Final Technical Report

Research, Development and Demonstration of Micro-CHP Systems for Residential Applications

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1 Executive Summary

ECR International and its joint venture company, Climate Energy, are at the forefront of the effort to deliver residential-scale combined heat and power (Micro-CHP) products to the USA market. Part of this substantial program is focused on the development of a new class of steam expanders that offers the potential for significantly lower costs for small-scale power generation technology. The heart of this technology is the scroll expander, a machine that has revolutionized the HVAC refrigerant compressor industry in the last 15 years.

The liquid injected cogeneration (LIC) technology is at the core of the efforts described in this report, and remains an excellent option for low cost Micro-CHP systems. ECR has demonstrated in several prototype appliances that the concept for LIC can be made into a practical product. The continuing challenge is to identify economical scroll machine designs that will meet the performance and endurance requirements needed for a long life appliance application. This report describes the numerous advances made in this endeavor by ECR International.

Several important advances are described in this report. Section 4 describes a marketing and economics study that integrates the technical performance of the LIC system with real-world climatic data and economic analysis to assess the practical impact that different factors have on the economic application of Micro-CHP in residential applications.

Advances in the development of a working scroll steam expander are discussed in Section 5. A rigorous analytical assessment of the performance of scroll expanders, including the difficult to characterize impact of pocket to pocket flank leakage, is presented in Section 5.1. This is followed with an FEA study of the thermal and pressure induced deflections that would result from the normal operation of an advanced scroll expander. Section 6 describes the different scroll expanders and test fixtures developed during this effort.

Another key technical challenge to the development of a long life LIC system is the development of a reliable and efficient steam generator. The steam generator and support equipment development is described in Section 7.

Just one year ago, ECR International announced through its joint venture company, Climate Energy, that it was introducing to the USA market a new class of Micro-CHP product using the state-of-the-art Honda MCHP gas fired internal combustion (IC) engine platform. We now have installed Climate Energy Micro-CHP systems in 20 pilot demonstration sites for the 2005/2006 heating season. This breakthrough success with IC engine based systems paves the way for future advanced steam cycle Micro-CHP systems to be introduced.

2 Project Summary

ECR International continues to be engaged in the development of combined heat and power (CHP) appliances for the residential market. ECR has developed a combined heat and power system which is a variant of the Rankine thermodynamic cycle. In this cycle work is extracted from saturated steam using a scroll expander. This particular adaptation of the Rankine cycle has been given the name of Liquid-Injected Cogeneration (LIC). This variant is particularly well suited for small scale power systems required for the residential cogeneration market.

The development of the scroll device as the expander to extract work from the saturated steam in the cycle is a major focus of ECR's development effort. The design and operation of the LIC system has been patented thru the United States Patent and Trademark Office (Patent # 6,234,400 - B1). This report presents the progress that ECR has made on the development of the LIC system.

3 Liquid Injected Cogeneration (LIC) Cycle

3.1 Background on LIC Technology

Liquid Injected Cogeneration, LIC, is a simple but unconventional, power cycle conceived by ECR International to meet the particular requirement for simultaneous generation of heat and electric power on a small scale. The simultaneous generation of heat and power is also known as cogeneration or Combined Heat and Power (CHP). LIC is based on the instantaneous heating of pressurized water which is then injected into a two phase expander to drive an AC rotary generator. Condensation of the two-phase mixture discharged by the expander is used to heat air or water for space heating, as shown in Figure 1.

Figure 1 - LIC Cycle Diagram

A feed water pump moves water from a reservoir to a direct-fired, compact, monotube heat exchanger heating the liquid water to produce saturated steam. Because the water is pressurized, the saturated steam is above the normal 212 °F boiling point found at atmospheric conditions. The working fluid exiting the heat exchanger is a two-phase flow of water and steam. The pressurized vapor and liquid mixture is injected into the

two-phase scroll expander. As the vapor is expanded it releases work that drives the output shaft that is directly coupled to a conventional, rotary electrical power generator.

The two-phase mixture exiting the expander flows into a condenser where the exhaust heat is transferred to the space heating and water heating systems of the building. Liquid condensate from the condenser flows back to the feed water pump to complete the cycle.

The liquid injected cycle has been developed to specifically avoid the safety and operational problems associated with traditional steam engine technology and to take advantage of new materials and manufacturing technology.

3.1.1 Scroll Compressor Operation

Scroll machinery technology found its first major application as a refrigerant compressor in the air conditioning industry. The concept of compressing a gas by turning one involute form, or “scroll”, against another around a common axis is nearly a century old. It wasn’t until the late 1980’s that the theory was converted into practical reality through the use of sophisticated computer-assisted design and manufacturing methods.

The introduction of scroll compressor technology into air conditioning systems has allowed for quieter, more efficient and more reliable systems. The scroll compressor is a relatively simple design. Two spiral-shaped parts (scrolls) fit inside one another. One of the spiral-shaped parts remains stationary while the other orbits around the stationary part. This orbital motion causes "continuous" crescent-shaped gas pockets to be formed, as shown in Figure 2 below.

The orbiting motion draws gas into the outer pocket and seals it as the orbiting continues. This continuous orbiting motion causes the crescent-shaped gas pocket to become smaller and smaller in volume as it nears the center of the scroll form. Once at the center, the gas pocket is fully compressed and is discharged out of a portion of the non-orbiting scroll member. The sequence of images in Figure 2 shows the compression of a pocket of gas inside the scroll compressor.

4 Prospective Economics of LIC Technology

This section provides a prospective evaluation of the economics for the application of residential, warm-air, micro-CHP heating systems in typical North American residences. The costs and values associated with assessing the economics of warm air small-scale combined heat and power systems have potentially significant variation among the different geographic regions. This report looks mainly to achieve a rational approach to and a first assessment of the comparative evaluation of the economics of this new technology with respect to traditional heating and electric power supply practices, while allowing for incorporation of important external factors such as competitive pressures in the energy supply business, environmental issues, and security concerns.

4.1 Background

The analysis in this report is developed for the patented Climate Energy warm air micro-CHP system concept shown in Figure 3. This system consists of a liquid-cooled engine-generator unit and a special warm air furnace. For the illustrated configuration, all components are located indoors. The liquid-cooled generator can be any of number of candidate technologies such as Stirling engine, internal combustion engine, steam engine, etc. The special furnace is designed to warm air with either the heat produced by the engine-generator alone or in combination with an auxiliary high performance combustion gas heat exchanger, itself having two major components (a non-condensing section and a condensing section).

For typical residential applications, the range of the electric power generation capacity of interest is 1 to 3 kW (electric), with heat production during electric power generation in the 3.5 to 30 kW (thermal) range. This assumes a thermal-to-electric conversion efficiency of 20% at the 1 kW level and 10% at the 3 kW level. Maximum heat supply capacity with operation of the auxiliary burner would be, typically, 30 kW (thermal). The warm air furnace incorporates a high-efficiency variable speed electric motor driving the main air blower to achieve low electric power consumption and improved space temperature control, especially during periods when the electric generator is the sole source of heat. Fuel utilization efficiency and space heating performance will be similar to that of a two-stage/modulating furnace with a 90%+ AFUE (Annual Fuel Utilization Efficiency per DOE test standards). The system includes the feature of auxiliary water heating by the cogeneration unit to a gas-fired storage water heater, as the output of the cogen unit alone is likely to be below the acceptable water heating capacity.

System operation is assumed to be such that, during the heating season, priority is given to the supply of heat, by the heat-producing electric generator. For the 1 kW generator the auxiliary burner would run only during periods of high heating demand. It is not necessary for the larger 3 kW electric package. The system control uses both the room thermostat and the outdoor temperature to control the operation of the system. The electric generator is operated only when there is demand for heat, either space heating or

water heating. The electric generator is assumed to be coupled to the grid through an inverter and that the unit electric generator has capability as an emergency supply of electric power to the residence during grid outages. The availability of net metering is assumed – the monthly electric bill is determined on the net difference between the electric power consumed and produced on-site over the monthly period.

Figure 3 – Warm Air Micro-CHP System with Water Heating

4.2 Economic and Energy Factors

4.2.1 Cost of Electricity and Natural Gas

Cost of electric power at the residential level ranges from about 8 to 20 cents/kWh with a national average of about 9 cents/kWh. High electric power costs lead to quicker payback for cogeneration investments and rates are generally high in the Northeast region, a prime cold-weather candidate territory for micro-CHP. The New England region of the Northeast had an average retail cost of 11.94 cents/kWhr in the year 2001 and the Mid-Atlantic region of the Northeast has an average of cost 11.45 cents/kWhr in the same year. The average for the state of New York for that year was 14.06 cents/kWhr. Retail costs generally do not include a demand charge and time-of-use is not a factor.

Retail cost of natural gas varies within the range of about \$7 to \$14 per thousand cubic feet (approximately 1,000,000 Btu). There are typically no demand or time-of-use charges associated with the cost of natural gas service.

4.2.2 Cost of Conventional Heating Appliance

The cost of a conventional warm air heating appliance to the user is composed of two main elements: the equipment and the installation. Total costs will depend on the quality level and efficiency level of the appliance. For this study, the following costs are assumed for the reference warm air heating system. These are believed to be representative of industry-wide costs. The costs reported are the retail price to the customer for the complete appliance. Installation cost for all units is estimated at \$1200.

	Price to Owner
80% AFUE*, single speed	\$800
80% AFUE, two- stage/modulating	\$1300
90% AFUE single speed	\$1400
90% AFUE two-stage/ modulating	\$1800
*Annual Flue Utilization Efficiency (per DOE test)	

The 90% AFUE with variable firing and air circulation rate is the most appropriate baseline warm air heating system for comparison to the Climate Energy warm air Micro-CHP with respect to efficiency and comfort performance. Nevertheless, cost in comparison to the lower efficiency and comfort performance option (80% AFUE, single-stage) is also of interest, as such systems dominate the current market.

4.2.3 Cost of Backup Electric Power Supply

Two cost and feature levels are associated with residential backup power supplies. These are for manual, gasoline-fueled portable generators and permanently installed automatic emergency generators fueled by natural gas or propane. Cost variation within each category is a function of quality and branding. Minimum acceptable size for a portable unit is about 2 kW and minimum size for a permanently installed unit is, in practice, about 5 kW. The permanently installed units obviously have equipment and installation costs. Portable units, for comparative purposes, appropriately include the cost

of equipment and trade-installation of a manual transfer switch. Warm air micro-CHP systems under consideration can be expected to have a generation capacity similar to the smaller portable units, with the convenience features of the typically larger permanently installed units. As an attempt to establish a baseline for comparative cost estimates, the following costs for backup electric power supplies are referenced below.

Typical Backup Electric Generator Costs

Type	Size	Includes	Equipment Cost	Installation Cost	Total Cost
Manual-Portable System	2-5 kW	Electrician installed transfer switch	\$750	\$500	\$1250
Automatic Emergency Backup System	5-10 kW	Natural Gas installed to unit	\$3,000	\$2,500	\$5,500

4.2.4 Cost of Warm Air Micro-CHP Systems

For the system shown in Figure 3, the total equipment cost is mainly the sum of the cogeneration unit and the special warm air heating furnace. This assumes that all controls are contained within either the heat-engine generator or the micro-CHP furnace. Additional costs may be incurred for a circuit isolation switch for the emergency power operation.

The selling price of the micro-CHP furnace component will be similar to, but a bit higher than, the cost of the 90% AFUE modulating furnace as it includes the variable speed high efficiency motor drive that is used in such furnaces. Additional costs are attributable to the extra heat transfer coil, fluid tank, and circulating pump. An additional cost is estimated at \$300 to the customer, for a total Micro-CHP unit cost of \$2,100 to the customer.

The cost of the heat and power producing engine/generator unit is the major, and the yet-to-be-determined, variable. This economic study has the purpose of assessing the impact of the various cost levels for the engine-generator unit (and does not deal with actual or projected costs of a specific unit).

The range of cost for the micro-CHP unit that would put the whole micro-CHP warm air system in range of interest to the consumer is seen to be roughly \$2,000 to \$4,000. The lower end \$2,000 cost is expected to represent a very attractive price-point and would

obviously be most desirable. The \$4,000 figure represents what might be considered a retail price target that would be realistic from the business perspective of a unit manufacturer in the early years of micro-CHP introduction (based on limited worldwide experience to date). For this study, which is directed at the long-term economic prospects for micro-CHP, a number of case studies are conducted at a hypothetical \$2500 cost to the consumer for the micro-CHP engine generator unit.

The installation cost of the two main components of the micro-CHP system will be higher than the \$1,200 figure mentioned above for the conventional warm air furnace. A major factor here is whether or not the engine-generator unit is installed outdoors, remote from the furnace component, or indoors and adjacent to the furnace. Electric power, gas, and control connections will be needed for an outdoor unit and an indoor unit will need to be vented. It is assumed that the two components of the system are engineered to use “plug in” type connections between the furnace and engine-generator and that the engine-generator connects only to the furnace, with all electric and gas utilities connected directly to the furnace component. Conventional site-constructed gas piping will connect the gas utility to the furnace. The engine-generator can be assumed to be connected via a small diameter flexible gas conduit between the furnace and itself. (Note the size of the gas line between the 30 kW furnace and the 1 to 2 kW engine generator need only about one-half the furnace gas line size). A preliminary estimate of the total installation cost for an indoor warm air micro-CHP system (generator and furnace) unit is \$1,700, or \$500 over that of a conventional warm air furnace. This estimate reflects costs levels that should be achievable with a highly integrated two piece system.

4.2.5 Heat Energy Usage

Total space heating energy usage is a function of building size and quality and the local weather conditions, normally characterized as “heating degree days”. Building size and quality determine both the total heat energy use as well as the sizing of the heating equipment. 100 million BTU or more of annual space heating usage, for example, is reflective of typical residential space heating requirements in the northern half of the United States. Water heating typically adds about another 20 million BTU per year. Typical sizing of a residential space heating furnace is 30 kW.

Information provided by the United States Department of Energy, Energy Information Administration, can be used to determine typical heating energy use intensity of residential buildings ([HTTP://www.eia.doe.gov/emeu/efficiency/ee_report_html.htm](http://www.eia.doe.gov/emeu/efficiency/ee_report_html.htm)).

These have been summarized in a study at Clark University.

(www.clarku.edu/departments/hero/Presentations/AAG_Poster_Hua.pdf). Mean household annual space heating energy intensity provided by these resources is

	Apt > 5 units	Apt 2-4 units	Mobile homes	Single Attached	Single Detached
Mean Heating Energy Intensity (BTU/HDD/sq. ft)	5.03	11.69	10.86	9.05	7.38

Normal design practice results in about a 100% over-sizing of the furnace capacity with respect to the actual capacity needed at the minimum expected outdoor temperature (coldest expected temperature). The maximum continuous heat demand is thus about 15 kW, with a traditional single-firing-stage furnace having about a 50% burner on time during the coldest day of the year. The common practice of over sizing of the heating capacity stems from the desire to be able to provide reasonably rapid response to large changes in thermostat set point temperature (as commonly occurs with set-back thermostats).

For a micro-CHP warm air system that provides space heat from both the engine-generator and the auxiliary burner (if present), economic analysis requires determining what fraction of the annual space heating will be provided by the engine-generator. With the assumption that the engine-generator is the priority source of heat, this computation can be made from the historic ambient temperature-duration data for any particular locality. For the economic analysis, it is reasonable to assume that there is linear relationship between the required heating supply rate and the difference between room temperature and the outdoor temperature, as shown in Figure 4. Notably, the greatest duration of time is at the milder ambient temperatures, thus favoring the greatest use of the engine-generator as the heat source.

Another consideration in determining the economics of residential micro-CHP is that, under most net metering arrangements, there would be much reduced or no value with any electric power generation in a particular month that is in excess of actual electric power monthly usage. Thus, for example, full credit can not be taken for 1,200 kWhr of production if monthly usage is only say, 1,000 kWhrs, which could be a situation in some system configurations in coldest (high heat usage) months or for systems with over-sized generators.

Figure 4 - Typical Steady-State Space Heating Demand as a Function of Outdoor Temperature (Northern US Home)

4.2.6 Electric Energy Usage

Typical total electric use in single family detached homes in the Northern US is about 10,000 kWhr per year. More likely, for warm-air micro-CHP candidate homes, which would generally be above average in size, a 12,000 kWhr per year consumption would be more representative. For northern US homes having a limited air conditioning demand, the monthly average can be assumed to be roughly one twelfth of the annual amount, about 830 kWhr/month for typical or 1,000 kWhr/month for the likely candidate homes.

4.2.7 Economic Incentives

From the customer perspective, the economics of micro-CHP systems may be influenced by any of a number of possible incentives that are derived from utility and community interests such as electric power load demand control, fuel conservation, fuel switching, pollutant emissions, etc. These can, for example, be reflected in capital cost reductions (subsidy) by the local utility, tax credits, or preferential fuel pricing. No generalizations can be made regarding such factors. The economic model applied herein includes a feature that allows for input of a capital cost reduction in order to examine the effect of such potential incentives on the economics of micro-CHP. An example of a current economic incentive is described below.

On March 27, 2001, the California Public Utilities Commission announced new incentive programs to encourage residential and commercial customers to install grid-tied renewables and clean DG resources.

The Self-Generation Incentive Program provides incentives to encourage customers to produce energy using microturbines, small gas turbines, wind turbines, photovoltaics, fuel cells, and internal combustion engines. The incentives include payments of \$1 - \$4.50/Watt, depending on the technology used, and will be funded through the end of 2007. AB 1685 of 2003 extended the program expiration date from 12/31/04 to 1/1/08, as well as providing funding of approximately \$500 million. The bill also expands some program requirements, including the definition of ultra clean and low emission DG.

The program has 3 incentive levels:

Level 1

Technologies: Photovoltaic, Fuel Cells operating on renewable fuel, Wind Turbines
Rebate: lesser of \$4.50/W or 50% of project cost
System Size: 30 kW minimum to 1.5 MW maximum*

Level 2

Technologies: Fuel Cells operating on non-renewable fuel and utilizing sufficient waste heat recovery
Rebate: lesser of \$2.50/W or 40% of project cost
System Size: No minimum, maximum up to 1.5 MW*

Levels 3-N and 3-R

Technologies: Microturbines, internal combustion engines and small gas turbines, both utilizing sufficient waste heat recovery and meeting reliability criteria

Level 3-N (nonrenewable fuel) rebate: lesser of \$1.00/W or 30% of project cost (\$1,000 for 1 kW)

Level 3-R (renewable fuel) rebate: lesser of \$1.50/W or 40% of project cost

System Size: No minimum, maximum up to 1.5 MW*

* Maximum system size is 1.5 MW, however, output capacity above the first 1.0 MW is not eligible for incentives.

Customers of PG&E, SDG&E, Edison and SoCal Gas may contact their program administrator for an application, program handbook, and additional eligibility information. PG&E, SCE, and SoCal Gas will administer the incentive program in their service territories; San Diego Regional Energy Office will administer the program in SDG&E's territory.

Incentives on the part of natural gas companies would relate to the opportunity to switch customers from oil to gas. Such incentives have been proved for traditional heating equipment. At this time there is no near-term expectation for a competitive micro-CHP technology operating on fuel oil.

4.3 Analysis Technique and Model

The economic analysis of a micro-CHP analysis has two parts. These are the capital cost summations and the determination of the equipment run times. The run time determinations for micro-CHP systems are more complex than for conventional heating systems since it is necessary to determine the relative contribution of the engine-generator and the auxiliary heater to the total annual energy usage.

4.3.1 Climatic Data

An economic analysis model for the application of a warm air micro-CHP system in a residential building has been formulated as an Excel Spreadsheet using Visual Basic for Applications (VBA) to perform the annual energy utilization simulation. The simulation program reads TMY2 ("Typical Meteorological Year") weather data files. A TMY2 is a data set of hourly values of solar radiation and meteorological elements that represent a 1-year period based on sources from the National Solar Radiation Data Base (NSRDB).

It consists of months selected from individual years and concatenated to form a complete year. TMY2 data sets are available for many USA and International sites and provide a means of characterizing the heat loads that would be incurred during a “Typical” year.

The National Renewable Energy Laboratory (<http://rredc.nrel.gov/solar/pubs/tmy2>) provides a selection of 239 TMY2 stations for the United States and its territories. The stations are National Weather Service stations that collected meteorological data for the period of 1961-1990.

More than 230 International locations are available in “Energy Plus” weather format (http://www.eere.energy.gov/buildings/energyplus/cfm/weatherdata_int.cfm#top). These files have an EPW extension (for Energy plus Weather). The original source for these data sets is either International Weather for Energy Calculations (IWEC), the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) or the National Solar Radiation Data Base (NSRDB). The International EPW data files were converted into TMY2 file format by the Energy Plus V 1-1-1 WeatherConverter program.

4.3.2 Simulation Model Inputs

Model inputs include the “heating intensity” and floor areas of the residence, equipment and installation costs of both a baseline conventional warm air heating system and a proposed warm-air micro-CHP system, as well as relevant energy costs. Typical heating intensity values for buildings are reported in the literature. Provision is made for including domestic water heating as well as space heating. From the weather data, the building heating requirement, and the energy appliance performance characteristics the model generates a profile of the heat energy demands and electrical power production as a function of the ambient temperature.

Histograms provide a graphic display of the distribution of heating degree days as a function of ambient temperature and the degree-days and electrical production on a monthly basis. (See the attached Case Study results for the input and output table of the model).

The model allows input of a temperature above which the Micro-CHP System is not allowed to run without an actual demand by the room thermostat. For example, 55 F is thought to be a good value for the ambient temperature above which the Micro-CHP would not be allowed to operate without an actual room thermostat demand. This operating logic is different from the standard operating mode of a space heating system wherein operation is always solely determined by the room thermostat. However, such improved logic is seen to be necessary to take full advantage of the Micro-CHP system and is believed to be consistent with good indoor thermal comfort control.

The model includes a provision for the Micro-CHP to supply heat for water heating. The model requires input of an estimate of the percentage of the total amount of water heating

energy supplied by the LIC system over the year when it is available for such water heating (i.e., not space heating). The total annual water heating load is typically significantly smaller than the annual space heating load.

As presently configured, the model does not put a cap on the value of the electric generation during any monthly period and thus assumes all generation would be a credit against consumption that would otherwise be supplied by the grid. (The graphical output of the model does allow the user to inspect the monthly generation profile to determine if this “excess” generation is likely to be a problem with any particular Micro-CHP system.

A projected simple payback for purchasing Micro-CHP instead of a conventional warm air heating system is computed based on the electric power savings. A major factor that affects the payback, and one for which there is a wide range of reasonable estimates, is the value of the emergency generation capability capital cost credit. Assumption of such a credit reduces the payback period. Also, the model allows for stipulation of an economic incentive credit, which can also have a major impact on the payback period.

A further refinement has been made to the simulation model, though not included in the attached case studies. The simulation model now includes the option of including both a standard natural gas cost and a reduced natural gas cost, as some gas utilities offer a special DG tariff that provides lower cost natural gas for operators of distributed generation systems. This reduced natural gas cost where available also serves to shorten payback periods.

4.4 Case Study Results

4.4.1 Introduction

As indicated in the introduction, the particular purpose of this report is to scope the economics of warm air micro-CHP systems for residential applications in the United States. Key issues of concern at this time are:

- 1) What size should the engine-generator be?
- 2) What would the allowable cost range for a reasonable payback?

Sizing is of interest mainly because the cost of the engine-generator, to a first approximation, will depend on its electric power (kilowatt) rating. A smaller, well-utilized machine would favor a quicker return, rather than a poorly utilized larger machine. So the question becomes what size gives the best electrical generation benefit with the lowest capital cost (but also meeting the minimum desirable capacity as an emergency power generator). Once the “optimum” sizing is recognized, the question of the allowable cost range follows from the desired payback range, applying what external economic factors that may be relevant, such as utility promotional credits and tax incentives.

The assignment of a value to the backup emergency generator feature of the Micro-CHP system is novel with regard to assessing the comparative economics of space heating technology options. As discussed previously, the frame of reference for this value is bounded by the costs of portable gasoline generators and permanently installed gaseous fueled units.

4.4.2 The Case Studies

To begin the economic study, the following baseline values have been assumed for key variables:

Case Study # 1 - Baseline Assumptions

Parameter	Assumption
Location	Worcester, Massachusetts (7000 Heating Degree Days)
Cost of Electric Power from Utility	20.5 cents/kWhr
Cost of Natural Gas	\$13/ million BTU
Building Type/size	Detached Single Family, 2500 square feet
Electric Capacity of LIC System	3.0 kW
Water Heating Load	15% of space heating
Comparison Conventional System Type	High Efficiency, Modulating Warm Air Furnace with Separate Gas-Fired Storage-Type Water Heater
Conventional System Cost	\$1,800 (Furnace) + \$400 (Water Htr.) = \$2,200
Capital Cost of LIC Generator Unit	\$6,500
Addition Installation Cost	\$1,000
Incremental Capital Cost of LIC System	\$5,300
Credit for Emergency Generator Feature	\$2,500

The results for this baseline case are shown in Figure 5.

Figure 5 – Case 1: Baseline Energy and Economic Model

The complete result for Case Study 1, and all other case studies, is provided on the following pages. Notable results for Case Study 1 are

Parameter	Result
% of Space Heat Provide by LIC Generator	~100 %
Annual LIC run time (full or part time)	1,540 hours
Annual Electric Generation	5,434 kWhr
Annual Electric Generation Credit	\$1054
Annual Additional Natural Gas Cost	\$59
Annual Energy Cost Saving	\$995
Payback Period (with \$2,500 Backup Gen. Value/Credit)	2.9 years
Payback Period (no Back Gen. Value/Credit)	5.4 years

Case Study #1 Baseline Results

The result for Case Study 1 indicates the baseline case parameters yield a reasonable payback period of 2.9 years. Payback without any credit for the backup generator feature is longer, but not excessive. The LIC generator unit will supply all of the space and water heating needs.

Case Study 2 simulates the economics of installing the LIC in a larger home. This home size is assumed to be 3,500 square feet, in comparison to baseline assumption of 2,500 square feet. This analysis indicates that the payback period (without the generator credit) is reduced to 3.8 years if a larger home is assumed. Thus, as could be expected, larger homes are better candidates on account of the greater energy use. This case is shown in Figure 6.

Figure 6 – Case 2: Energy and Economic Simulation for Larger Home

The impact of an incentive provided by utility or government is assessed in Case Study 3. An economic incentive has a large positive effect on payback. Providing a \$1,000 incentive toward the baseline system cost reduced the payback period from 5.4 years to 4.5 years (not including generator credit). This Case is shown in Figure 7.

Figure 7 - Case 3: Utility or Government Credit

Climate also influences payback. The results provided by Case Study 1, 4 and 5 yields the following results. The detailed outputs for Cases 4 and 5 are not shown.

Case Study	Location	Heating Degree Days	Payback (years)
4	Newark, NJ	5205	6.3
1	Worcester, MA	7089	5.4
5	Minneapolis, MN	8048	3.9

Interestingly, the colder climate does reduce the payback below that for the baseline case assumptions. Although this seems reasonable, cases that assume a smaller generator with an auxiliary heater have shown very little improvement in payback for cold climates. This can be attributed to the fact that the additional heating load occurs mostly during cold months when the engine-generator is operating essentially continuously in both climate areas, thus cannot provide an increased heat contribution to the space heating load in the colder climate.

4.5 Assessment of the Case Study Results

The case studies are intended to help identify the most important factors in the economics of warm air micro-CHP for representative cases. They represent reality only in so far as the major system components can be delivered to the market at the prices indicated. Assuming that the stated component prices to the consumer are reasonable expectations for the long term, the following general conclusions are evident:

- 1) Application of the warm-air micro-CHP system as shown in Figure 3 will be in a wide range of US climate zones with moderate heating demands and is not limited to what would be considered very cold regions. Heat supplied by a micro-CHP generator in the 3 kW electric size range can supply all of the annual demand for space heating in a typical residence.
- 2) Payback periods for a wide range of applications should generally be reasonable, say to 4 to 6 years, so long as equipment costs can be held at levels not significantly different from those assumed. Larger homes, higher electric costs and low gas costs favor quicker payback.
- 3) Assigned or perceived value for emergency backup capability and availability of incentives for utilities (or taxes) significantly impact the payback of the energy conservation feature of the system. Increased electrical rating of the electric-generator for the backup power mode of operation (over the cogen mode) affects the payback period to the extent that it justifies an increased credit for a backup power system.

- 4) An engine-generator with output of 2-3 kW, while in the cogeneration mode, is the size range of interest, assuming a 10% heat-to-electric conversion efficiency and general system configuration as shown in Figure 3.
- 5) Self-generated electric power can be upwards of 50% of the total annual electric consumption, with most or all electric loads handled during winter months, assuming a net metering connection to the grid.
- 6) Total annual gas consumption of the warm air micro-CHP system would be similar to that of a conventional mid-efficiency warm air heating system, with the typical upgrade of such a standard system to a warm-air micro-CHP system resulting in little or no additional gas costs to the owner.
- 7) Total annual gas consumption increases in comparison to installation of conventional high-efficiency warm air heating systems. (about 10% increase).
- 8) Annual operating time can be on the order of 4,000 hours or more, suggesting that 20,000 hours equipment lifetime (before overhaul) is consistent with payback periods that could extend to 5 years.

5 Scroll Expander Design and Analysis

5.1 Scroll Expander Performance Prediction Model

A key factor in the performance of any scroll machine is the amount of fluid that is allowed to flow between the flank to flank gaps. It turns out that the leakage between the tip seals and the opposing scroll is a relatively small component to the total flow from one trapped pocket to another. A representative scroll machine is shown in Figure 8.

Figure 8 - Pockets and Leakage Paths in Scroll

Labels on this image show the inlet, the trapped pockets and the leakage paths at the flank gaps. Note that depending on the crank position the inlet and the two P1 pockets may be connected or separated (as shown). Similarly, the exhaust and the P3 pocket may also be connected (as shown) or separated.

To a good approximation the pressures and trapped steam characteristics in both sets of P1, P2 and P3 pockets are the same. However, as each pair of pockets propagate through

the machine these pressures are reduced by the nominal pocket expansion and leakage to the next lower pressure pockets. They also receive steam from the adjacent higher pressure pocket.

A finite difference model has been developed that determines the performance and sizing of a scroll expander by calculating for each finite instant the amount of steam entering and leaving each pocket. Initially the pressure distribution is not known, so an estimate is made. The properties of the trapped steam pockets are assessed using a polytropic ideal gas model, with a polytropic exponent that reflects that the steam is being cooled as it is expanded. The leakage from one pocket to another is assumed to be isentropic.

Figure 9 shows a chart of the typical pressures in each pocket over one revolution. The zero degree crank angle for the first P1 pocket occurs at cutoff, where this pocket is first isolated from the inlet. The model begins with the known inlet steam pressure and assumed pressure distribution to compute the leakages, pocket steam masses, temperatures and pressures for each pocket at each finite crank angle. The pressure of P1 at the end of the first revolution must match the pressure in P2 at the beginning of the revolution. Similarly, the P2 pressure at 360 degrees must match the P3 pressure at 9 degrees. The assumed pressures are adjusted until this balanced condition is achieved.

Figure 9 – Pressure in each pocket during one crank revolution

5.1.1 Single Case Analysis Tool

The model is implemented as a Visual Basic for Applications program within Excel. The main sheet of this model is shown in Figure 10. The green cells represent input data and the blue text represents computed output data.

Figure 10 - Single Case Input/Output Sheet

Both the scroll performance and the overall cycle performance are computed by this model. The normal solution method involves specifying the scroll geometry and operating conditions and the component efficiencies. The model will solve for the resultant power produced, along with many key design parameters. The model also allows the specification of a target amount of delivered electric power and the model iteratively solves for the scroll pitch that will allow this power to be provided.

5.1.2 Series Case Analysis Tool

The sheet shown in Figure 10 represents a single case analysis. As a further enhancement, the model also allows a series of cases to be solved in one run and automatically produces charts of multiple variables plotted against the desired input variable. This series calculations worksheet is shown in Figure 11. This creates a powerful tool for the modeling and sizing of scroll machines.

Calc Series		☒ Input Power, Calc Pitch	Series Calculation Scroll and Cycle Evaluation Worksheet				
		☒ Input Pitch, Calc Power					
Input and Output Parameters	Parameter Unit		Case 1	Case 2	Case 3	Case 4	Case 5
Target Net Electric Power Delivered	W		2000.000	2000.000	2000.000	2000.000	2000.000
Aspect Ratio=Vane Height/Orbit Dia	in		1	1	1	1	1
Wall	in		0.1575	0.1575	0.1575	0.1575	0.1575
Cutoff Angle	Deg		0.00	0.00	0.00	0.00	0.00
Ending Scroll Angle	Deg		1350.0	1350.0	1350.0	1350.0	1350.0
Speed	rpm		3600	3600	3600	3600	3600
Gap	in		0.00300	0.00300	0.00300	0.00300	0.00300
Polytropic Exponent (999 for Isen.)	-		1.350	1.350	1.350	1.350	1.350
Supply Pressure	psia		100	110	120	130	140
Supply Temp	F		850	850	850	850	850
Exit Pressure	psia		7.5	7.5	7.5	7.5	7.5
Cutoff/Supply Press Ratio	-		0.9	0.9	0.9	0.9	0.9
Degrees per point	Deg/pt		2	2	2	2	2
Tip Seal Width	in		0.100	0.100	0.100	0.100	0.100
Tip Seal Friction Coefficient	-		0.025	0.025	0.025	0.025	0.025
Number of Effective Flank Gaps			1	1	1	1	1
Boiler Efficiency	%		85.0%	85.0%	85.0%	85.0%	85.0%
Pump Efficiency	%		17.0%	17.0%	17.0%	17.0%	17.0%
Recuperator Efficiency	%		80.0%	80.0%	80.0%	80.0%	80.0%
Generator/Inverter Efficiency	%		87.0%	87.0%	87.0%	87.0%	87.0%
Scroll Mech. Effy (999%=Auto)	%		999.0%	999.0%	999.0%	999.0%	999.0%
Burner Power	W		125	125	125	125	125
Condenser Fan Power Factor	W per Btu/hr		0.006	0.006	0.006	0.006	0.006
Condenser Pressure	psia		7.5	7.5	7.5	7.5	7.5
Sump Temperature	F		175	175	175	175	175
Cutoff Pocket Volume	in^3		1.080	0.963	0.868	0.791	0.725
Exhaust Pocket Volume	in^3		4.320	3.851	3.473	3.162	2.901
Ideal Volume Ratio	-		4.000	4.000	4.000	4.000	4.000
Number of Points	-		406	406	406	406	406
Number of Loops	-		4	5	5	5	5
P at Exhaust	psia		20.582	23.161	25.803	28.506	31.269
T at Exhaust	F		425.476	430.327	434.825	439.045	443.034
gamma	-		1.290	1.291	1.292	1.293	1.293
Gas Constant, Rg	ft lbf/lb R		85.216	85.161	85.107	85.052	84.997
Ideal Volume Flow	acfm		4.500	4.012	3.618	3.294	3.022
Actual Volume Flow	acfm		6.574	5.998	5.528	5.138	4.807
Ideal Mass Flow	lb/hr		34.831	34.175	33.644	33.204	32.832
Actual Mass Flow	lb/hr		50.884	51.097	51.412	51.793	52.220
Indicated Shaft Power Output	W		2737.781	2742.669	2747.892	2753.355	2758.999
Isen. Power Psupply-Pexit	W		4184.394	4310.164	4433.519	4553.983	4671.481
Scroll Isentropic Effy (Indicated)	%		65.43%	63.63%	61.98%	60.46%	59.06%
Minimum Thrust Load	lb		1019.00	1052.93	1087.62	1122.82	1158.37
Maximum Thrust Load	lb		1452.00	1505.85	1560.68	1616.12	1671.96
Average Thrust Load	lb		1206.43	1254.19	1302.43	1350.96	1399.66
Scroll Outer Diameter	in		6.976	6.776	6.603	6.451	6.316
Vane Height	in		0.679	0.651	0.626	0.604	0.585
Tip Seal Friction	W		6.541	7.274	7.998	8.714	9.424
Bearing Friction from Rotation	W		124.585	124.585	124.585	124.585	124.585
Thrust Bearing Friction	W		35.706	37.119	38.547	39.984	41.425
Total Friction	W		166.832	168.979	171.130	173.283	175.434
Mechanical Efficiency	%		93.9%	93.8%	93.8%	93.7%	93.6%
Input Firing Rate	Btu/hr		72908.15	73076.76	73396.46	73816.55	74306.11
Water/Steam Heat Input	Btu/hr		61971.93	62115.24	62386.99	62744.07	63160.20
Recuperated Heat	Btu/hr		4785.60	4901.52	5021.26	5143.31	5266.72
Heat Rejected by Condenser	Btu/hr		52630.62	52757.26	53011.19	53349.62	53746.49
Recup Steam Out Temp	F		225.10	226.07	226.97	227.81	228.61
Recup Water Out Temp	F		268.23	270.08	271.79	273.40	274.93
Gross Brake Power	W		2570.95	2573.69	2576.76	2580.07	2583.56
Generator/Inv Power Loss	W		334.22	334.58	334.98	335.41	335.86
Gross Electric Generation	W		2236.73	2239.11	2241.78	2244.66	2247.70
Condenser Fan Power Load	W		92.55	92.77	93.22	93.82	94.51
Pump Power Load	W		19.17	21.34	23.56	25.85	28.19
Total Electric Loads	W		236.73	239.11	241.78	244.66	247.70
Resultant Pitch	in		0.9940	0.9655	0.9408	0.9191	0.8999
Net Electric Power Delivered	W		2000.00	2000.00	2000.00	2000.00	2000.00
Net Fuel to Electric Effy	%		9.36%	9.34%	9.30%	9.24%	9.18%

Figure 11 - Complete Series Sheet for 5 Cases

5.2 Thermal Deformation in an Elevated Temperature Scroll Expander

5.2.1 Analytical Rationale and Approach

The intent of this finite element analysis is to compute the impact that temperature variations and pressure loading and the mechanical restraints have on the critical flank seal gaps in a scroll. The assumed thermal boundary conditions and pressure loading are based on our current scroll modeling program. The dynamic loading on the bearings and the scroll are not considered.

To evaluate displacements the scrolls need to be restrained. The back of the scrolls are assumed to have geometry similar to four keys. The back faces of these keys are restrained in the axial direction (toward the opposing scroll) and the sides of the keys are assumed to be restrained to being parallel to the coordinate planes (no rotation or net translation). These restraint planes are shown as green in Figure 12.

A key to this analysis is to recognize that the two scrolls are subjected to exactly the same boundary conditions from the thermal and pressure loading. The pressure and temperatures in the trapped pockets radiate out in concentric cylinders. The displacements for any point on a scroll will be the same for the corresponding location on the opposing scroll. Therefore, only one scroll needs to be analyzed to find the changes in gap between it and the mating scroll as long as the direction of the displacements are correctly translated and the reference pinch point locations are correctly picked.

Figure 12 Scroll Rear Restraints

Figure 13 shows where the reference locations are for this study. Note that the inside and outside of the vane are marked with location tags. The white tags represent the fixed scroll and the yellow tags represent the orbiting scroll. For example, at some point in a crank revolution the outside edge of the fixed scroll labeled O14 will come into “contact” with the inside edge of the orbiting scroll at the tag labeled I14. Note also that the points

of contact on the inner face are rotated 180 degrees relative to the point of contact on the fixed scroll.

The manner that the orbiting scroll displacements need to be adjusted can be visualized if the scroll shown in Figure 13 is taken as the fixed scroll. A displacement of the orbiting scroll at the yellow I14 tag to the right, up and out of the paper would be transformed to a displacement to the left, down and into the paper when viewed according to the perspective (and coordinate system) of Figure 13. Therefore, in the fixed scroll coordinate system the orbiting scroll displacements are simply the negative of the values computed in the X, Y and Z directions.

As the triad in Figure 13 shows, the “standard” coordinate system used in these calculations assumes that positive X is to the right, positive Y is up and positive Z is out of the paper. The analysis starts by assuming that both scrolls are identical and are perfectly positioned to yield zero gap at the flanks and between the vane tips and the opposing scroll base. Thus, any displacement computed would represent a change in the gap or a movement sideways relative to the gap direction (the non-critical direction).

Two gaps are computed at each of the reference locations. One is between the tip of the fixed scroll to the base of the orbiting scroll. The other is between the base of the fixed scroll and the tip of the orbiting scroll. Except for the case where the scrolls are isothermal, these gaps are not typically the same value. For each reference location there will also be two sideways displacements and two Z direction displacements.

Figure 13 - Reference Point Specification Drawing (White for Fixed Scroll, Yellow for Orbiting Scroll)

5.2.2 Scroll Geometry and Material Properties

The scroll geometry is based on the advanced HT LIC scroll design as defined by the parameters in the Table below. This table also shows several operating parameters and the computed outlet conditions. This data was used to create the solid model scroll shown in Figure 13 above.

Geometry Specification for Scroll FEA Analysis

Scroll Geometry and Gas Parameters		
Involute Pitch = $2 \cdot \pi \cdot R_g$	in	0.9000
Aspect Ratio=Vane Height/Orbit Dia.	-	1
Wall Thickness	in	0.1575
Cutoff Scroll Angle	Degrees	0.00
Ending Scroll Angle	Degrees	1350.0
Crank Speed	rpm	3600
Vane Height	in	.585
Polytropic Exponent	-	1.350
Supply Pressure	psia	150
Inlet Temp	deg F	900
Exit Pressure	psia	7.5
Cutoff/Supply Press Ratio	-	0.9
Pocket P at exhaust	psi	23.245
Pocket T at exhaust	deg F	399.683
Indicated Scroll Shaft Power	W	2709.08
Net Electric Power Delivered	W	1996.46

The material properties used in this analysis came from published literature for SGL EK 3245 antimony filled graphite and for Nitronic 60 Stainless Steel. No data was found on the thermal conductivity of Nitronic 60. Data for Nitronic 30 was used instead. Cosmosworks has data for a variety of aluminum and titanium alloys, and this data was used for this study. Figures 14 through 17 present the material properties for each of these materials.

Material Properties				
Model Type:	Linear Elastic Isotropic			
Units:	English (IPS)			
Category:				
Name:	Nitronic 60			
Property	Description	Value	Units	Temp Dependency
EX	Elastic modulus	26200005	psi	Constant
NUXY	Poisson's ratio	0.298	NA	Constant
GXY	Shear modulus	1.0000002	psi	Constant
DENS	Mass density	0.27500013	lb/in ³	Constant
SIGXT	Tensile strength	111000.02	psi	Constant
SIGXC	Compressive strength		psi	Constant
SIGYLD	Yield strength	70700.012	psi	Constant
ALPX	Thermal expansion coef	9.8e-006	/Fahrenheit	Constant
KX	Thermal conductivity	0.00024305933	BTU/(in.s.F)	Constant
C	Specific heat	0.10989011	Btu/(lb.F)	Constant

Figure 14 - Properties of Nitronic 60

Material Properties				
Model Type:	Linear Elastic Isotropic			
Units:	English (IPS)			
Category:				
Name:	2024-T3			
Property	Description	Value	Units	Temp Dependency
EX	Elastic modulus	10500732	psi	Constant
NUXY	Poisson's ratio	0.33	NA	Constant
GXY	Shear modulus	4061056.7	psi	Constant
DENS	Mass density	0.10043389	lb/in ³	Constant
SIGXT	Tensile strength	70343.303	psi	Constant
SIGXC	Compressive strength		psi	Constant
SIGYLD	Yield strength	50038.02	psi	Constant
ALPX	Thermal expansion coef	1.2888889e-005	/Fahrenheit	Constant
KX	Thermal conductivity	0.00160497	BTU/(in.s.F)	Constant
C	Specific heat	0.2090301	Btu/(lb.F)	Constant

Figure 15 - Properties of Aluminum (2024-T3)

5.2.3 Temperature and Pressure Boundary Conditions

A moderate dry steam convection coefficient of $57 \text{ W/m}^2\text{K}$ was used to represent heat transfer from the steam to the scroll base and vane faces. All boundary conditions were applied to the surfaces in contact with a pocket at the cutoff position. For example, Figure 18 shows a typical constant pressure of convection boundary condition zone.

Figure 19 shows the assumed steam temperature distributions as a function of crank angle. Similarly, Figure 20 shows the assumed pressure distributions. The average values for the temperature and pressure were used as these fixed conditions. This is entirely appropriate for the convection boundary as the impact of heat transfer changes are relatively slow to propagate through a material, and therefore average values should be used. The pressure actually fluctuates, but the contribution of pressure loading to the vane deflection is relatively small, so using the average value is reasonable.

Figure 18 - Typical Pocket Boundary Condition Faces

Figure 19 - Pocket Steam Temperatures Over 1 Revolution

Figure 20 - Pocket Steam Pressures over 1 Revolution

The average values used in this study are shown in the Table below.

Temperature and Pressure Boundary Conditions

Pocket Description	Pocket Pressure, psig	Pocket Temperature, Deg F
Inlet zone	150	900
Pocket 1	75	680
Pocket 2	35	480
Pocket 3	25	412
Pocket at exhaust	7.5	400

In addition to the reference temperature and pressure boundary conditions, runs were also made with an aluminum scroll with a water cooled backside and an isothermal aluminum scroll, with and without pressure loading. These were performed to characterize the role of pure isothermal heating of a scroll. A separate run was made to evaluate a Nitronic 60 scroll with pressure loading but no temperature loading (room temperature).

5.2.4 Results of Analysis

Seven cases have been evaluate to determine the temperature distributions, stress levels and resulting changes to the gap and other displacements as a result of applying thermal and pressure loading to the scroll set. The Table below summarizes the results for these evaluations.

Summary Results of Analysis

Result Parameter:	T max Deg F	T min Deg F	Max URES in	Gap X (Hit X) in	Gap Y (Hit Y) in	Gap Z (Hit Z) in	Stress Ksi
Scroll Material							
Nitronic 60 P and T	574	408	0.0138	.0023 (.0003)	.0024 (.0005)	- (.0062)	19.6
Nitronic 60 only P	77	77	0.0003	< .0003 < .0003	< .0003 < .0003	< .0003 < .0003	4.0
Alum P and cooled to 200 F	236	205	0.0066	.0012 (.0002)	.0010 (.0001)	- (.0024)	2.9
Alum P and Isothermal	250	250	0.0086	.0014 (.0001)	.0013 (.0001)	- (.0032)	2.9
Alum no P and Isothermal	250	250	0.0086	.0012 (.0001)	.0012 (.0001)	- (.0031)	.01
Titanium P and T	617	399	0.0067	.0012 (.0003)	.0012 (.0005)	- (.0032)	8.8
EK3245 P and T	566	411	0.0035	.0009 (.0005)	.0004 (-)	.0030 (.0015)	4.0

In general, heating a scroll set tends to increase the flank gaps and decrease the tip to base gaps. The maximum interference (termed a Hit) measured for all cases was only 0.0005 inches but the maximum increase in gap was as large as 0.0024 inches. Given that we want to control this gap to around 0.001 inch the increase in gap is a significant effect.

5.2.4.1 Nitronic 60 Scroll Displacements

This study shows that the Nitronic 60 scroll has the greatest growth in the resulting flank gap of any of the materials. This is due to the relatively high expansion coefficient, about twice that of Titanium and nearly six times that of EK3245! The aluminum has a higher expansion coefficient than even the Nitronic 60, but was not subjected to very high temperatures in the evaluation.

The maximum stress for all cases was for the Nitronic 60 scroll. This is primarily due to the high coefficient of thermal expansion (CTE) and the high temperature gradients in this case. The temperature distributions for the Nitronic scroll are shown in Figure 21 below.

Figure 21 - Nitronic 60 Temperature Distribution

5.2.4.2 Water-Cooled Scroll Displacements

The water cooled aluminum scroll had lower displacements than it would have if not cooled, but the values were in the 0.0012 inch range. Because this case still has significant scroll to scroll movement it was decided to evaluate the displacements for the idealized case where the scroll is assumed to be isothermal.

5.2.4.3 Isothermal Scroll Displacements

The isothermal aluminum scroll analysis still shows that the gap will increase as the scroll heats up. This is due to the expansion of the scroll while the crankshaft dimensions remain constant. For the aluminum 250 F isothermal scroll the maximum gap opened up by 0.0012 inches. The change in the required orbit diameter that would accommodate the expanded scrolls is simply:

$$\Delta D_{\text{orbit}} = \text{CTE} \times D_{\text{orbit}} \times (T_h - T_{\text{amb}})$$

$$\Delta D_{\text{orbit}} = 12.9 \times 10^{-6} \text{ 1/F} \times 0.585 \text{ in} \times (250\text{F} - 77 \text{ F}) = 0.0013 \text{ in.}$$

This displacement matches the range of motion seen in the analysis for a given direction for the isothermal aluminum scroll case. For example, in the left to right direction the maximum gap is .0012 in and the maximum hit is .0001 in for a range of 0.0013 in.

Note that for the isothermal scroll with no pressure loading the mechanical stress levels drop to essentially zero.

5.2.4.4 Pressure Loading Impact

For all cases the pressure loading has a minimal impact on the flank gap, given the assumption that the scrolls are fixtured through perfect linkages (no bearing orbit growth). For example, the Nitronic scroll run with only a pressure loading had a maximum node displacement of 0.0003 inches. Also, the isothermal aluminum scroll case was run with pressure loading and without pressure loading and the gap changed by only 0.0002 inches.

5.2.4.5 Titanium Scroll Displacements

Of the metal scroll materials considered the Titanium material yielded the lowest gap growth and lowest stress levels. Given this case did not attempt to cool the scroll this is an encouraging result. Titanium has the added benefit of being considerably lighter than steel alloys.

5.2.4.6 EK3245 Graphite Scroll Displacements

The very low CTE of the Graphite resulted in both low stress levels and low displacements. The maximum stress was only 4 ksi, well below the tensile strength of the material. There may still be issues with regards to stress concentrations at the base of the vanes that are not captured by the current analysis, but this is an interesting result.

5.2.5 Detailed Excel Spreadsheet Result Format

The full Excel spreadsheet that is used to assess model displacements for each of the primary cases evaluated contains approximately 20,000 rows of nodal data. The first page of the complete spreadsheet is shown below in Figure 22.

9:08 Saturday October 15 2005					Update Data using Ctrl-Alt-F9 just before using data									
Nitronic 60 Scroll Set, Pitch = .9in, wall=.157in, VH=058in														
Units in														
Temp in at 900 F. Pin at 150 psi														
Node	UX	UY	UZ	URES	Locations and Displacements for Fixed Scroll Bottom Outside Edge									
					Locations and Displacements for Fixed Scroll Top Outside Edge									
1	4.35E-04	1.07E-03	1.70E-03	2.06E-03	OB1	2703	-0.001136	-0.000815	0.001720	OT1	3568	-0.001304	-0.000881	0.004540
2	2.67E-04	1.09E-03	1.70E-03	2.04E-03	OB2	1309	-0.001798	0.000123	0.001686	OT2	2422	-0.002016	0.002003	0.004475
3	6.04E-04	1.02E-03	1.70E-03	2.07E-03	OB3	2250	0.000102	0.001775	0.001630	OT3	1900	0.001048	0.002074	0.004391
4	7.73E-04	9.57E-04	1.70E-03	2.10E-03	OB4	1574	0.002673	-0.001043	0.001580	OT4	1457	0.003103	-0.001069	0.004261
5	9.37E-04	8.68E-04	1.69E-03	2.12E-03	OB5	1286	-0.001063	-0.004452	0.001579	OT5	976	-0.001104	-0.004889	0.004172
6	1.09E-03	7.58E-04	1.69E-03	2.15E-03	OB6	1242	-0.005294	0.000119	0.001524	OT6	995	-0.005668	0.000189	0.003936
7	1.24E-03	6.25E-04	1.69E-03	2.19E-03	OB7	2190	0.000098	0.005171	0.001524	OT7	1533	0.000148	0.005315	0.003886
8	1.36E-03	4.74E-04	1.69E-03	2.22E-03	OB8	1617	0.005965	-0.001027	0.001494	OT8	1544	0.005897	-0.001045	0.003713
9	1.47E-03	3.06E-04	1.69E-03	2.26E-03	OB9	1216	-0.001024	-0.007645	0.001457	OT9	1348	-0.001049	-0.007484	0.003644
10	1.56E-03	1.24E-04	1.69E-03	2.30E-03	OB10	1147	-0.008384	0.000072	0.001424	OT10	1366	-0.008146	0.000076	0.003526
11	1.62E-03	-6.73E-05	1.69E-03	2.34E-03	OB11	2096	0.000067	0.008238	0.001449	OT11	2003	0.000077	0.007907	0.003542
12	1.55E-03	-1.05E-03	1.70E-03	2.52E-03	OB12	1670	0.008909	-0.000966	0.001551	OT12	1500	0.008399	-0.000893	0.003565
13	2.00E-03	-1.04E-03	1.67E-03	2.81E-03	OB13	1106	-0.000992	-0.010485	0.001717	OT13	1385	-0.000993	-0.009869	0.003671
14	2.00E-03	-9.47E-04	1.67E-03	2.78E-03	OB14	1060	-0.011164	0.000037	0.002001	OT14	1419	-0.010432	-0.000027	0.003923
15	1.98E-03	-7.00E-04	1.67E-03	2.69E-03										
16	1.94E-03	-4.63E-04	1.67E-03	2.60E-03										
17	1.87E-03	-2.37E-04	1.68E-03	2.52E-03										
18	1.78E-03	-2.49E-05	1.68E-03	2.45E-03										
19	1.67E-03	1.71E-04	1.68E-03	2.38E-03										
20	1.55E-03	3.51E-04	1.68E-03	2.32E-03										
21	1.41E-03	5.12E-04	1.69E-03	2.26E-03										
22	1.12E-02	-6.90E-04	4.33E-03	1.20E-02										
23	1.11E-02	-8.74E-04	4.32E-03	1.19E-02										
24	1.09E-02	-9.49E-04	4.27E-03	1.17E-02										
25	1.11E-02	-9.97E-04	3.64E-03	1.17E-02										
26	1.13E-02	-1.05E-03	3.01E-03	1.17E-02										
27	1.15E-02	-1.17E-03	2.39E-03	1.18E-02										
28	1.17E-02	-1.09E-03	2.43E-03	1.20E-02										
29	1.17E-02	-8.92E-04	2.43E-03	1.20E-02										
30	1.15E-02	-7.88E-04	3.06E-03	1.20E-02										
31	1.14E-02	-7.37E-04	3.69E-03	1.20E-02										
32	1.24E-02	-9.00E-04	2.64E-03	1.27E-02										
33	1.23E-02	-1.67E-03	2.71E-03	1.27E-02										
34	1.22E-02	-2.44E-03	2.76E-03	1.28E-02										
35	1.21E-02	-3.19E-03	2.80E-03	1.28E-02										
36	1.19E-02	-3.93E-03	2.83E-03	1.28E-02										
37	1.16E-02	-4.66E-03	2.87E-03	1.29E-02										
38	1.14E-02	-5.38E-03	2.90E-03	1.29E-02										
39	1.11E-02	-6.07E-03	2.94E-03	1.29E-02										
40	1.07E-02	-6.74E-03	2.97E-03	1.30E-02										
41	1.03E-02	-7.39E-03	3.00E-03	1.30E-02										
42	9.86E-03	-8.02E-03	3.01E-03	1.31E-02										
43	9.39E-03	-8.62E-03	3.03E-03	1.31E-02										
44	8.89E-03	-9.19E-03	3.03E-03	1.31E-02										
45	8.35E-03	-9.73E-03	3.02E-03	1.32E-02										
46	7.78E-03	-1.02E-02	3.00E-03	1.32E-02										
47	7.19E-03	-1.07E-02	2.97E-03	1.32E-02										
48	6.56E-03	-1.11E-02	2.93E-03	1.33E-02										
49	5.92E-03	-1.15E-02	2.89E-03	1.33E-02										
50	6.05E-03	-1.17E-02	2.26E-03	1.34E-02										
51	6.73E-03	-1.13E-02	2.31E-03	1.33E-02										
52	7.38E-03	-1.08E-02	2.34E-03	1.33E-02										
53	8.00E-03	-1.03E-02	2.37E-03	1.33E-02										
54	8.59E-03	-9.79E-03	2.39E-03	1.32E-02										
55	9.15E-03	-9.23E-03	2.40E-03	1.32E-02										
56	9.67E-03	-8.62E-03	2.39E-03	1.32E-02										
57	1.02E-02	-7.99E-03	2.38E-03	1.31E-02										
58	1.06E-02	-7.32E-03	2.36E-03	1.31E-02										
59	1.10E-02	-6.63E-03	2.33E-03	1.31E-02										
60	1.14E-02	-5.92E-03	2.30E-03	1.30E-02										
61	1.17E-02	-5.18E-03	2.26E-03	1.30E-02										
62	1.20E-02	-4.42E-03	2.22E-03	1.29E-02										
63	1.22E-02	-3.65E-03	2.18E-03	1.29E-02										
64	1.24E-02	-2.86E-03	2.15E-03	1.29E-02										
65	1.25E-02	-2.06E-03	2.10E-03	1.28E-02										

5.2.6 Summary of FEA Scroll Analysis

This analysis shows that it will be nearly impossible to reduce the gap for a Nitronic 60 scroll below about 0.002 inches unless it is actively cooled. The Titanium scroll can be designed to yield approximately 0.001 inch gaps if very close tolerances are attained for the room temperature geometry.

Active water cooling of an aluminum scroll does not completely eliminate gap growth, unless the scroll temperature is held at room temperature. The uniform expansion of the aluminum would require a proportionately larger orbit diameter, but the orbit diameter is dictated by the crank arm length which does not expand significantly.

The Titanium scroll worked well as it had only a 0.0012 inch displacement without any cooling. Although this displacement may be tolerable, it may be possible to significantly reduce the Titanium Scroll displacements by including a water cooling system. This approach could result in gaps being held to less than 0.001 inches.

A graphite scroll works very well from a thermal and displacement standpoint. If the actual stresses and erosion issues are resolved then graphite would be a good candidate scroll material.

5.3 Prototype CHP System

5.3.1 ECR International M3 Cogeneration Prototype

ECR's has built a series of LIC system prototypes with each prototype improving the on the cost, size and performance characteristics of the system. The M3 prototype shown in Figure 23 below has been able to reduce the overall size of the system to that of a comparable residential heating unit. This Figure shows the relative size of the unit as compared to an existing gas fired boiler. The size reduction has been achieved with advances in boiler, inverter, and generator design.

Figure 23 - M3 Micro-CHP with Commercial Boiler

The M3 Prototype as shown is configured to be used as a new or replacement resident heating appliance. The heating output can range from 80,000 to 200,000 BTUs/hr. The electrical output is from 2 KW to 3 KW. The electrical output is connected to the existing electrical service in the home. Electrical power not directly consumed by the residence is feed back to the electric grid. The result is that the net usage of residence electrical power consumption from the grid is reduced and the electric meter measuring the flow of electrical power into the house slows down or runs backwards. This yields a direct credit on the customer's electric bill for the net amount of power produced.

5.4 M3 Prototype Scroll Expander

5.4.1 Design

The scroll expander used in the M3 prototype was ECR's third generation of scroll expander. In this generation of expander hard coat anodized aluminum was introduced as the scroll material to reduce cost and weight. The mechanical control was designed with the main and idler shaft configuration.

In this new generation the bearing design has been modified to improve accuracy of the moving scroll orbit. As shown by Figure 24, a combination of angular contact and rolling bearing assemblies were designed into the construction of this expanded to achieve a more rigid operation. Also shown in Figure 24 is the introduction of a flexible diaphragm to separate the exhaust steam from the bearing and shaft assemblies.

Figure 24 - M3 Scroll Expander

5.4.2 Operation and Performance

The M3 prototype has been run to test the operating characteristics of the system as a combined heat and power appliance. Operating at a heat input of 100,000 BTUs/HR, the M3 was able to deliver over 2 KW of electrical power. The device has been demonstrated numerous times and has a smooth startup, low noise and low vibration operation.

Performance testing of the M3 scroll expander at ECR's dynamometer lab produced 2.4 KW of shaft power at an isentropic efficiency above 45 %.

Operation of the device was limited by the life time of the flexible barrier used to separate the exhaust steam from the bearings and shaft components. After replacing several of the flexible diaphragms it became apparent this arrangement was not going to produce a long term solution. In our next generation of expanders we addressed this issue with the introduction of a metal bellows as both the orbit control mechanism and the exhaust steam barrier.

Our attempt to make the operation of the bearing assembly more rigid was partially successful. We continued to have contact between the moving and fixed scroll at speeds above 2700 RPM.

We also used this expander to experiment with new materials for the tip seals. The common tip seal material used for compressors and vacuum pumps is TEFLON or a mixture of materials with TEFLON as the base material. The thermal properties of TEFLON in the expected operating range of the LIC expander are not consistent an expectation of a long operating life. We used this generation of expander to initially test a combination of tip seal materials and scroll coatings.

6 Scroll Expander Fabrication

Figure 25 shows a cross section of the baseline scroll expander design. In November we were able to finalize the bellows design, bearing selection and the detailed design of the scrolls. Components from this baseline design are being used to construct three scroll variants using the following materials

- 1) Nitronic 60
An alloy similar to 304 stainless steel with anti-galling characteristic.
- 2) Titanium 6AL4V
The lower density of titanium should provide weight reduction and improved mechanical performance.
- 3) SGL Carbon: EK3245 Graphite
A low density and non-galling antimony filled graphite.

Fabrication of the components for the first Nitronic 60 expander was completed in February and assembly was completed in March 2006. In April fabrication of the Titanium and Graphite scrolls was completed.

Figure 25 - Baseline Scroll Expander

Figure 26 is picture of the S1 scroll expander components.

Figure 26 - S1 Scroll Components

Figure 27 shows the assembled Nitronic 60 expander with the fixed scroll removed. The yellow color of the fixed scroll on the bench is the result of a PVD titanium nitride coating applied to reduce seal wear.

Figure 27 - Nitronic Scroll Expander

6.1 Expander Testing

Assembly and preliminary performance testing of the Nitronic 60 scroll expander with steam was started in March 2006. This expander is shown installed in our dynamometer lab in Figure 28. The expander dynamic balancing has been verified by driving the expander with an electric motor up to 3000 RPM. In a similar test, the friction torque was measured, and found to be greater than predicted. At 3000 RPM the friction torque was 2.3 Newton-meters which results in a friction power of 700 Watts.

Figure 28 - S1 Scroll Expander Installed on Dynamometer

Initially, the operating speed of the expander was limited to 1800 RPM, and the inlet steam temperature to about 400F, until tuning of the alignment between the fixed and orbiting scrolls is completed. This final tuning is required to eliminate contact between the scroll vanes and possible damage to the surfaces. Tuning of the alignment requires the relative position of the two scrolls to be adjusted within 0.002" from their present position. The present alignment is controlled by dowel pins locating the fixed scroll to the housing. Final alignment will be accomplished by replacing the existing dowel pins with precision offset axis pins which will be rotated to tune the position of the fixed scroll relative to the orbiting scroll. Once the final alignment is attained, the location of the fixed scroll will be secured with permanent tapered pins. We are continuing to evaluate variations in our manufacturing and assembly procedures to reduce the misalignment at first assembly.

Within the initial speed limitation we were able to generate 1.4 KW of shaft power at the end of March 2006 at an isentropic efficiency just under 40%. As we increased the shaft

speed up to 1800 RPM we observed a steady increase in efficiency. We expect a continuation of this trend up to target operating speed of 3600 RPM.

With partial improvement of the scroll alignment during April 2006, the operating speed was increased to 2800 RPM and the output power increased to 1.9 KW. During May the alignment of the scrolls is to be improved and continued improvement in performance is expected.

Fabrication of the titanium orbiting scroll was delayed to allow for possible modification as a result of our testing of the Nitronic 60 expander. We made one modification to the assembly method for securing the orbiting scroll to the bearing hub. The titanium scrolls with the design modification have been received. Assembly and testing is scheduled for May 2006. Testing of the graphite scroll expander is scheduled after completion of the testing and evaluation of the titanium scroll expander.

6.2 Scroll Surface Finish and Coating

Seal life is the primary determinant of the required scroll expander maintenance cycle. We have found that proper preparation of the seal surface finishes has a significant impact on wear rates. In parallel to our efforts to maximize expander performance, we have constructed a surface finishing apparatus to improve seal life. This apparatus is used to prepare the seal surfaces prior to the application of surface coatings. Application of specific surface coatings is expected to also increase seal life.

The scroll polishing apparatus was completed in January of 2006 and put to work. The apparatus is shown in Figure 29 opened up, with a scroll to be polished mounted in the apparatus, and the mating brass polishing tool in the foreground. Figure 30 shows the apparatus assembled for operation. This mechanism works by holding the brass lapping tool in place while an orbiting mechanism below the scroll drives the work piece. The orbiting mechanism uses three crank shafts to accurately move the work piece while

Figure 29 - Polishing Apparatus Open

Figure 30 - Polishing Apparatus Assembled

avoiding collision of the scroll vane flanks with those of the polishing tool. The orbit is controlled to polish the scroll base to within 0.025” of the scroll vane flank.

The apparatus was tested using an early Nitronic 60 fixed scroll that did not fully meet our dimensional requirements. The result of running the apparatus was to progressively improve the scroll base surface finish from an initial machined finish of Ra 14 to a finished Ra of 3.5 in three two hour runs with increasingly fine lapping compounds. This scroll was then shipped to Northeast Coating in Kennebunk, ME for electron beam titanium nitride coating. Figure 31 shows this scroll after coating. We were able to achieve a uniformly polished surface at the scroll base without further polishing after coating.

The polishing of the titanium fixed scroll base surface has required adjustment of the process we used to polish the first Nitronic 60 scrolls. To polish the Nitronic 60 fixed scroll we ran the scroll in our polishing apparatus with increasingly finer grits (500, 800 and 1200) of water based lapping compound for two hours at each grit size. We found that titanium would successfully polish with 500 and 800 grit compounds but only if we used petroleum based lapping compound and did not use the 1200 grit compound. When the 1200 grit compound is used, galling of the surface finish occurred, resulting in the creation of ring shaped scratches on the scroll base surface, as shown in Figure 31. By using the 800 grit lapping compound for an extended period of time, we were able to polish the surface to a surface roughness of Ra 5.0.

The polished titanium scroll was then shipped to Electropolishing Systems in Plymouth, MA for application of an AMS 2488D type 2 anodic coating. This coating is intended to increase seal life and provide an anti-galling protection. Figure 32 is a picture of the polished and coated titanium scroll.

**Figure 31 - Titanium Scroll After Polishing
with 1200 Grit Compound**

**Figure 32 - Nitronic 60 Scroll After Polishing and
Titanium Nitride Coating**

6.3 Three Idler Crank Scroll Design

A design commonly used for scroll mechanisms is the three idler crank and a single main shaft configuration. In this design, three idler shafts provide the motion control definition while the main shaft transfers power out of the mechanism. In our application of this design, we have separated the radial and thrust loads within the scroll expander between the main shaft and three idler shafts respectively. In previous configurations of our steam expanders, a single main shaft and angular contact bearings located on the shaft are required to support both thrust and radial loads created during operation of the expander. This single shaft configuration produced conflicting design specifications for the bearings. The first specification required precise orbit motion under significant radial loads resulting in a rigid bearing assembly. This rigid bearing assembly resulted in significant rotation resistance and therefore significant power losses. Separating the loads between the two bearing assemblies eliminated these conflicting requirements.

In the new design, the pair of angular contact bearings at the center of the orbiting scroll is replaced with a single needle roller bearing. This change prevents transmission of any thrust loading to the main shaft and prevents friction losses at the scroll center bearing and the associated housing bearings. The needle bearing provides precise control of the moving orbit path while allowing transmission of the output torque load onto the main shaft.

The bearings mounted on the three idler shafts are angular contact bearings. This assembly has two specific roles. The first role is to provide an anti-rotation resistance to prevent the scrolls from locking while the second role is to provide support for the thrust load created by the pressurized steam between the scrolls. In this design, the angular contact bearing are designed to fit into the assembly in a way to prevent radial loading on the idler bearing assembly. As described earlier, the radial load is supported by the needle bearing at the center of the moving scroll. The thrust load is split between the three supporting idler shafts. Because the shafts are mounted at the perimeter of the moving scroll, they also provide stability by resisting possible tipping of the moving scroll during operation.

Figure 33 is a picture of the internal mechanism showing the relative position of the three idler cranks and the main shaft. In this picture the angular contact bearing are removed from one idler crank at the lower right to show the configuration of an idler shaft. The main shaft is shown in black with the bearings supporting the shaft within the housing visible. The bearings supporting the main shaft within the housing are deep groove ball bearings. The needle bearing supporting the moving scroll on the main shaft is not visible. The counterweight to offset the centrifugal forces generated by the moving scroll is shown in green. A partial cut-away of the moving scroll shows the interleaving of the scroll vanes.

Figure 33 - Internal Orbiting Mechanism

Figure 34 is cross-section of the complete expander assembly. This cross-section includes a view of the main shaft and bearing configuration with part of the needle bearing visible at the end of the shaft on the right. This section also shows the positioning of one idler shaft assembly within the support housing. The reduced over-all depth of the expander assembly does represent a reduction in size of the expander assembly below previously constructed expanders using a metal bellows to control the moving scroll orbit path. To prevent steam from entering into the shaft and bearing cavity, a perimeter face seal made from antimony filled graphite runs on the flat surface of the component shown in yellow in Figure 34.

Figure 34: Cross-Section of Three Idler Crank Scroll Expander

Based on testing of prototype systems we expect this expander to operate with an isentropic efficiency in the range of 50% to 60 %. The expander has a steam inlet at the center of the scroll, as shown to the left in Figure 34. The steam exhaust port is directly below the scrolls. The expander operating speed is to be between 3000 and 3600 RPM with a power output is above 2.5 KW.

7 Steam Generator and Support Equipment

7.1 Burner and Boiler

Superheated steam for the scroll expander is produced in a sheet metal boiler where water flowing through a small diameter tube is heated and converted to steam. The heat is delivered by a commercial diesel/JP-8 burner from Espar, modified to meet our requirements. Hot gases from the burner pass axially over two concentric coils made of small diameter tubing. The inner coil consists of bare tubing. The outer coil is made of fin tubing. The two coils are connected in series. Boiler feed water enters the outer coil at the end farthest from the burner. Superheated steam exits the bare tube coil at the location where the combustion gases are hottest. The targeted combustion efficiency is 85%.

The complete boiler with attached burner is approximately 6" in diameter and 12" long. A cross-section of the boiler and burner assembly is shown in Figure 35.

Figure 35 - Cross-Section of Boiler and Burner Assembly

The commercially available Espar Hydronic 10 oil burner consists of a burner tube, high velocity blower, fuel pump, igniter, and inlet plenum, integrated in a single aluminum casting. Of these, only the burner tube, fuel pump, and igniter are being used. We have modified the electronic circuit for pulsing of the fuel pump which allows the firing rate to

be increased to 70,000 Btu/hr. A compact inlet plenum was designed allowing a higher output combustion air blower to be connected. This is necessary to maintain a minimum excess air level of about 25% at the highest firing rate, which the existing Espar blower cannot do.

During December 2005 tests were carried out on the burner to find a suitable combustion air blower. The pump frequency was increased to 12 Hz, producing a fuel flow with an equivalent firing rate of 73,300 Btu/hr when burning diesel fuel and 69,300 Btu/hr when burning JP-8. A 70,000 Btu/hr firing rate is sufficient to produce the desired steam flow (50 lbs/hr) and boiler exit temperature (850 F) if we raise the combustion air temperature to 400 F using the energy in the flue gas from the boiler. This has been successfully demonstrated in a performance test where we used a commercially available electric air heater to preheat the approximate 15 cfm combustion air from 70 F to 350 F.

The original Espar blower, when running at its maximum speed (~8,500 rpm), was capable of providing only about 6% excess air when burning diesel fuel at the 12 Hz rate. Furthermore, at 8,500 rpm this blower is very noisy. We replaced the Espar blower with a blower from Pabst EBM. Excess air levels of 25% to 30% were easily obtained with the EBM blower running at less than full speed. Furthermore, the noise level is now much lower.

To test the boiler and various other components from the flow circuit, a test loop was designed allowing these components to be tested under design operating conditions without the presence of the expander, which is undergoing development on a parallel track. Components to be tested include the boiler, the feed water pump and booster, combustion air blower and preheater, recuperative heat exchanger, air-cooled condenser, vacuum pump, and flow controls.

7.2 Test Loop

So far the initial version of the test loop has tested only the burner and boiler, feed water pump, and combustion air blower. The objective was to demonstrate that the desired steam exit flow and temperature can be produced under design operating conditions, and that the composition of the combustion products exiting the boiler is within acceptable parameters. A water-cooled atmospheric pressure condenser was installed to condense the steam and return cold water to the sump. In the near future additional components will be added to produce the expander discharge conditions, allowing testing of the recuperator, the sub-atmospheric sump and vacuum pump, and the air-cooled condenser. The initial version is illustrated in Figure 36. The completed test loop is shown in Figure 37.

Figure 36 - Burner and Boiler Test Loop

Figure 37 – Boiler and System Component Test Bench

On the left are the burner and boiler, with the exhaust stack for the combustion gas and desuperheater boil-off behind. The black instrument to the right of the boiler and in front of the pressure gauges is the exhaust gas analyzer. In the blue and gray box in the center is the DC power supply for the feed pump. Further to the right is the control circuitry for the combustion air blower and the fuel pump. At the right end of the bench is the control console, which is currently used mainly for the flow rate and temperature instrumentation. It is planned to gradually install all of the controls and power supplies for the test loop in this box as they are developed, and additional panel openings have been made in anticipation of this. At the extreme right, the small box with the red knob is an emergency shutoff switch. This switch turns off the fuel pump, but leaves the combustion air blower and boiler feed pump running to assist in cooling down the system.

Figure 38 shows a chart of key test data taken over a period of approximately 80 minutes.

Figure 38 – Boiler test loop data

7.3 Test Loop Supervisory Control System

Thermocouple instrumentation was selected, purchased and received for the burner/boiler test loop. Omega Engineering Inc. CNi16D53 or CNi1653 modules for monitoring boiler steam outlet temperature, sump temperature and Exhaust Gas Temperature (EGT) were

specified with analog outputs that can be interfaced with the A/D converter channels on the microprocessor. Although these thermocouple displays would not be suitable for the eventual operating environment of the system, they will be useful beyond the immediate flow-loop test requirement, in developing the system software.

The EGT thermocouple will serve during start-up to indicate whether or not the flame is lit, information that will be used to automate the start sequence. A unit will be installed to monitor the steam temperature at the inlet to the back-pressure regulator that provides the simulated function of the expander in the test loop. This unit as well as the boiler output steam temperature display will be programmed to provide an alarm function in conjunction with onboard relays should the monitored temperatures exceed safe ranges. The relays will be wired in series with the fuel pump output to shut down the burner fuel flow. The safety loop will also incorporate the output of a flow rate instrument should feedwater flow drop below a safe level.

For the purpose of simulating the operation of the feedwater pre-heater and combustion air pre-heater in the test loop, the boiler feedwater and combustion air will be heated electrically. Two modules in the CNi16 series were purchased and received to control these temperatures.

Conclusion

This LIC development program has provided many significant results regarding the marketing and technology of scroll based Micro-CHP systems. A detailed marketing assessment has been conducted which considered both the influence of climatic conditions and equipment and utility costs in a simulation analysis. This simulation determined the run times and cost savings for a LIC Micro-CHP system operating during a typical meteorological year in several regions of the country. Payback times on the order of 4-6 years are seen for reasonable economic assumptions. Approximately half the total annual electric power needed by a typical home can be produced by the Micro-CHP system.

Advances have been made in the design, fabrication and testing of a high temperature steam scroll expander. This challenging development effort resulted in this programs first expander, the S1 scroll. Additional scroll development has resulted in the design and fabrication of a three idler crank version of a scroll expander which takes advantage of the load sharing possible between the idlers and the main power crank.

One aspect of scroll design that is inherently difficult to predict is the impact of leakage from one trapped pocket to another. In this program a state-of-the-art modeling tool has been developed that fully characterizes the scroll as well as the overall cycle efficiency. Use of this tool has allowed an accurate prediction of the net electric power delivered by a given scroll geometry.

High temperature operation tends to improve cycle efficiency. However, high temperatures also induce thermal growth in a scroll machine that can make it difficult to hold the critical vane to vane gap within the required tolerances. A Finite Element Analysis has been conducted with a range of materials and operating conditions to characterize the magnitude of the gap tolerance issues which result. This has provided insights into the scroll design and materials benefits.

ECR International, and its joint venture business Climate Energy, have made groundbreaking progress in the development of the technology and marketing basis for a Liquid Injected Cogeneration (LIC) business. Through Climate Energy, ECR has installed into 20 pilot program test sites its Micro-CHP Honda IC engine based cogeneration systems. One such test site installation is shown in Figure 39.

Figure 39 - Climate Energy Micro-CHP Installed in Massachusetts Home

These IC engine based systems have been praised by homeowners and have resulted in electric power savings exceeding \$100/month. This has resulted in an associated reduction in net energy consumption and emissions of CO₂ and other pollutants needed to provide the electric power supplied by the Micro-CHP units. This successful pilot program shows that if the LIC technology can be brought forward, with its lower price point than the IC based system, then its potential for introduction to the residential Micro-CHP market is excellent.

Our plan moving forward with these Micro-CHP systems is to install 250 units during 2007 in eastern Massachusetts, with some units also earmarked for New York and Connecticut. During this process Climate will be working to ensure that our service and support capabilities are more than sufficient to handle the customer care needs of the homeowners.

As the advanced LIC technology moves forward it has the promise of reducing the installed costs for Climate's Micro-CHP systems. This lower cost solution will provide additional options for our customers and will be able to take advantage of the marketing, distribution and installation infrastructure currently being put into place.