GAS FIRED COOLING

Benefits

Operational Benefits

- Operating savings; gas vs. electric rates
- Reduced demand charges
- Simplified zone control (chilled water loop)
- Modulated multi-unit operation
- Low power requirement (.27 kw/ton)
- Ideal for peak-shaving applications
- Integrates to building control systems
- No capacity loss over time

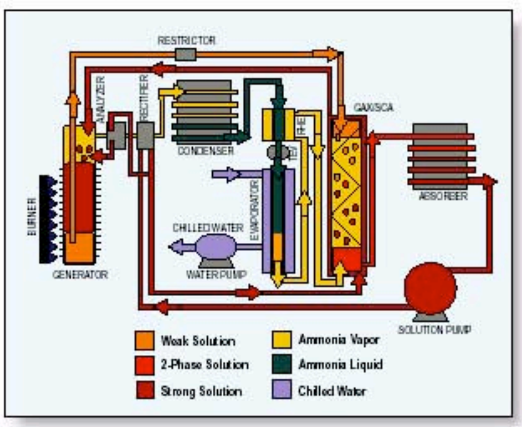
Service Benefits

- Easy service access
- High quality and reliability
- Sealed systems; few moving parts
- Reduced service costs; no compressor unit
- Reduced maintenance costs; no cooling tower

Installation Benefits

- Self contained, air-cooled package
- No cooling tower required
- Eliminates machinery room
- Minimizes roof penetrations
- No three-phase power required
- Single point utility connection
- Ideal for renovation projects

GAX Absorption Cycle



Features

Standard Features

- UL listed for safety
- UL listed for natural or propane gas
- Environmentally sensitive
  - R-717 (ammonia) refrigerant
  - No CFC's, HCFC's, or HCF's
- Thermal expansion valve (TEV) for improved efficiency at all ambient conditions
- Unique power burner system
  - Direct spark ignition
  - Low NOX (<25 ppm)
  - Variable speed combustion fan
- Condenser fans
  - Multi-speed motors match ambient and load conditions
  - High efficiency, low noise, composite fan blades
- Pre-cooler for improved performance at high and low ambient conditions
- Chiller base elevates unit above ground
  - Air circulation, moisture control
  - Four (4) side fork lift access
  - Direct installation on pad or roof stringers
- Unit mounted vibration isolation for solution pump
- Unit mounted chilled water strainer
- Unit mounted chilled water circuit air bleed
- Easy access for cleaning condenser/absorber air coil

Optional Features

- Factory installed and wired chilled water pump (1/3 HP)
- Factory installed chilled water expansion tank
- Factory installed and wired hot water boiler
- Factory mounted, piped, and wired multiple chiller arrays (Coolpak™) in 5RT increments from 10-40RT

Product Summary

- Thermally activated
- Uses reliable and plentiful natural or propane gas
- No CFC's or HCFC's for cooling
- Flexible – adapts easily to many applications
- Low operating costs
- Low maintenance costs
- Low installation costs
- Long product life
- Lowest total life cycle costs

Cooling Technologies, Inc.

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Toledo, OH 43624 | f 419.724.9007 | www.coolingtechnologies.com

APPENDIX IV

VAC Solar Panel Technical Information
Duke Solar LLC

TECHNICAL DATA

for

EVACUATED TUBE SOLAR COLLECTORS

and

SOLAR COLLECTOR MODULES

VAC 2000 / 2005 / 2008

DUKE SOLAR ENERGY, LLC

1. SPECIFICATIONS

1-1. VAC 2000 (Single Collector Tube)

Table 1

No.	Item	Specification	
		Short Tube	Long Tube
1	External dimensions of pipe [mm (inch)]	Ø 115 x L2135	Ø 115 x L3800
2	Glass thickness [mm (inch)]	3 ($^1/_8$)	
3	Weight [kg (lb)]	7.5 (16.5)	12 (26.45)
4	Net Aperture [m^2 (ft^2)]	0.22 (2.37)	0.405 (4.358)
5	Vacuum [millibar]	less than 10^{-3}	
6	Operating Pressure [bar]	less than 12	
7	Flow rate range in series connection [liter/min (USA gallon/min)]	1.5 – 10 (0.4 – 2.7)	

1-2. VAC 2005 (Panel)

Table 2

No.	Item	Specification
1	Number of collectors [piece]	5
2	External dimensions [mm (inch)]	L2250 x W708 x H192 (L90.55 x W27.87 x H7.56)
3	Mounting pitch [mm (inch)]	L1815 x W590 (L71.45 x W23.23)
4	Gross area [m^2 (ft^2)]	1.63 (17.54)
5	Aperture area [m^2 (ft^2)]	1.1 (11.84)
6	Weight [kg (lb)]	47 (103.6)
7	Heat transfer fluid capacity [liter (gallon)]	3.35 (USA 0.885)
8	Operating Pressure [bar]	less than 12
9	Flow rate range in series connection [liter/min (USA gallon/min)]	1.5 – 10 (0.4 – 2.7)

1-3. VAC 2008 (Panel)

Table 3

No.	Item	Specification
1	Number of collectors [piece]	8
2	External dimensions [mm (inch)]	L3975 x W1100 x H220 (L155.5 x W43.5 x H8.7)
3	Mounting pitch [mm (inch)]	L3400 x W1080 (L133.8 x W42.5)
4	Gross area [m^2 (ft^2)]	4.41 (47.47)
5	Net absorber area [m^2 (ft^2)]	3.24 (34.86)
6	Weight [kg (lb)]	145 (319.7)
7	Heat transfer fluid capacity [liter (USA gallon)]	9.47 (2.5)
8	Operating Pressure [bar]	less than 12
9	Flow rate range in series connection [liter/min (USA gallon/min)]	1.5 – 10 (0.4 – 2.7)

2. MATERIALS

2-1. VAC 2000

Table 4

No.	Component	Material	Specification
1	Glass tube	Borosilicat glass Antireflective coated	Expansion coefficient : 32.5 x 10^{-7} / ℃ Specific gravity : 2.23 Transmittance : 96% minimum at air mass 1.5 solar spectrum
2	Reflector	Aluminum	Silver coated
3	Heat transfer absorber tube	Stainless steel 316L (ASTM A 249)	Ø 15.88 x t1.0mm With sputtering coating
4	Sealing	Glass-to-metal connection – house-keeping seal	
5	Getter	Barium	

2-2. VAC 2005

Table 5

No.	Component	Material	Specification
1	Enclosure	Stainless steel 304	
2	Inner tube	Stainless steel (316L)	Ø 15.88 x t0.89mm With spattering coating
3	Manifold	Copper	15.88 x t0.81mm
4	Rubber packing	E.P.D.M	
5	Mounting metal	Stainless steel	
6	Insulation of absorber tube	Vacuum	
7	Insulation of manifold	Rockwool	25.0mm (1 inch) thickness

2-3. VAC 2008

Table 6

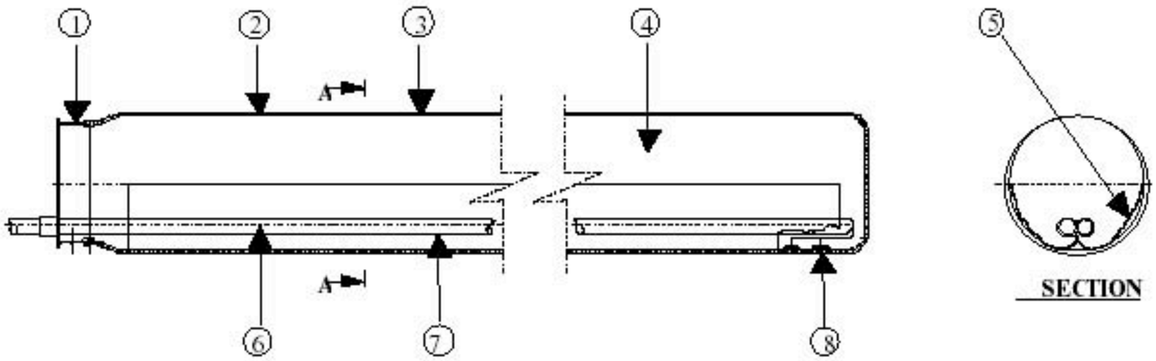
No.	Component	Material	Specification
1	Enclosure	Stainless steel 304	
2	Inner tube	Stainless steel (316L)	Ø 15.88 x t0.89mm With selective sputtered coating
3	Manifold	Copper	Ø 22.22mm
4	Rubber packing	E.P.D.M	
5	Mounting metal	Stainless steel	Zinc coated and painted
6	Insulation of absorber tube	Vacuum	
7	Insulation of manifold	Rockwool	25.0mm (1 inch) thickness

3. PERFORMANCE

Table 7

No.	Item	Performance
1	Instantaneous energy collection efficiency	See Fig. 1
2	Pressure loss	See Fig. 2,3
3	Characteristics of selective coating	2008 2005 Absorption rate ? : more than 0.95 0.97 Emittance ratio ? : less than 0.06 0.08
4	Durability of selective coating	After exposure to weathering test (50ºC to 100ºC) for 100 hours, the difference of ? and ? are less than 0.01 : No rusting observed
5	Heat resistance of selective coating	After heating at 500ºC for 100 hours in vacuum, the difference of ? and ? are less than 0.01 After heating at 450ºC for 100 hours in air, (indoor oven) the difference of ? and ? are less than 0.01
6	Durability of enclosure	No rusting observed after salt-water spray test for 96 hours.
7	Vibration resistance	No leakage and no defect observed after vibrating at 1 G.
8	Stagnation resistance	No leakage and no defect observed after exposing to the sun for one year.
9	Vacuum stability	No deterioration observed after exposing to the sun for 15 years

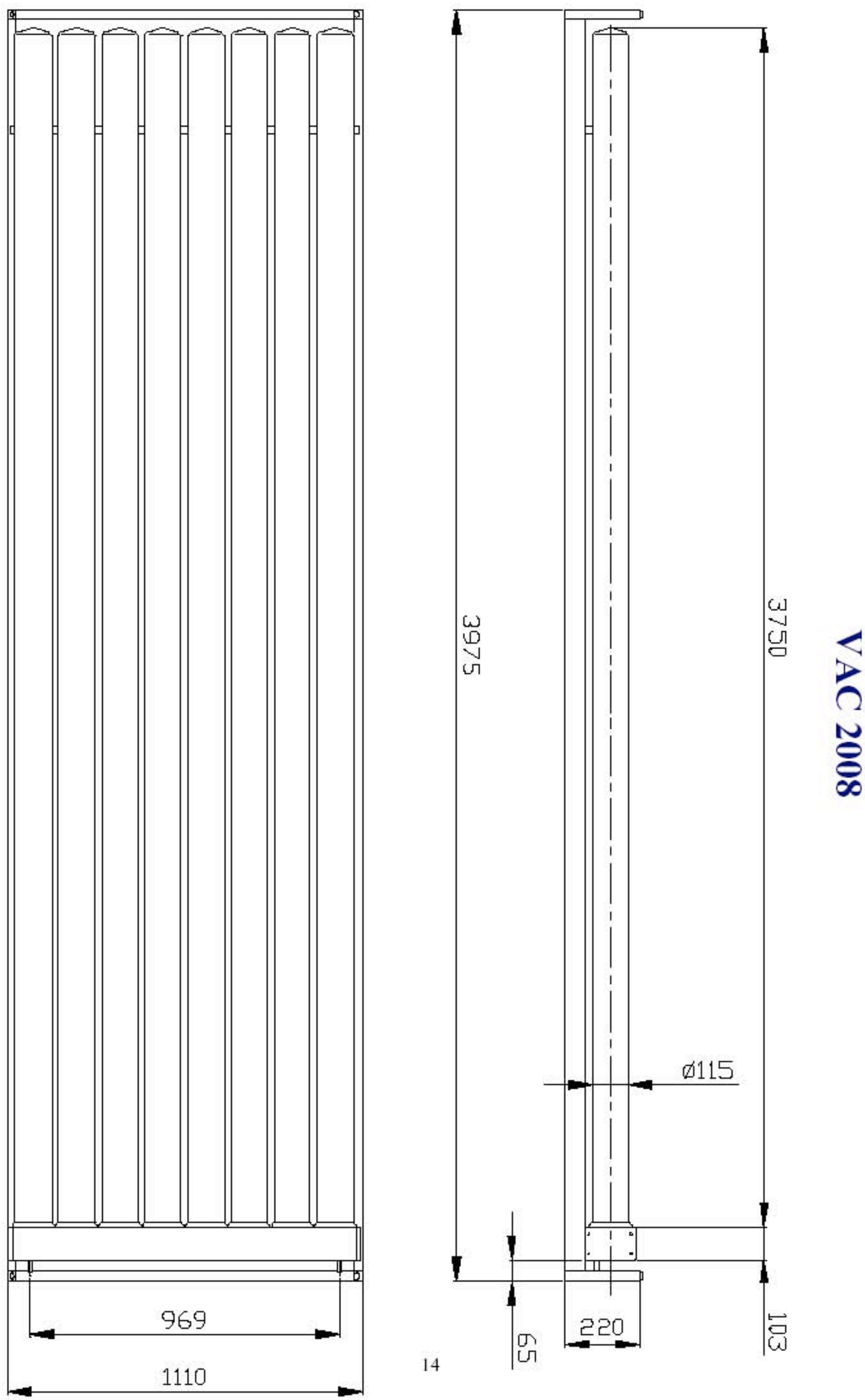
VAC 2000



1. Glass-to-metal connection
2. Glass envelop (BoSi)
3. Anti reflection coating: 97% transmission of solar radiation (A.M.S 1.5)
4. Evacuated space
5. Concentrating mirror Reflector
6. Absorbing Tubes (patented geometric concept)
7. Solar coating (CERMET)

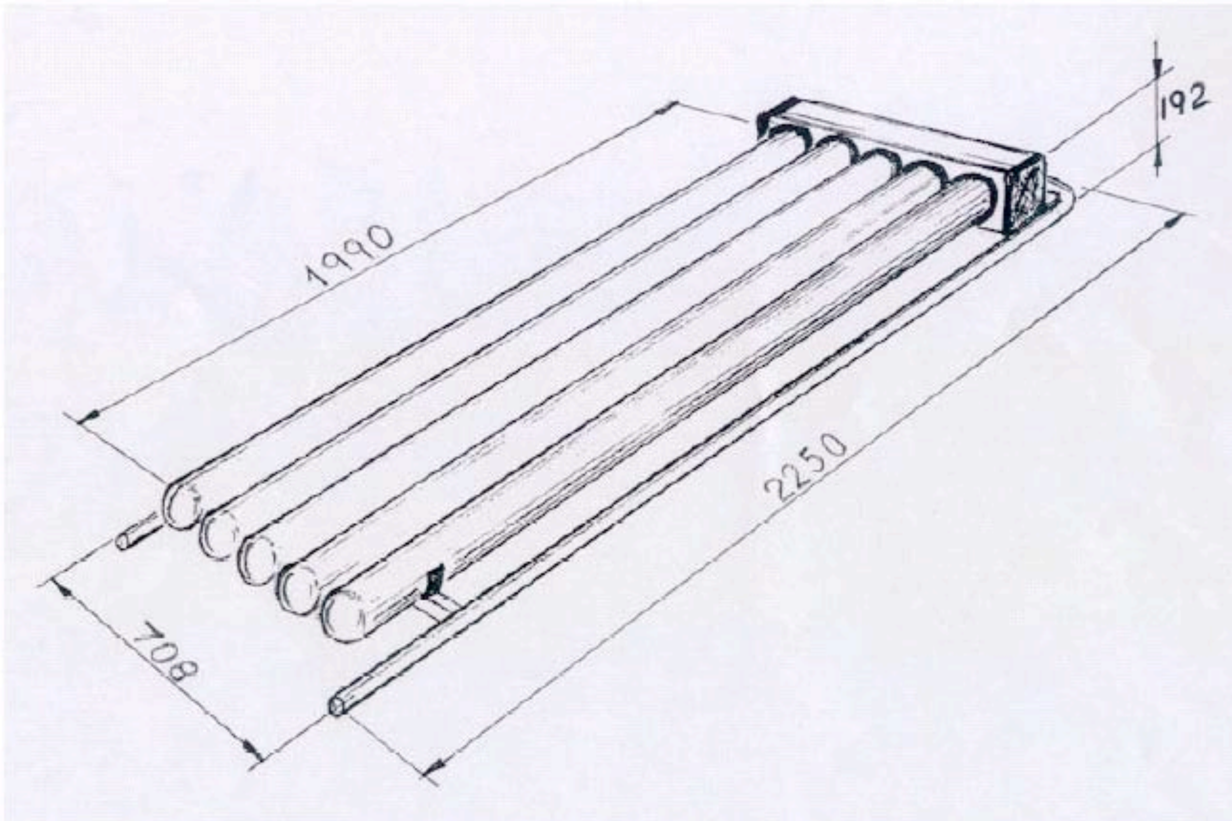
   Absorbtance (A.M.S 1.5) - 97%

   Emissivity - 0.07 at 250°c
8. Barium getters - vacuum indicators



3975
3750
VAC 2008
Ø115
969
1110
220
65
103
14

VAC-2005



APPENDIX V

High Temperature Heat Transfer Fluid Information

Table 5: High Temperature Heat Transfer Fluid Comparison

Manuf.	Product	Max Bulk (F)	Max Film (F)	Density @ oF lbm/ft3	@ oF	Thermal Cond. @ oF btu/hr ft F	@ oF	Specific Heat @ oF	@ oF	Viscosity @ oF (cSt)	@ oF	Pump-ability (F)	Type	$/gal
	Water			62.0	72	0.350	72	1.000	72	1.00	72	32		
Radco	XCELTHERM HT	660	710	56.6	400	0.065	400	0.547	400	1.05	400	-35	aromatic	$23.00
Radco	XCELTHERM XT	650	700	54.0	400	0.061	400	0.543	400	0.44	400	-70	aromatic	$16.00
Radco	XCELTHERM 550		601	46.9	482	0.068		0.689	482	0.76	482	-9	min. oil	$ 3.65
Solutia	Therminol 55		635	44.6	482	0.050	482	0.640	482	0.50	482	-18	syn. oil	$ 7.80
Solutia	Therminol 59		653	50.1	482	0.056	482	0.582	482	0.72	482	-56	syn. oil	$14.65
Solutia	Therminol 66	650	705	55.1	400	0.061	400	0.528	400	0.94	400	27	aromatic	
Royal Purple	Hytherm 707	700/500	725	43.0	400	0.121	400	0.665	400	1.60	400	?	syn oil	
Dynalene	Dynalene HT	660	716	57.0	400	0.061	400	0.530	400	0.88	400	23	aromatic	$27.00
Dow	Dowtherm MX	625	675	51.8	390	0.060	390	0.524	390	0.55	390	-10 ?	aromatic	$10.00
Dow	Syltherm 800	750	800	48.0	400	0.058	400	0.460	400	1.31	400	-40 ?	silicone	$32.00
Dow	Dowtherm T		601	44.4	482	0.057	482	0.632	482	0.73	482	14	syn. oil	$ 5.70
Dow	Dowtherm Q		675	49.4	482	0.052	482	0.556	482	0.35	482	-22	syn. org.	
Dow	Dowtherm HT		700	52.6	482	0.060	482	0.569	482	0.92	482	25	syn. oil	$24.60
Dow	Dowtherm RP		700	54.1	482	0.058	482	0.550	482	0.79	482	32	syn. oil	$25.25
Dow	Dowtherm G		750	54.1	482	0.057	482	0.565	482	0.57	482	-4	syn. oil	$25.61
Paratherm	Paratherm NF		650	50.0	400	0.070	400	0.610	400	1.20	400	-13	oil	$ 9.05
Paratherm	Paratherm HE	600	640	46.5	400	0.070	400	0.610	400	1.50	400	19	oil	$ 7.25
PetroCanada	Calflo HTF	620	650	45.4	400	0.077	400	0.600	400	1.46	400	5	oil	$23.00
PetroCanada	Calflo FG	620	650	45.0	400	0.074	400	0.600	400	1.44	400	0	oil	$22.00
PetroCanada	Petrotherm		621	44.2	482	0.075	482	0.659	482			39		$ 6.00
PetroCanada	Calflo AF		621	43.4	482	0.075	482	0.642	482	1.02	482	30	org. oil	$12.50
Sasol	Marlotherm N		644	45.2	482	0.067	482	0.668	482	0.59	482	14	syn.oil	$ 5.48
Sasol	Marlotherm FP		617	43.7	482	0.068	482	0.666	482	0.79	482	19	org. oil	$ 7.28
Chevron	HTO GD 46		601	44.0	482	0.068	482	0.650	482			18	paraf. oil	$ 5.94
Chevron	HTO GD 22		649	43.2	482	0.065	482	0.655	482			19	paraf. oil	$ 6.50

Paratherm NF® HEAT TRANSFER FLUID

Non-Fouling, Completely Non-Toxic Ⓤ PAREVE **ENGINEERING BULLETIN NF 1200**

Precise, Uniform Temperature Control™ in Closed-Loop Heat Transfer Systems

The Paratherm **NF**® heat transfer fluid is highly efficient, thermally stable and cost-effective. Completely non-toxic, it is exceptionally safe to use and is easy to dispose. Used fluid can be safely combined with spent lubricating oils and recycled locally (EPA, citation 57FR21524). The **NF** fluid is specified in a broad variety of applications, world wide. It is tough and durable with a proven record of success under demanding conditions, yet is easy and safe to handle.

Fluid Fouling

Unlike conventional heat transfer fluids, the Paratherm **NF** will not cause hard carbon formation on heated surfaces. Conventional heat transfer fluids, when severely overheated, will produce sooty carbon at the film layer. Much of this carbon immediately adheres to the heated surface and bakes on, forming a crust. As layer-upon-layer builds up, heat transfer – and in many cases flow – is impaired. Although nearly impossible to remove without scraping, sandblasting or using chlorinated solvents, the carbon *can* ultimately break loose, and large chunks of it can circulate through the system impeding flows and fouling components. Where fouling is extreme, heater tubing and electrical elements will stress and prematurely fail.

Under similar extreme overheat conditions, the **NF** fluid evolves small carbon granules. These granules remain in suspension and are easily filtered out.

Environmental Safety

The Paratherm **NF** has passed Bioassay. Rainbow trout, fresh water shrimp and Gulf shrimp were exposed to ***water-accommodated fractions*** of the **NF** fluid. No organisms died, and there were no ill effects. In the event of a release, you can use the same simple clean-up procedures employed for light lubricating oils. Once gathered, the **NF** fluid can be combined with spent lube oils and sent to the local oil recycler. There it can be converted into another useful material, preserving natural resources. The crystal-clear Paratherm **NF** contains no chlorinated hydrocarbons, aromatics, heavy metals, or sulfur or nitrogen compounds.

Fluid Toxicity

The Paratherm **NF** fluid is *completely* non-toxic. It is certified by the FDA and USDA, by Canadian Agriculture and Health & Welfare and by New Zealand MAF for use with food and pharmaceuticals. It also carries the USDA's H-1 incidental food contact rating and is certified kosher by the *Orthodox Union* (O-U), the world's premier kosher certifying agency. Do not breathe vapor mists of any fluid (see the Material Safety Data Sheet for further information on these, and other conditions).

Typical Properties*

Physical Properties

Feedstock		NF/USP Hydrotreated hydrocarbon base
Appearance		Transparent, Colorless, Bright
Taste & Odor		None
Maximum Recommended Film Temperature		650°F (343°C)
Minimal Optimal Temperature		120°F (49°C)
Flash Point (coc)	ASTM D-92	345°F (174°C)
Flash Point (pmcc)	ASTM D-93	335°F (168°C)
Fire Point (coc)	ASTM D-92	385°F (196°C)
Autoignition	ASTM D-2155	690°F (366°C)
	ASTM E659-78	691°F (367°C)
Atmospheric Boiling Point, 10% Fraction,	ASTM D-1160	650°F (343°C)
Vapor Pressure, psia @ 600°F		4.720
Coefficient of Thermal Expansion**		0.000304/°F 0.000547/°C
Average Molecular Weight	ASTM D-2502	350
Density, lb/gal @ 75°F (24°C)	ASTM D-1298	7.25
Pour Point (Crystal Point)	ASTM D-97	-45°F (-43°C)
Pumpability: Centrifugal @ 2,000 centipoise		-13°F (-25°C)
Heat of Vaporization (Calculated)		90.72 BTU/lb

Electrical and Optical Properties: Available On Request

** These are typical laboratory values, and are not guaranteed for all samples.*

***Note: Normal practice is to size the expansion tank so that it is 1/4 to 1/3 full when the system is cold, and 2/3 to 3/4 full when the system is at the maximum normal operating temperature.*

Vapor Pressure

The **NF** fluid has an extraordinarily low vapor pressure – less than 1/3 of an atmosphere at its maximum operating temperature of 600°F. This and the fluid's exceptionally low pressure drop permit the designer considerable latitude in the choice of lower-cost equipment.

Efficiency

The lower a heat transfer fluid's viscosity, the less energy will be required to pump it through the system. Paratherm **NF**'s viscosity is among the lowest of available high temperature heat transfer fluids. This means that less horsepower is needed for a given duty, and that a smaller pump and motor can be specified. And lower power consumption continues to produce savings year after year.

Water In the System

The **NF** fluid is manufactured from natural feedstocks and offers the same superb metal-coating and lubricating properties as the finest natural oils. However, any water allowed to stand in piping, components or especially expansion tanks of thermal oil systems can cause severe corrosion. Because the Paratherm **NF** is immiscible with water (and is also slightly less dense), any water can be easily drained from the system's low points, or the component's drain valves. Crack the low-point valve and allow fluid to drain into a beaker or clear water glass. If you see a phase separation (one liquid "floating" on top of the other), continue to drain until no separation is observed.

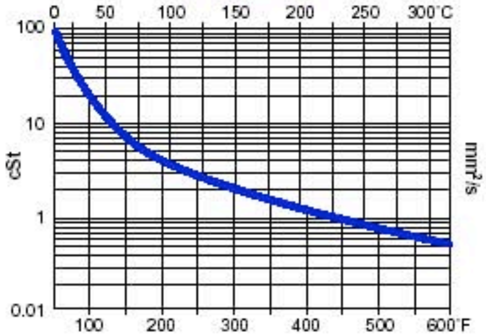
Storing Your Fluid

Containers of heat transfer fluid should be kept in non-hazardous dry areas only. Until ready for use, the container's tamper-evident safety seals *must* remain intact. Liquids should not be allowed to pool on the tops of steel drums. In the afternoon and evening when temperatures decrease, the heat transfer fluid will cool and contract slightly. A partial vacuum is created in the drum, and, if the bung's O-ring seal is not perfect, liquid standing on the top of the drum can be drawn through, contaminating the fluid. If drums must be temporarily stored outside, store them on their sides.

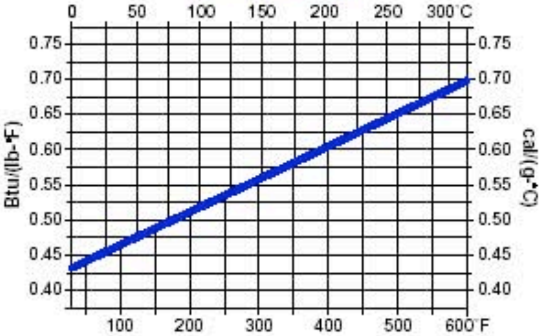
Pre-Cleaning the System

For optimal performance of both your system and its heat transfer fluid, we strongly suggest that piping, valves and other components be thoroughly cleaned before installation. Mill scale, weld spatter and slag, quench oils, protective lacquer and varnish coatings, and dust and dirt can act to degrade the fluid, and can damage pumps and valves. And lodging in restrictions, these contaminants can easily create the same low flow conditions that cause premature failure of systems and fluid.

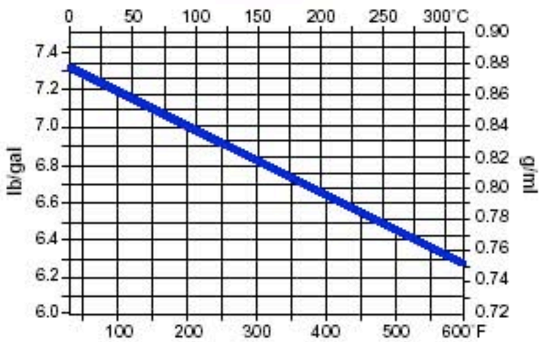
VISCOSITY



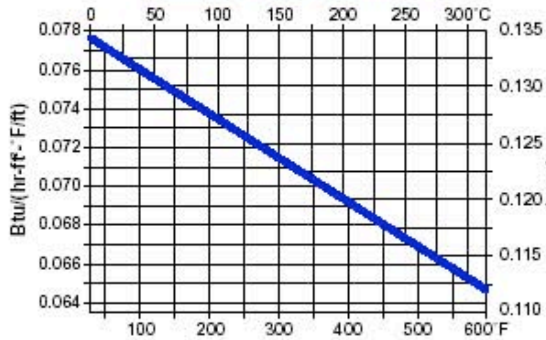
SPECIFIC HEAT



DENSITY



THERMAL CONDUCTIVITY



Inerting the System

Immediately after completing installation of the system, we suggest you purge with inert gas. Such purging can eliminate air and water vapor, and can substantially reduce corrosion. And while purging, you can leak test the system using simple soap-bubble detection methods. Finally, when you charge the system, any gas that becomes dissolved in the fluid will be inert, and fluid oxidation at start-up will be almost nil.

Charging the System

When charging the system, we suggest you fill from the bottom (a point near pump suction) using a small positive displacement pump — not the system pump. Charging from the system's low point can help reduce trapped air in the system, which will substantially reduce the entrainment of gas bubbles and resultant pump cavitation.

Fluid Disposal

Because Paratherm heat transfer fluids are produced from natural U.S. feedstocks, they are exceptionally safe to use. Easy to dispose, used Paratherm fluids can be safely combined with spent lubricating oils and recycled locally (EPA, citation 57FR21524). Paratherm strongly encourages recycling of used heat transfer fluid to conserve natural resources and to minimize the problem of liquid waste in landfills.

Fluid Analysis

Overheating, oxidation and contamination of your heat transfer fluid can significantly reduce its ability to perform. Product quality will suffer, and in severe cases considerable damage to your thermal oil system can result. Periodic analysis of your fluid can allow you to detect problems in the early stages and achieve substantial savings.

Quality Control

We thoroughly test each batch of heat transfer fluid to ensure absolute conformance to tight product specifications. Each shipment is traceable to its master batch, with test results archived at Paratherm.

Technical Assistance

Our technical expertise is available to you in the conceptual stage, during planning and design, and through system construction, start-up and operation. We want to work closely with you in the recommendation of proven thermal fluid systems, components, supplies and procedures. We also advise on system cleaning and repair, should these become necessary. And if your application calls for fluids that we are unable to provide, Paratherm will assist you with the names and phone numbers of competitors' engineers that can help.

Additional Information

Paratherm has available technical data sheets covering a variety of thermal fluid and system topics including fire prevention, system performance tracking, recommended components, draining, flushing and charging and fluid analysis, among others. We'd be pleased to forward these to you upon request.

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