A Study of Optimizing the System Integration of Combined Heat and Power (CHP) with Absorption Cooling for Cold Storage Applications: Design Considerations, Modeling and Life Cycle Costing

Gregory W. Stevens and Ronald K. Ishii, Alternative Energy Systems Consulting, Inc.

ABSTRACT

This study examines the concept of integrating combined heat and power (CHP) and absorption cooling with mechanical refrigeration at a cold storage facility, using a combination of system design analysis, simulation modeling and life cycle costing. An analysis methodology and computer simulation tool were developed and used to compare generation technologies, optimize absorption chiller and generator sizing, and evaluate various integrated system configurations. Other elements of the study include evaluating optimal CHP system operating schedule, assessing optimal integration of absorption chiller with mechanical refrigeration system (i.e., unload cooling tower, provide sub cooling) and calculating life cycle cost.

CHP systems produce both useable electrical and thermal energy. When employed properly CHP systems are more fuel efficient than simple cycle applications since a significant amount of the waste heat generated from the power plant can be captured and utilized onsite. Reciprocating engines, microturbines and gas turbines have been used successfully as prime movers in various CHP applications (i.e., heating hot water, domestic hot water, absorption space cooling). In the case of space cooling, the CHP and absorption chiller are configured to operate in sequence; the recovered waste heat from the prime mover is used to drive the absorption chiller (either direct- or indirect-fired). Simultaneously, the generator is serving a portion of the host site's electric load. In the case of industrial refrigeration, the generator would operate in the same manner, while the absorption chiller supplements the operation of the existing mechanical refrigeration system. This new integrated approach could be an attractive alternative, as the re-configured system may provide several benefits: 1) improvement in mechanical refrigeration system efficiency, 2) reduction in overall facility energy consumption, 3) reduction in energy costs and 4) reduction in product loss because of increased power reliability.

Introduction

Cold storage facilities typically have high energy usage profiles and operate under tight profit margins, making energy costs and reliability critical operation considerations. Products, such as ice cream, must be stored between 0 °F and 10 °F to avoid spoiling and losing product and profit. These low storage temperatures require significant refrigeration and a reliable power source, making a cold storage facility an ideal study site for investigating this new integrated approach. Currently, CHP and absorption chiller integrated systems are used widely for space cooling in commercial buildings where supply air temperatures are higher than what is needed for cold storage. However, to this point, the integration of CHP and absorption cooling with mechanical refrigeration has not been investigated in great detail. Consequently, this concept is the focus of this study.

Background

Cold storage is a significant market in the U.S. and a critical link in the supply chain of moving food products from the "farm to the shelf". As energy prices continue to escalate, so do storage costs and consequently retail prices. At the end of 2003, the general refrigerated storage capacity in the United States totaled 3.16 billion gross cubic feet, which was a 4 percent increase when compared to two years earlier. In that same year, the five States with the largest gross general warehouse capacity (in million cubic feet) were: California with 449; Florida, 253; Washington, 189; Wisconsin, 167 and Texas with 159 (DOA 2004).

In recent years, California has experienced an increase in the retail price of many products requiring cold storage, due in part to escalating energy costs. Since industrial refrigeration systems consume significant amounts of energy (both kW and kWh), their impact on the electric grid is extensive. The integrated system proposed in this study could be a viable approach to reducing the electric load on the grid from industrial refrigeration systems across the country and at the same time decreasing the operating costs of cold storage facilities. The perceived economic benefits would be realized by utility rate payers, cold storage facility owners and retail customers alike.

Study Objective

The focus of this study is determining the optimal configuration of an industrial refrigeration system integrated with absorption cooling and a reciprocating engine-generator with heat recovery. A reciprocating engine-generator has been selected as the technology of choice, as opposed to a microturbine or gas turbine, because the capital cost is typically lower.

Approach

This study involves developing a computer model that simulates the hourly refrigeration load and energy consumption of a typical cold storage facility that houses product at 35 °F year round. Further analysis compares the energy consumption and life cycle costs of a typical industrial refrigeration system to the proposed integrated system.

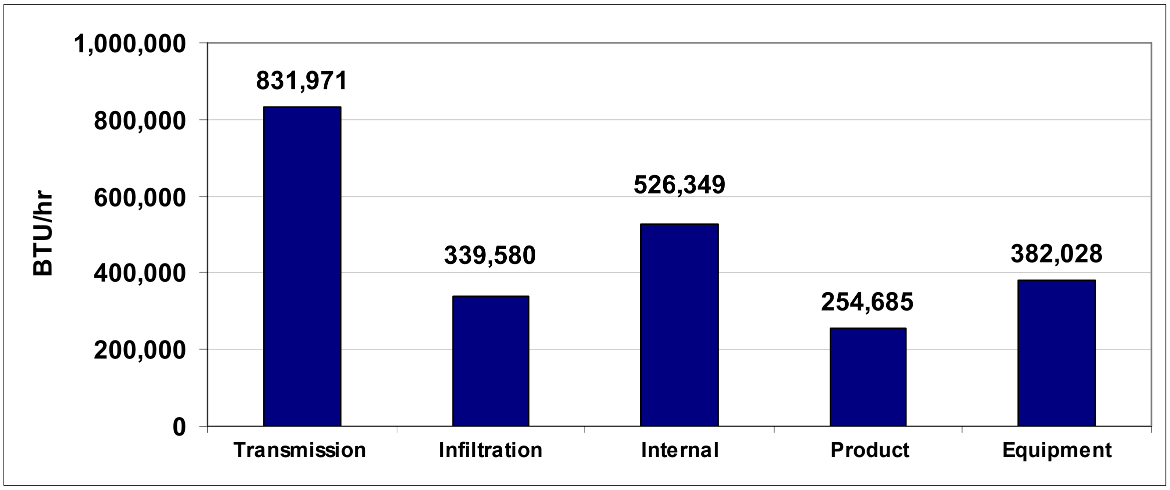
Computer Model

The computer model consists of three major components. The front end of the model simulates hourly refrigeration loads and associated energy consumption of system elements for an entire year. The intermediate component uses hourly refrigeration loads and resulting system energy consumption to evaluate the sizing and performance of the absorption chiller and engine-generator. The first two components consist primarily of the following equipment performance curves: vapor compressor, evaporator, evaporative condenser, subcooler, refrigerant piping, reciprocating engine-generator, waste heat recovery and absorption chiller. The back end of the model assesses the life cycle costs of each case study.

Cold storage facility. A model of a typical cold storage facility was created, including size, construction type and representative energy systems. The model facility is comprised of 200,000 ft^2 of conditioned space and typical warehouse construction. The product storage conditions are 35 ˚F and 65% relative humidity year round. The case study location is San Diego, California.

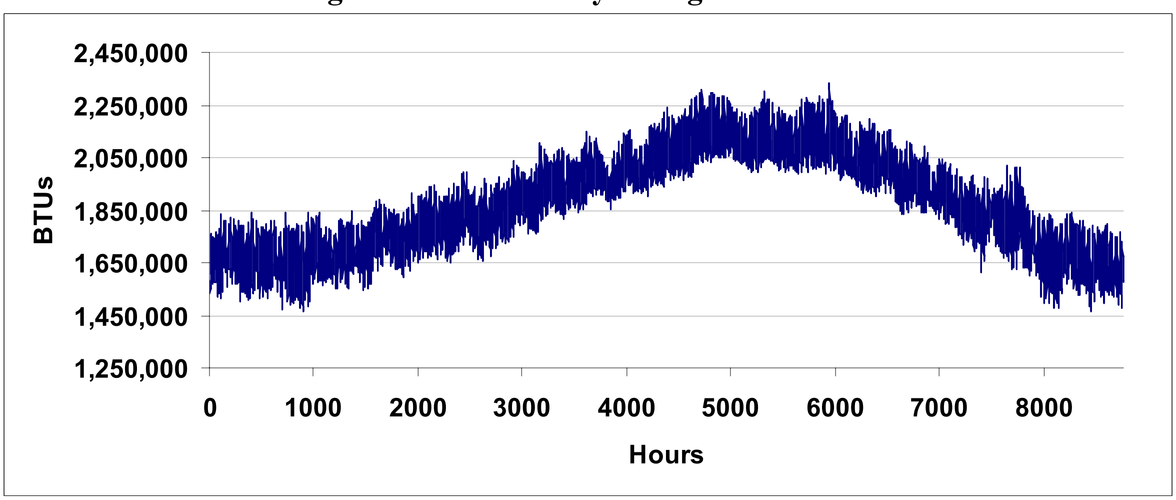
Facility refrigeration loads. Cooling loads were classified into different categories: transmission, infiltration, internal, product, and equipment[1] (ASHRAE 1997). Hourly loads were calculated using typical hourly weather data for San Diego (CIMIS 1982). The maximum hourly cooling load of the mock facility is 195 Tons. The maximum hourly load (Btu/hr) for each refrigeration load category is shown below in Figure 1.

Figure 1. Maximum Hourly Refrigeration Loads



The simulated hourly refrigeration load profile of the study facility for a typical year is illustrated in the following chart, Figure 2. The x-axis represents the hours in a given year.

Figure 2. Total Hourly Refrigeration Load



[1] Transmission - heat gain through walls, floor and ceiling; Infiltration - air exchange through doorways and/or cracks; Internal - process motors, lighting, people and trucks; Product - heat gain by new product and generated by products in storage; Equipment - fan motors, defrost, etc.

Refrigeration system. The simulated refrigeration system consists of an ammonia vapor compression cycle comprised of the following components: a single screw compressor with slide valve, expansion valve, recirculating evaporator fans and a variable speed evaporative condenser. A screw type compressor was selected over a reciprocating because it has less moving parts, consequently reducing maintenance costs while extending equipment life.

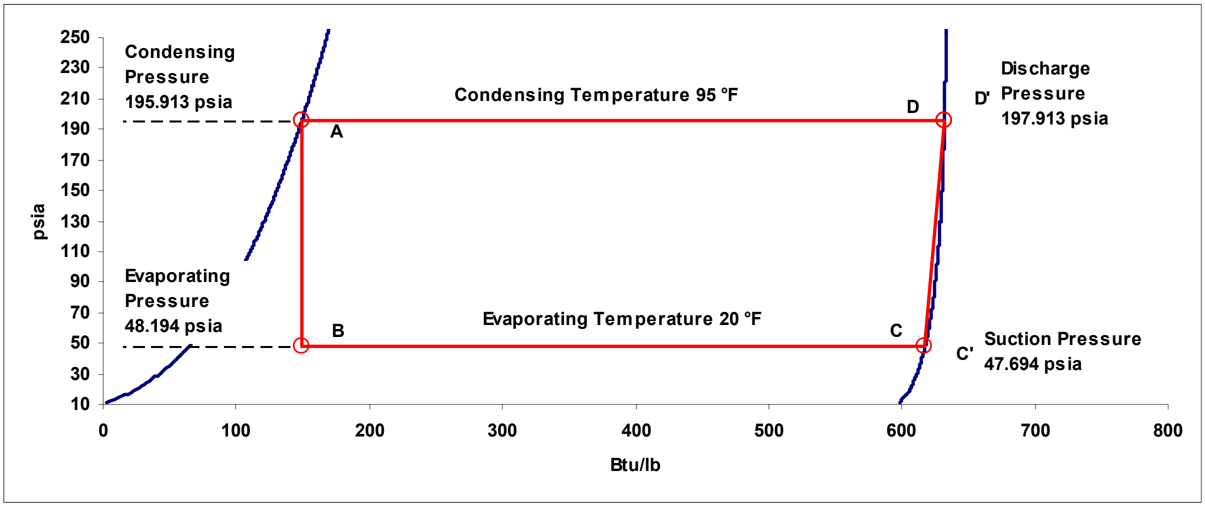
Refrigeration loads and energy consumption of each system component were modeled under normal operating conditions for the study site during a typical weather year. Part load operation of each system component was modeled using manufacturer performance curves[2]. The model accounts for suction and discharge pressure losses as a result of expected friction in the refrigerant lines. The base case refrigeration system was modeled under the conditions summarized below in Table 1.

Table 1. Refrigeration System Design Conditions

System Design Conditions	Values
Evaporating Temp. (°F)	20.0
Evaporating Press. (psia)	48.3
Suction Press. Loss (psia)	0.5
Suction Superheat (°F)	0.0
Condensing Temp (°F)	95.0
Condensing Press. (psia)	196.1
Discharge Press. Loss (psia)	2.0
Subcooling (°F)	0.0

The simulated state conditions of the R-717 vapor compression cycle are illustrated in the following pressure vs enthalpy diagram, Figure 3.

Figure 3. R-717 Vapor Compression Cycle - Pressure vs Enthalpy



[2] *Vilter Pro Programs*, provided by Vilter Manufacturing Corporation, 5555 South Packard Avenue, Cudahy, Wisconsin. Selected equipment includes: VSM-601 Compressor, FP-34-83-1-FA-W Evaporator Fans and VSC-301 Evaporative Condenser.

The nominal cooling capacity for a vapor compression cycle can be calculated knowing the specific enthalpy of the refrigerant leaving both the condenser and evaporator (Points A and C shown in the diagram above), as well as the refrigerant mass flow through the evaporator. The cooling capacity of the simulated screw compressor is calculated as follows:

$$\begin{aligned} Capacity &= (h_a - h_c)\,[\text{Btu/lb}] \times \text{refrigerant flow}\,[\text{lb/hr}] \\ &= (617.593 - 149.2) \times 5{,}670.4 \\ &= 2{,}655{,}976\ \text{Btu/hr}\ (221.4\ \text{Tons}) \end{aligned}$$

As a result, the compressor's refrigeration capacity is greater than the calculated maximum refrigeration load of the case study facility (195 Tons). This fact is unavoidable, as the capacity rating of the next model down offered by Vilter is insufficient. However, the additional capacity is not an issue, as the selected compressor operates relatively efficiently at part load. Part load efficiency curves of the simulated screw compressor can be seen in Figures 4 and 5.

Figure 4. Compressor Efficiency (BHP per Ton) vs Refrigeration Load

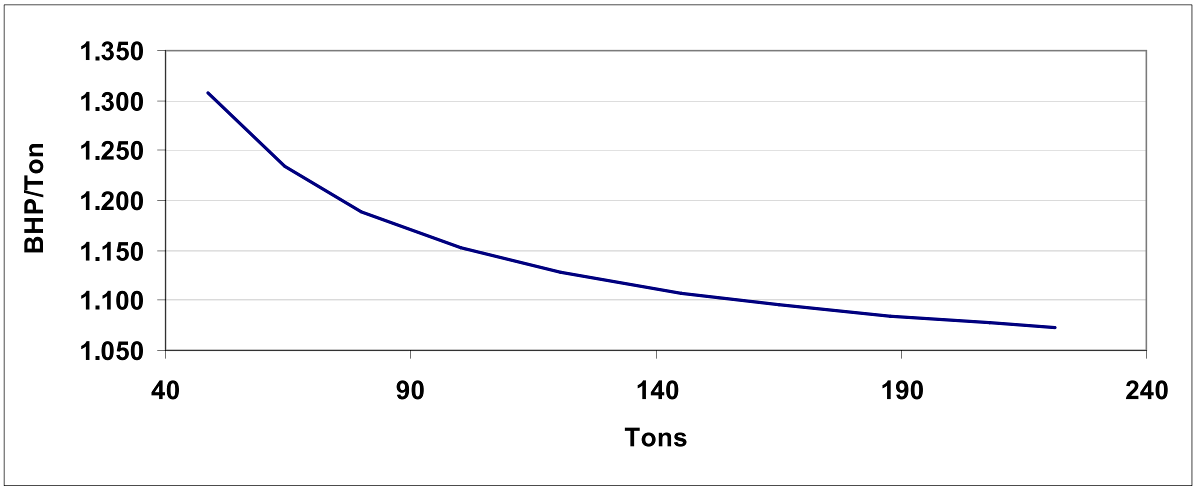
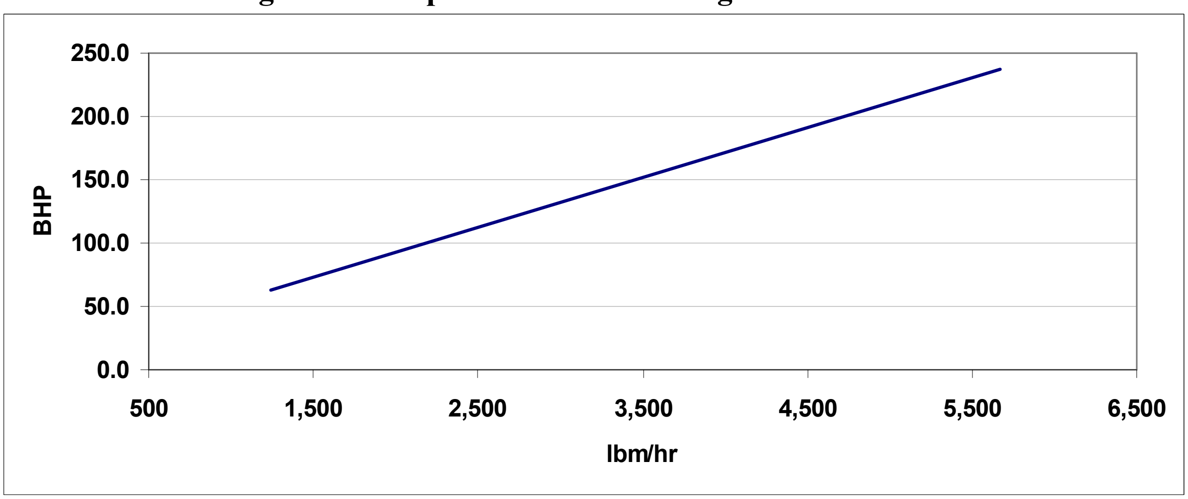
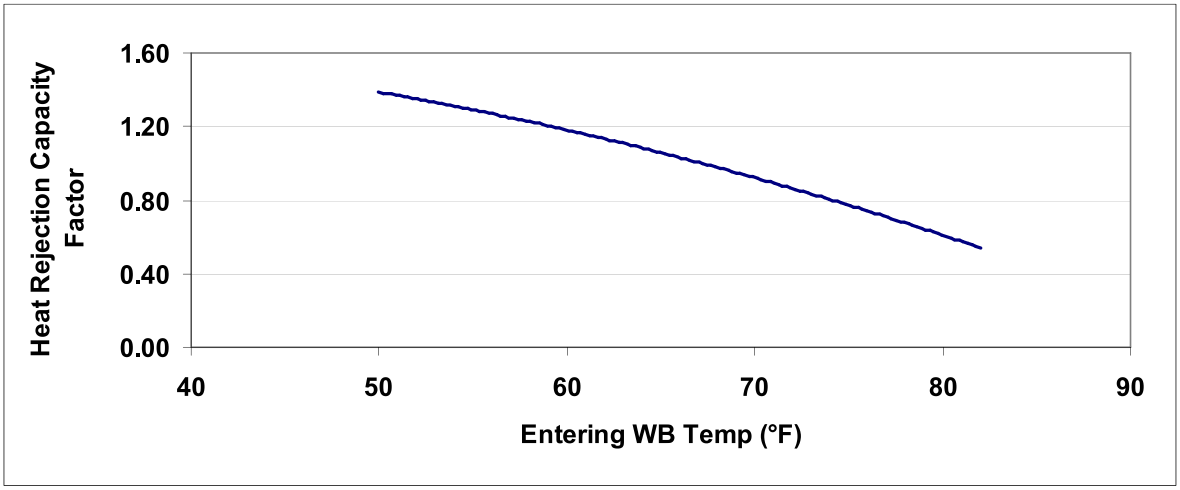


Figure 5. Compressor BHP vs Refrigerant Mass Flow



The next chart, Figure 6, shows the evaporative condenser performance as a function of entering air wet-bulb temperature.

Figure 6. Evaporative Condenser Performance



The evaporator fans were sized to meet the maximum refrigeration load of the facility. Performance specifications are listed below in Table 2.

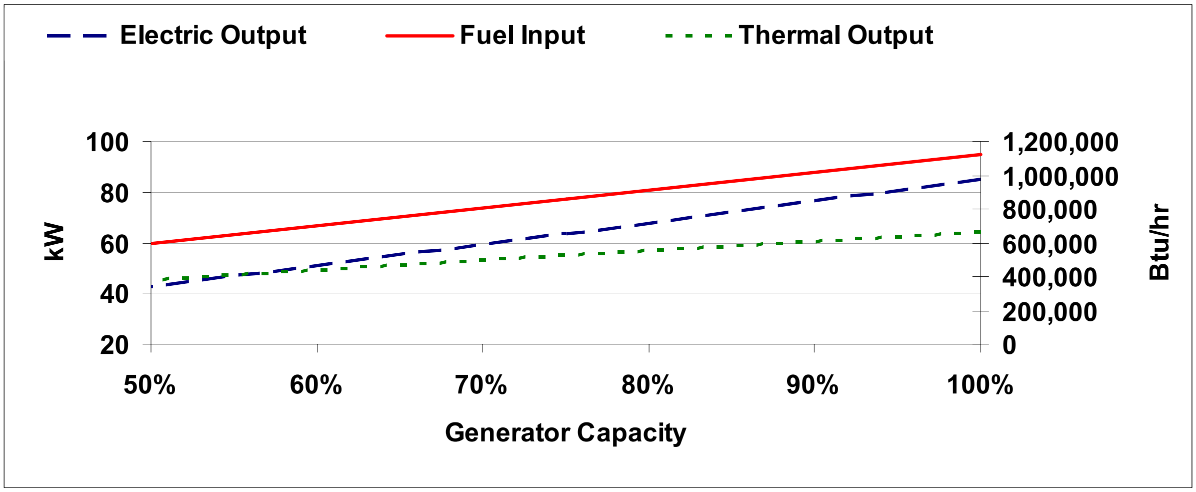
Table 2. Evaporator Fan Specifications

Type	Specification
Refrigeration Feed	Recirculation
Total Capacity (Tons)	195
Number of Units	10
Delta Temperature (˚F)	15
Evaporating Temperature (˚F)	20
Defrost	Water

CHP and absorption chiller. The CHP system consists of an 85 kW reciprocating engine-generator with jacket water and exhaust gas heat recovery[3]. The expected thermal process flow of the conceptual integrated system is as follows: engine waste heat is captured in a closed hot water loop → hot water supply feeds the hot-side of a 29 Ton absorption chiller[4] → chilled water supply feeds the cold-side of a water-to-vapor subcooler heat exchanger[5] → connected to the condenser discharge line in the vapor compression cycle. A preliminary screening analysis indicated the proposed location of the subcooler is optimal. Part load performance curves for the engine-generator are illustrated in Figure 7.

[3] Caterpillar Olympian Gas Generator Model G100 F-3.
[4] Century Corporation Absorption Chiller Model AR-D30L2.
[5] WTT America Inc.

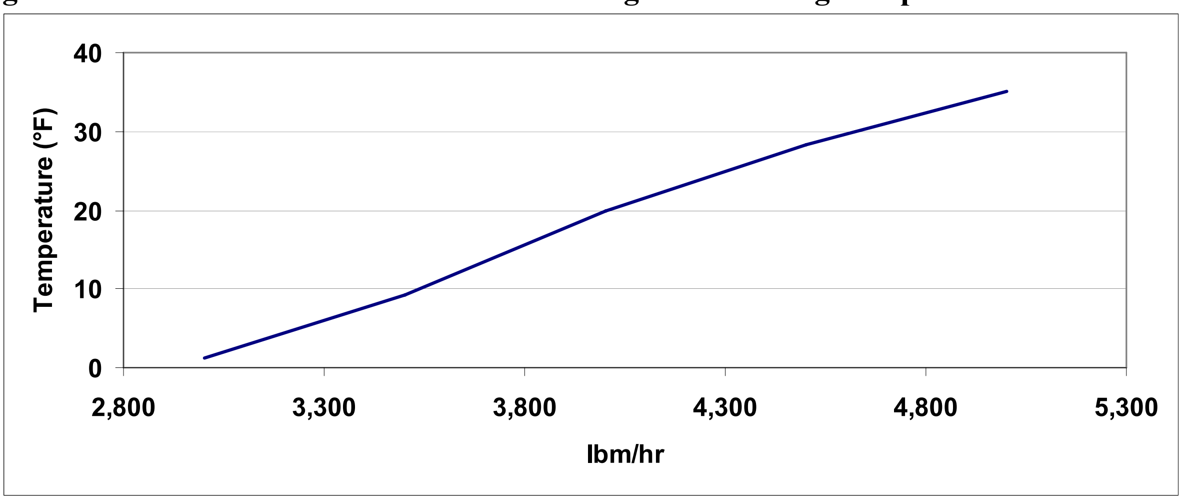
Figure 7. Engine-Generator Performance Curves



The absorption chiller and generator capacities were established based on the maximum allowable subcooling load (29 Tons), given the maximum refrigeration load (195 Tons) and system rated performance. The simulation model developed for this study allows the hourly subcooling load to dictate the absorption chiller output and generator thermal output, which in turn, determines the electric output of the generator. Using this sequence ensures that all of the energy produced from the generator (thermal and electrical) is utilized. The corresponding maximum thermal and electric outputs of the generator are 499,031 Btu/hr and 60.4 kW, respectively. The subsequent maximum fuel input rate (LHV) is 820,550 Btu/hr.

The hourly subcooling load is limited by the preset evaporating temperature of 20 °F. The performance curve for the countercurrent water-to-vapor subcooler is shown in the next chart, Figure 8.

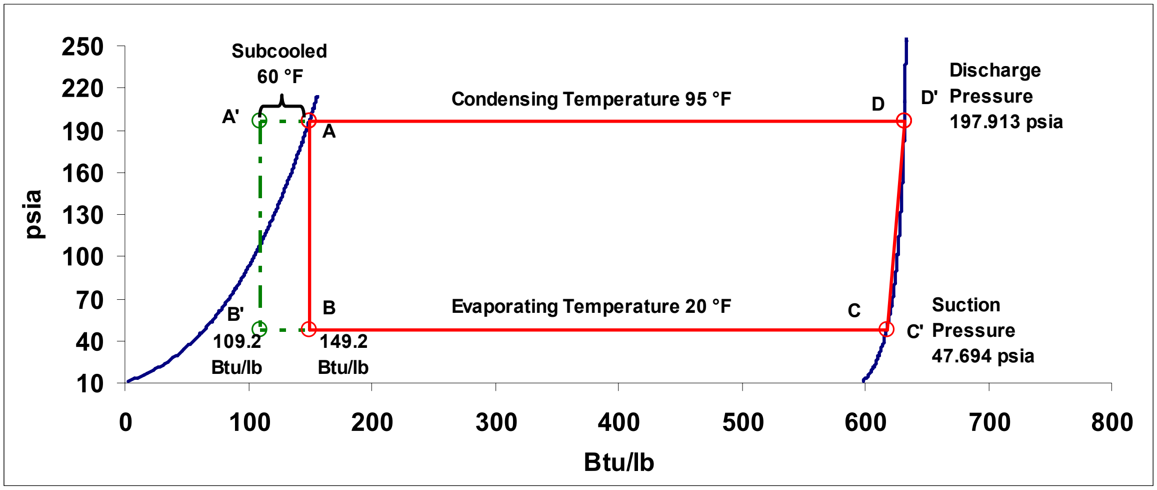
Figure 8. Subcooler Performance Curve – Refrigerant Leaving Temperature vs Mass Flow



One can see that the refrigerant leaving temperature (°F) on the hot-side of the subcooler varies as a function of refrigerant mass flow (lb/hr).

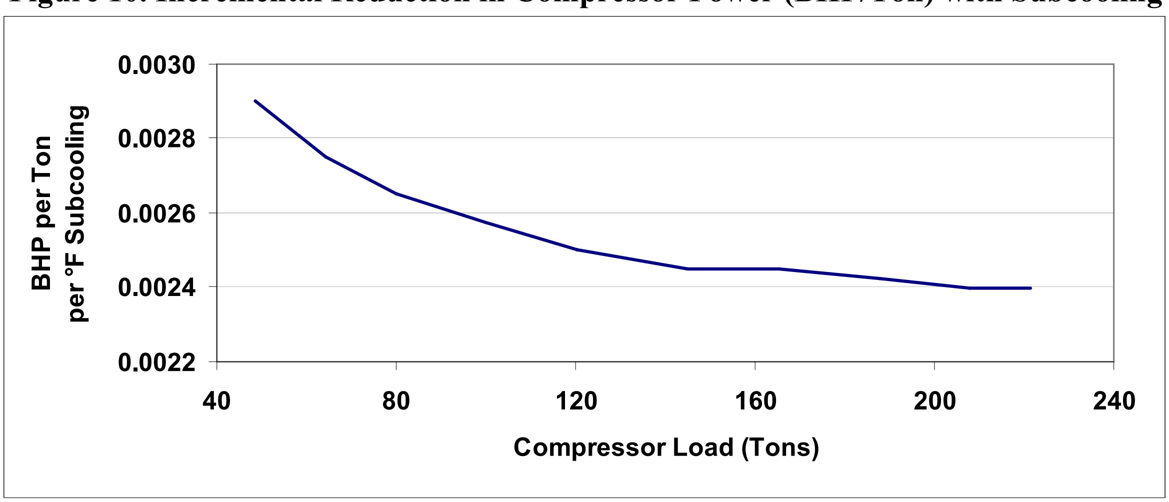
The simulated state conditions of the R-717 vapor compression cycle with subcooling are illustrated in the following pressure vs enthalpy diagram, Figure 9.

Figure 9. R-717 Vapor Compression Cycle With Subcooling - Pressure vs Enthalpy



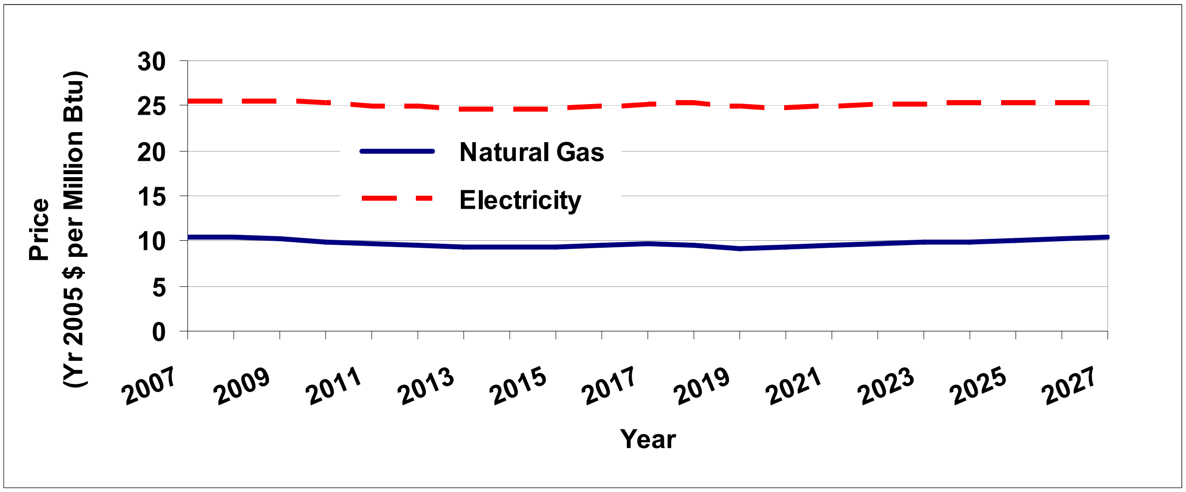
Essentially, subcooling reduces the specific enthalpy of the refrigerant from 149.2 Btu/lb at Point A to 109.2 Btu/lb at Point A', while reducing load on the compressor. Since the specific enthalpy state is lower entering the evaporator after subcooling, the required refrigerant mass flow (lb/hr) through the evaporator is less, given the same evaporator load (Btu/hr). For instance, at the peak refrigeration load (195 Tons) the required mass flow through the evaporator reduces from 4,983.5 lb/hr to 4,257.6 lb/hr when the refrigerant is subcooled. Consequently, the compressor power requirement is lowered from 210.5 BHP to 181.9 BHP. This equates to a 13.6% improvement in system efficiency for the entire refrigeration cycle (1.079 BHP/Ton to 0.933 BHP/Ton). As seen in the next chart, Figure 10, subcooling is more beneficial at part load when the compressor's operating efficiency is lower. The incremental reduction in compressor power consumption per ton of refrigeration produced is clearly greater at part load than at full load – over 16% more at 40 Tons than at 220 Tons.

Figure 10. Incremental Reduction in Compressor Power (BHP/Ton) with Subcooling



Life cycle cost. A twenty-year life cycle cost analysis was performed for both the base case and the proposed integrated system (CHP and absorption subcooling) using a discount rate of 6%. The economic model includes equipment installation costs of actual systems (CASGIP 2001-Present)[6], electric and gas utility rate schedules[7], 20-year energy price forecasting (EIA 2007). Shown in Figure 11, the twenty-year forecast of electricity and natural gas retail prices (in 2005 dollars) appears to be fairly flat over the evaluation period.

Figure 11. Energy Price Forecasts



Results

Hourly energy consumption and monthly and annual costs were evaluated for both the base case system and the proposed integrated system. In the following chart, Figure 12, the reduction in hourly energy consumption of the refrigeration system with subcooling is evident. Also plotted on the same chart is hourly grid electricity needs after considering onsite generation.

The simulated annual load factors for the engine-generator, absorption chiller and screw compressor are 77.3%, 80.7% and 81.4%, respectively. Engine-generator and absorption chiller ancillary electric loads were considered. The next chart, Figure 13, shows the annual benefit, cumulative benefit and cumulative benefit less capital for the proposed system.

[6] Initial capital cost estimates for the integrated system include the engine-generator - $165,622 (includes subcooler) and absorption chiller - $27,737. These are reported as installed costs.
[7] Small Industrial TOU customer in San Diego Gas & Electric Company's service territory.

Figure 12. Simulated Refrigeration System Energy Consumption

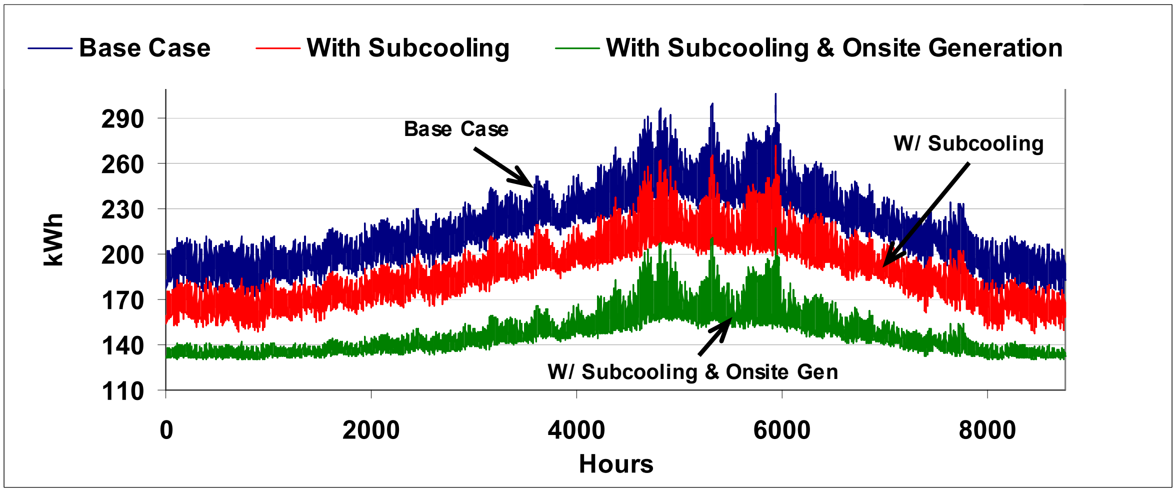
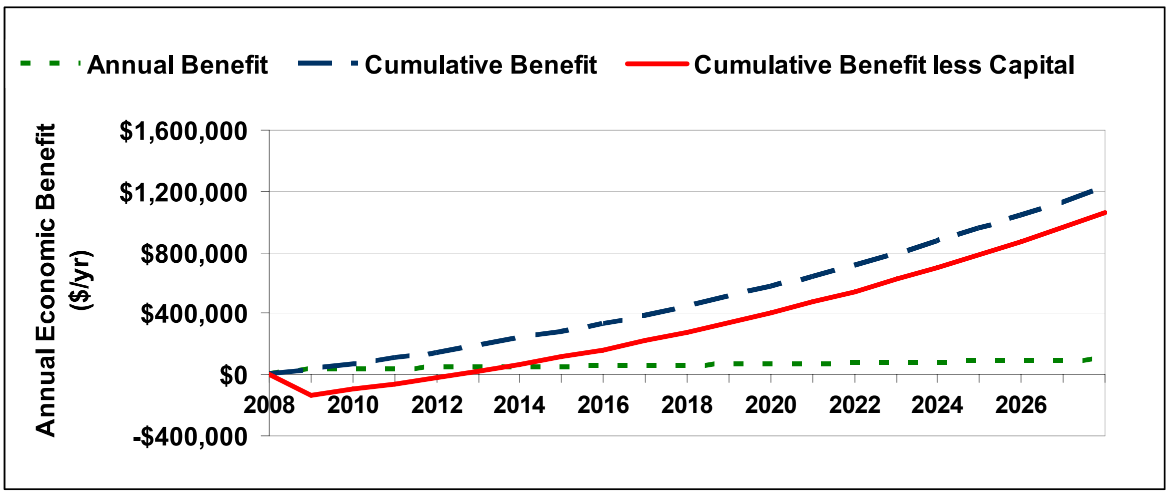


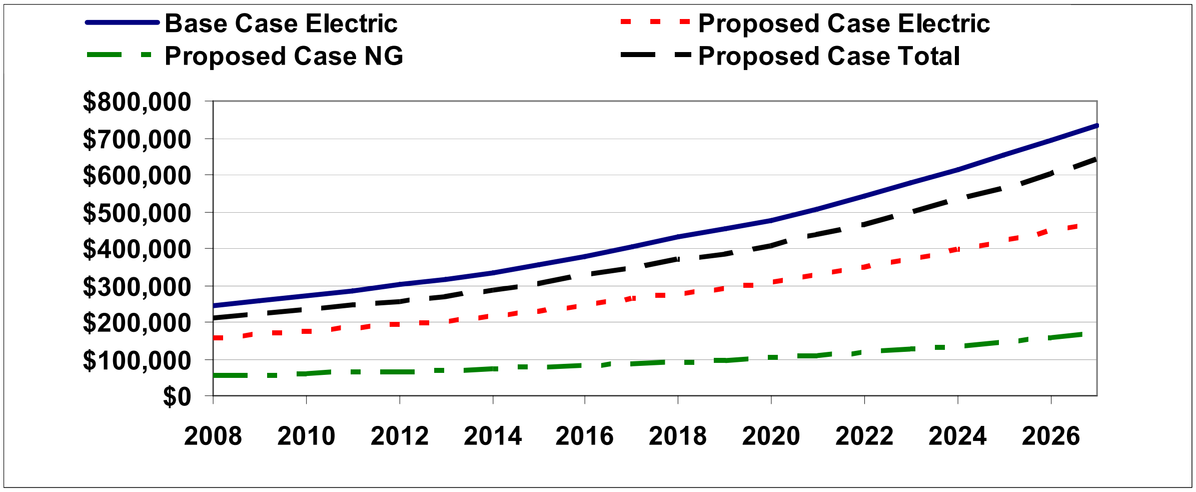
Figure 13. Cost Benefit Results



The net present value (NPV) of the base case over a 20 year life (2008 through 2027) is $4,793,059, and for the proposed integrated system the NPV is $4,319,385. In 2008, the expected levelized energy costs for the base case and proposed system are $0.123/kWh and $0.107/kWh, respectively. The generator's average fuel cost during the first year of operation is anticipated to be $0.95/Therm. The levelized O&M cost for the integrated system (CHP and absorption chiller) is estimated to be 0.005/kWh. The simple payback period for the proposed system is estimated to be just over 4 years. As you can see by looking at the previous chart, the cash flow (depicted by the red line) becomes positive near the middle of 2012.

Other costs related to modifying the existing refrigeration system to be compatible with the new equipment were assumed to be negligible. Annual electric and natural gas costs for both case studies are shown in Figure 14.

Figure 14. Annual Electric and Natural Gas Costs (Nominal $)



Additional detailed simulation results are summarized in Table 3.

Table 3. Detailed Simulation Results

Component	Refrigeration Only	W/ Subcooling Only	W/ CHP & Subcooling
Facility Grid Load (kWh/yr)	1,991,767	1,304,869	902,338
Facility Peak Grid Demand (kW)	305.4	277.6	217.2
Onsite Generation (kWh/yr)	N/A	N/A	402,531
Generator Peak Output (kW)	N/A	N/A	60.4
Generator NG Consumption (Therms/yr)	N/A	N/A	56,233
Generator Thermal Output (Therms/yr)	N/A	N/A	35,293
Subcooling Load (Ton-hrs)	N/A	205,878	205,878

Conclusions

In closing, the following conclusions can be drawn from this study:

- Employing both CHP and absorption cooling at a cold storage facility is shown to be feasibly cost effective. The NPV savings for the proposed integrated system when compared to the base case is estimated to be nearly 18%.

- Simulation results show an increase of 13.6% in the electrical efficiency of the entire refrigeration cycle when both CHP and absorption cooling are utilized.

- Increased compressor efficiency as a result of subcooling is shown to be higher at part load, over 16% more than at full load. This would mean greater efficiency improvements and economic benefits for systems having a lower annual load factor. In other words, systems operating much of the time at part load.

- Keeping in mind the previous point, one can assume that the improvement in electrical efficiency would be greater for refrigeration systems with compressors having a higher BHP/Ton full load rating than the compressor evaluated in this study – 1.073 BHP/Ton. Following this same premise one can also assume that the benefits would be enhanced for compressors having been in service for an extended period time. The logic being that compressor efficiency degrades over time.

- Refrigeration systems having a lower evaporating temperature than 20 °F should experience greater improvements in efficiency with subcooling than what is presented in this study. This is true because the potential for subcooling, which is based on the difference in condensing and evaporating temperatures, should increase as the evaporating temperature decreases, given the same condensing temperature. Therefore, more compressor work can be avoided.

- Further study on this topic should focus on several areas: 1) measuring performance of an actual refrigeration system at a cold storage facility similar to the case site evaluated in this study, 2) using measured data to calibrate the model developed in this study, 3) working with experts in generation, absorption cooling and refrigeration to better refine economic assumptions and cost data used in this study, 4) using calibrated model to identify worthy candidate sites for implementing CHP with absorption cooling 5) identifying and characterizing "packaged" integrated systems for various types of cold storage facilities, including engineering design and cost information.

References

[ASHRAE] The American Society of Heating, Refrigerating and Air-Conditioning Engineers. 1997. *Fundamentals Handbook*. Atlanta, Georgia: The American Society of Heating, Refrigerating and Air-Conditioning Engineers.

[CASGIP] California Self-Generation Incentive Program, California Public Utility Commission. 2001–Present. http://www.sdenergy.org/ContentPage.asp?ContentID=279&SectionID=276&SectionTarget=35. Sacramento, California: California Public Utility Commission.

[CIMIS] California Irrigation Management Information System. 1982. http://wwwcimis.water.ca.gov/cimis/data.jsp. Sacramento, California: California Department of Water Resources, Office of Water Use Efficiency.

[DOA] Department of Agriculture. 2004. *Capacity of Refrigerated Warehouses*. Washington, D.C.: U.S. Department of Agriculture.

[EIA] Energy Information Administration. 2007. *The Annual Energy Outlook 2007 (DOE/EIA-0383)*. Washington, D.C.: U.S. Department of Energy.