Performance of an Engine-Driven/Electric Motor Hybrid Refrigeration System Integrated with Desiccant Dehumidifiers in an Ice Rink

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ABSTRACT

The US Department of Energy has embarked on significant partnerships with industry with the intent of creating meaningful research, development and commercialization roadmaps. The natural gas industry and manufacturing trade allies[1] concluded that significant technology synergy exists to increase energy efficiency, lower emissions and improve operating economics in the commercial and institutional building sector. The Office of Distributed Energy Resources[2] facilitates the Cooling, Heating and Power (CHP) research, development and implementation focusing efforts on onsite power generation and HVACR technologies for integration in building and industry.

CHP systems are being examined, developed and promoted as an economically viable integrated power and HVACR system approach to environmentally responsible buildings. CHP systems can offer significant improvement in energy efficiency through the useful recovery of heat generated during energy conversion.

Using field data enhances the ability to project coupling of engine-driven equipment with desiccant dehumidification systems as a CHP technology and enables better prediction of national building energy trends in the future.

An indoor ice arena, in Blaine Minnesota, opened in October 1998. This arena holds four Olympic-sized ice sheets, making it one of the largest ice facilities in the United States. Each rink has permanent seating for 400 people plus temporary seating for an additional 200.

The refrigeration system consists of two electric motor-driven 8-cylinder reciprocating compressors, one motor-driven 16-cylinder reciprocating compressor, and one natural gas engine-driven 16-cylinder reciprocating compressor. The basic system design is chilling calcium chloride brine to 14.4 °F to freeze the ice sheets. The heat is recovered from engine jacket cooling water and also from the engine exhaust. This heat energy is used to preheat the reactivation air for the four desiccant units (one per sheet) for dehumidification and preheat boiler feed water for heating.

The site was monitored for performance during the fall/winter of 1999/2000 providing operating data, and economic calculations that offer useful information concerning CHP performance. Site data confirmed that the CHP system yielded the best primary energy efficiency performance, however there is room for improvement by:

1. Increasing amount of time the engine/heat recovery systems were operated by improving utility rate structures.
2. Increasing the amount of energy actually recovered through design changes.
3. Reducing cost of maintenance and diminishing maintenance uncertainty.

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[1] Manufacturing trade allies were in the onsite power generation and thermally activated heating, ventilating, air conditioning and refrigeration industry.

[2] Newly formed entity within the Office of Power Technologies

Ice Rink Description

The NSCF built an indoor ice rink complex (see Figure 1) that opened in October 1998. This arena holds four Olympic-sized ice sheets, making it one of the largest ice facilities in the United States. Each rink has permanent seating for 400 people plus temporary seating for an additional 200. The facility also includes: 16 locker rooms, two ice resurfacer rooms, a centrally heated spectator area, cafe space on the second level overlooking all four ice sheets, and space for a pro shop.

An extensive energy analysis led to a recommendation for a hybrid (gas and electric) refrigeration system design.

Reliant Energy – Minnegasco, the Minneapolis-based gas utility serving Blaine, has a poor summer load factor on its distribution system. Therefore, it was interested in developing a database involving CHP systems to improve system wide load factors while at the same time saving energy, reducing CO_2 emissions and saving the customer money.

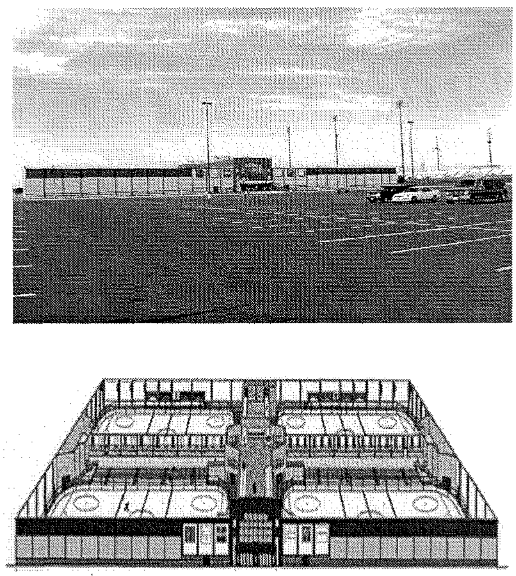


Figure 1. Ice Rink Building Design

Rink Energy Equipment Design

Each of the four Olympic-sized rinks is identical. The refrigeration system, as seen in Figure 2, consists of two electric motor-driven 8-cylinder reciprocating compressors, one motor-driven 16-cylinder reciprocating compressor, and one 16-cylinder natural gas engine-driven reciprocating compressor to maintain the ice sheet constant temperature. The R-22 refrigeration compressors, operating at 10 °F suction temperature, allow the chiller to supply brine at 14.4°F to the rinks, with a total capacity 495 tons.

The mechanical equipment room is located in the rear of the building, aligned with the central corridor. Figure 3 shows the general equipment room layout. Heat from the engine is recovered to preheat boiler water for space heating and regenerate four desiccant units that dehumidify and heat the rink areas. A plate and frame heat exchanger is used to transfer engine jacket water heat to the heat recovery loop, and a combination exhaust silencer/heat recovery exchanger provided additional energy to the recovery loop. (both devices are depicted in Figure 3)

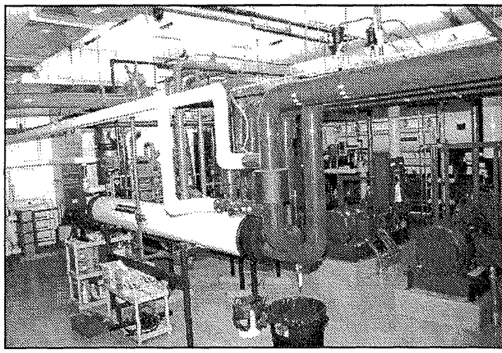


Figure 2. Mechanical Equipment Room

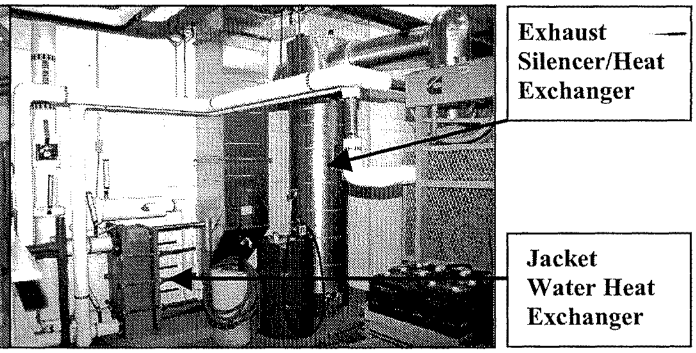


Figure 3. Heat Recovery System

The refrigeration system was field-erected by the installing refrigeration contractor. Typical of field-erected systems, a good amount of flexibility in piping layout is permitted - tailoring the space to accessibility and maintenance.

Figure 4 shows the engine/compressor driveline, with the speed reducing gear (1,700 rpm engine speed to 1,200 compressor speed), mounted within the main driveline base plate driving a 16-cylinder R-22 refrigeration compressor.

The remainder of the natural gas equipment consists of four desiccant dehumidifiers with heat recovery pre-heaters, three boilers, to provide supplemental hot water, and a natural gas generator to provide backup electricity in case of an electrical power failure. It is interesting to note the rink uses two natural gas ice resurfacers and even a natural gas lawn mower[3].

During the dehumidification season, the boilers do not operate and the recovered heat energy goes only to the desiccant dehumidifiers to preheat the regeneration air stream. During the heating season, the recovered heat is used to heat offices and common areas in the building and to provide spectator comfort in each arena.

Hybrid System Energy Management Philosophy

Sequencing of the engine-driven compressor and the main electric motor-driven compressor is simple. The sequence follows the electric utility's peak demand schedule, which essentially runs weekdays from 4:00 pm to 10:00 pm.

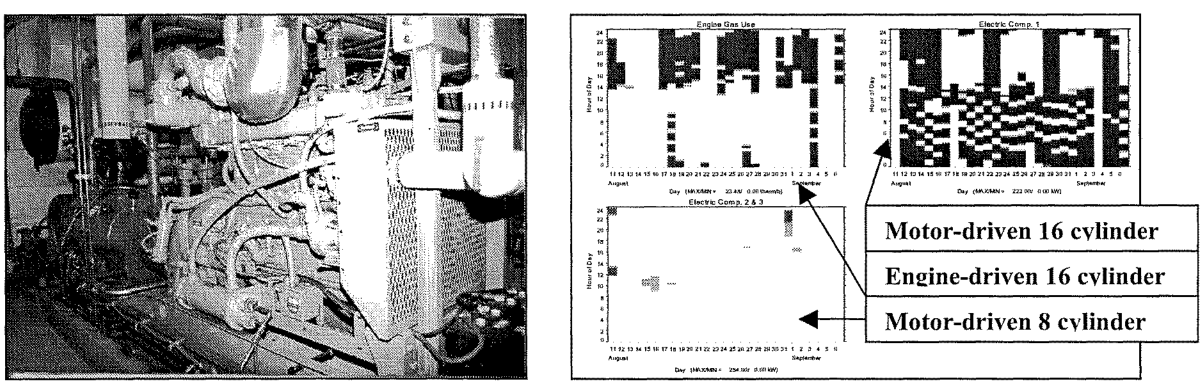


Figure 4. Engine-Driven Compressor Driveline

[3] The Natural gas resurfacers and lawn mower are fueled from an onsite fuel gas compressor.

The operating schedule is depicted on the right side of Figure 4. This shadow plot y-axis consists of 96 blocks per day (fifteen-minutes each) with 12:00 midnight existing on the x-axis. The x-axis contains about 30 days of operation with each vertical stripe representing a given day. The darker the block, the longer the compressor operated during a particular hour. Given this key, one quickly realizes that the engine-driven 16-cylinder compressor generally runs Monday through Friday from about 2:00 pm to midnight. The electric motor-driven 16-cylinder compressor runs Monday through Friday, largely from midnight to 2:00 pm, and most of the time Saturday and Sunday. The two electric motor-driven 8-cylinder compressors rarely are required. This operating sequence is driven by the rate structure of the local municipal utility (peak operating hours are weekdays 4:00 pm to 10:00 pm).

The essence of this paper is to explore the economic and energy efficiency implications of this system design and assess the impact of such cooling, heating and power (CHP) applications.

Rink Load Profiling

Energy use field data (typical data shown in figure 5 on the left) taken during September of 1999 provides ample statistical information to create performance algorithms for this compressor.

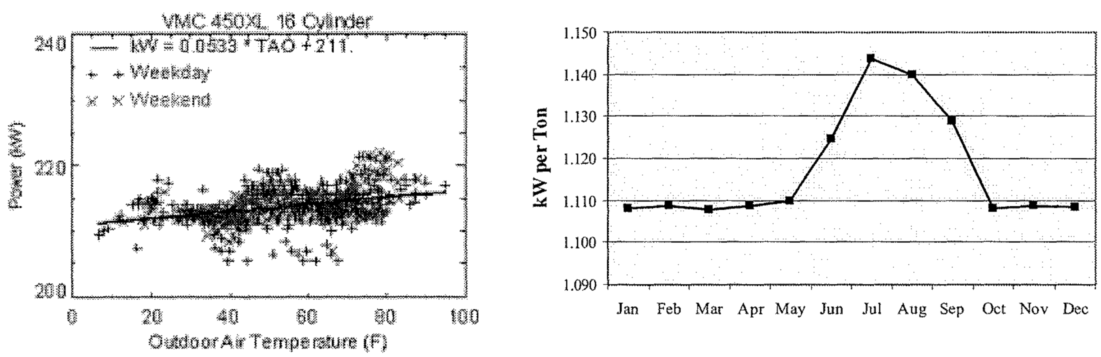


Figure 5. Sixteen-Cylinder Electric Motor-Driven Compressor Operation

Similar data from the other three compressors permits the completion of the algorithms:

8-cylinder electric motor-driven compressor:	*kWh/hour = (0.0202 * TAO + 109)*
16-cylinder electric motor-driven compressor:	*kWh/hour = (0.0533 * TAO + 211)*
16-cylinder engine-driven compressor:	*therms/hour = (0.0164 * TAO + 20)*

Using national weather service data for the Minneapolis region, the above algorithms can provide full-load performance information corresponding to the month of the year (graphically depicted in Figure 5-right). The control protocol for this installation is based on operating each compressor at full load and cycling the compressors on and off. When each compressor is running, it is operating at full load. Therefore, Table 1 depicts each compressor's annual performance.

Table 1. Compressor Full Load Performance at Maximum Monthly Temperature

Month	16 Cylinder Electric kW/Ton	8 Cylinder Electric kW/Ton	16 Cylinder Engine Therms/Ton-hour
January	1.109	1.083	0.117
February	1.108	1.084	0.116
March	1.108	1.084	0.116
April	1.108	1.083	0.116
May	1.110	1.086	0.117
June	1.121	1.098	0.119
July	1.146	1.111	0.120
August	1.144	1.096	0.119
September	1.131	1.103	0.119
October	1.108	1.084	0.116
November	1.108	1.084	0.117
December	1.109	1.085	0.117

Desiccant Dehumidification System Design

Before presenting the field data on the desiccant dehumidifiers at this particular site, it is important to understand the energy efficiency implications behind their selection. It should be noted that desiccant dehumidifiers are standard in all new professional hockey rinks and are generally installed in most year round rinks. The reason is simple and easily explained by examining unit operating data at Schwan's Super Rink[4] (Figure 6). The ideal task is to maintain the dewpoint of the air in the rink near or below the ice surface temperature, which generally ranges between 24 and 28°F depending on the type of skating (figure versus hockey), which means delivering air at very cold temperatures. This presents three real problems for conventional vapor compression equipment:

1. High capitol cost
2. High operating cost
3. Uncomfortable skating conditions

Therefore, it is generally assumed that desiccants are the base case design for ice rink applications, and there is no longer a need to demonstrate the energy savings of this approach over conventional vapor compression equipment.

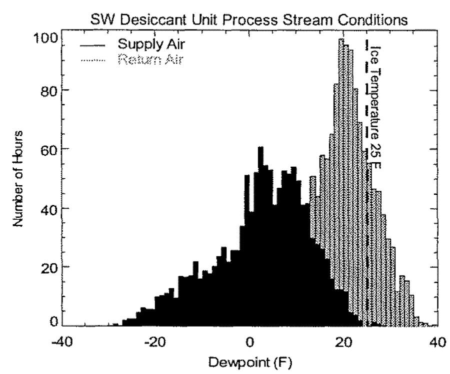


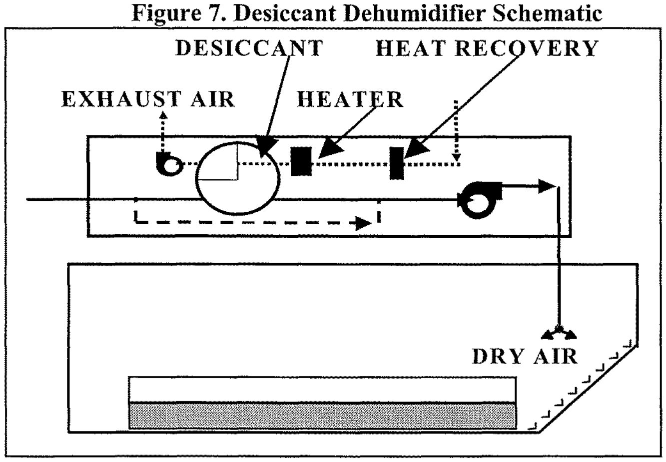
Figure 6. Desiccant Dehumidifier Humidity Conditions

[4] [IC] Industrial Center (IC). June 2000. Monitoring Report, Schwan's Super Rink, Blaine, Minnesota

Each rink is equipped with a dedicated desiccant dehumidification unit to eliminate condensation on cold surfaces and fogging, and allow for year-round operation of the facility (see Figure 8). Each desiccant unit includes a recovered energy coil (to recover thermal energy from the engine) as well as a direct-fired gas burner to regenerate the desiccant unit. The unit includes a desiccant wheel without a sensible heat exchanger since dehumidification and heating are simultaneously required (see Figure 7). The process fan in each unit runs continuously at low speed during unoccupied periods. During events the units are manually switched to high speed by the rink staff. Ventilation air needs are designed to accommodate 400 spectators and 20 ice skaters per rink. This yields the following calculation:

3,150 cfm = 420 people x 7.5[5] cfm per person
Total air circulated within each rink is 10,000 cfm to allow for temperature blending.

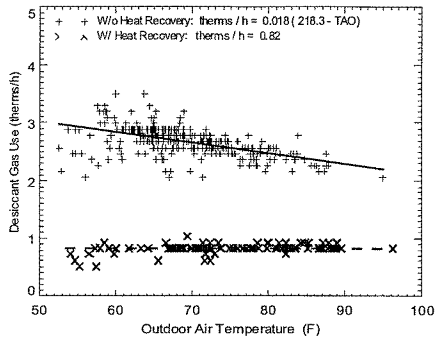
Figure 7. Desiccant Dehumidifier Schematic



Engine Heat Recovery Used for Dehumidification

The dehumidification units require 250 °F air to regenerate the silica gel desiccant. Heat recovery coils use a 30% ethylene glycol/water mixture at 180 - 205 °F supply from the engine thermal recovery system. A natural gas burner provides the temperature difference between ~170 °F air off the heat recovery coil and 250 °F air necessary for regenerating the wheel. The burner also operates to regenerate the wheel independently from the engine-driven compressor when it is not operating (i.e. off-peak operation). Field data (Figure 8 – left) shows natural gas use without heat recovery (+) and reduced gas use with heat recovery (x).

[5] Calculated on the basis of limited occupancy.



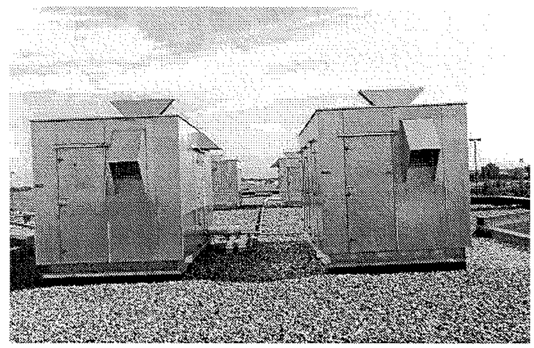


Figure 8. Desiccant Dehumidifier Gas Use (Left) and Units on Roof (Right)

Using the field data, the following algorithms can be developed to represent desiccant energy use and recovered energy in terms of actual natural gas energy savings. Using the algorithms and hourly weather data, Table 2 is constructed showing monthly energy use.

*Without Heat Recovery = 0.018 * (218.3 – TAO[6]) therms/hr*
With Heat Recovery = 0.82 therms/hr

Table 2. Desiccant Dehumidifier Gas Use per Month

Month	Desiccant Gas Use Without Heat Recovery (Therms)	Desiccant Gas Use With Heat Recovery (Therms)	Desiccant Gas Use Heat Recovery Savings (Therms)
January	-	-	-
February	107	107	-
March	729	478	251
April	3,018	2,230	788
May	4,158	3,278	880
June	5,539	4,482	1,057
July	6,574	5,420	1,154
August	6,463	5,328	1,135
September	5,631	4,537	1,094
October	3,530	2,751	780
November	1,715	1,139	576
December	423	250	173
Total	**37,887**	**30,000**	**7,888**

Engine Heat Recovery Used for Boiler Pre-Heating

The results of the regression analysis, from Figure 9's data, showed that the boiler gas use was reduced by 0.18 therms for each therm of engine gas consumption, implying that the heat recovery system is 18% efficent in the winter.

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6 TAO = Temperature air outside

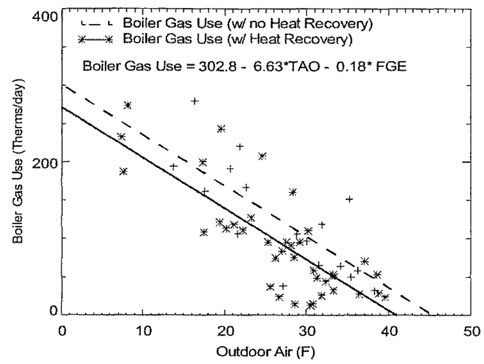

Figure 9. Boiler Gas Use

Table 3 shows monthly boiler energy use projections based on the imperical data taken at the site.

Table 3. Boiler Gas Use per Month

Month	Boiler Gas Use Without Heat Recovery (Therms)	Boiler Gas Use With Heat Recovery (Therms)	Boiler Gas Use Heat Recovery Savings (Therms)
January	7,288	6,740	548
February	5,223	4,695	528
March	3,654	3,035	619
April	464	271	193
May	26	7	19
June	-	-	-
July	-	-	-
August	-	-	-
September	1	-	1
October	258	113	145
November	2,398	1,888	510
December	5,648	5,038	610
Total	**24,959**	**21,786**	**3,173**

Thermal Recovery Efficiency

Finally, Figure 10 looks at the actual recovered thermal energy versus potential recoverable energy at 50% and 70% recoverable efficiencies; one can see room for improvement. In part, inefficiencies come from lack of coincident heating season loading, where heat is required to protect the building during very cold nights when the engine does not operate. More energy could be recovered by increasing the temperature of the recovery loop and by adding thermal storage. However, thermal storage would have to be measured against additional cost of such design modifications.

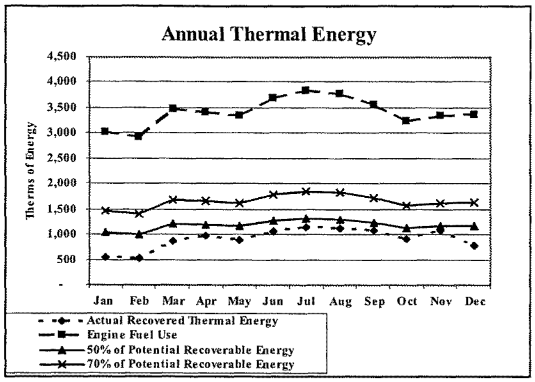

Figure 10. Thermal Energy Recovery, Actual versus Potential

This focuses the designer on a critical element of CHP design. Integrating energy recovery systems requires detailed assessment of the source load profile, as well as, the various load profiles regarding coincident heat and temperature requirements.

Optimizing Energy Management Strategy to Primary Energy Use

The next level of analysis is to understand the primary energy and economic implications of the combined refrigeration/ventilation air conditioning system, and to test if the present schedule of operation is both energy and economically efficient. The results found in Table 4 show that, in all operating modes, the gas engine-driven hybrid/heat recovery system will consume less primary energy. This is simply a result of utilization of energy recovery. Without energy recovery, the engine-driven compressor performance is very close to the electric motor-driven compressor performance.

The most efficient energy management strategy of the three options is operating the engine-driven compressor/heat recovery system 24 hours per day.

Table 4. Annual Primary Energy Use of Various Operating Scenarios

	9-Hour Case MMbtu	15-Hour Case MMbtu	Gas 24-Hour Case MMbtu	Electric 24-Hour Case MMbtu
Jan	902	890	814	952
Feb	849	836	761	897
Mar	968	946	864	1,049
Apr	985	965	902	1,078
May	1,065	1,043	961	1,149
Jun	1,077	1,046	925	1,177
Jul	1,144	1,107	952	1,253
Aug	1,156	1,118	960	1,263
Sep	1,029	996	880	1,133
Oct	1,044	1,024	951	1,131
Nov	940	913	837	1,043
Dec	914	896	822	987
	12,075	11,780	10,628	13,113

Optimizing Energy Management Strategy to Operating Cost

The economic outcome is as expected when one understands the electric and gas rate structures. As can be seen in Table 5, the lowest operating cost strategy is to run the engine-driven compressor during the peak electric rate block using the off-peak electric rider. The striking conclusion from Table 5 is the non-compressor related impact of the general service rate contributing $6,187 annually to the all-gas engine operation, while selecting the optimum off-peak rider for the 9-hour hybrid operations yields only $100 of building related savings.

Table 5. System Operating Cost Comparison

	9-Hour Weekday Operation		15-Hour Weekday Operation		Gas Engine 24-Hour Day Operation		All Electric Operation
	Off-Peak Rate Operation w/o HR	Off-Peak Rate Operation with HR	Off-Peak Rate Operation w/o HR	Off-Peak Rate Operation with HR	Standard Rate Operation w/o HR	Standard Rate Operation with HR	Standard Rate Operation w/o HR
	(Dollars)	(Dollars)	(Dollars)	(Dollars)	(Dollars)	(Dollars)	(Dollars)
Total Energy Cost	$53,060	$49,764	$53,538	$49,161	$55,119	$46,802	$67,766
Base Rate Savings	$(100)	$(100)	$(100)	$(100)	$(6,187)	$(6,187)	
Maintenance Cost Differential	$4,608	$4,608	$6,517	$6,517	$13,721	$13,721	$ -
Operating Cost	$57,569	$54,273	$59,955	$55,578	$62,653	$54,336	$67,766
Savings Gas Versus Electric	$10,196	$13,492	$7,810	$12,188	$5,113	$13,430	$ -

The off-peak rider formula will elicit various economic results depending on the size relationship of the marginal electric use being analyzed and the remainder of the load. The basis for the model demand calculation is 615 kW off-peak and 400 kW on peak (when gas is used) and 615 kW (when all electric). This yields an on-peak to off-peak ratio of 65%. Note that the optimal ratio would be .70/1.15 = 61% since the higher number prevails in the billing calculation. Assuming the marginal load being assessed created the difference between on-peak and off-peak demand, then, the impact of the building is maximized. (See Figure 11)

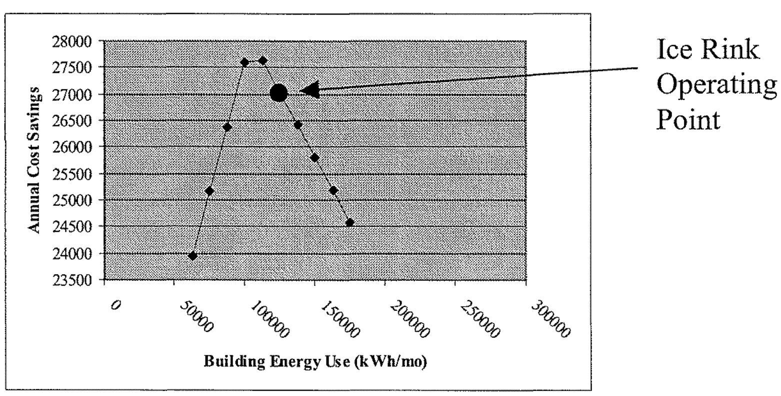

Figure 11. Impact to Building Size on Energy Savings

Operating the hybrid system, with the engine operating 9-hours per weekday, yields the best economic performance, however it is only by $62 over the all gas operation (Figure 12). It should be noted that the off-peak rider also increases the penalty should the engine-compressor not be able to perform. Therefore, the trade-off becomes one of operating in the hybrid mode and the likelihood of incurring a penalty for engine non-availability in any given month or operating in the all gas mode 24-hours per day and remaining within the annual maintenance budget.

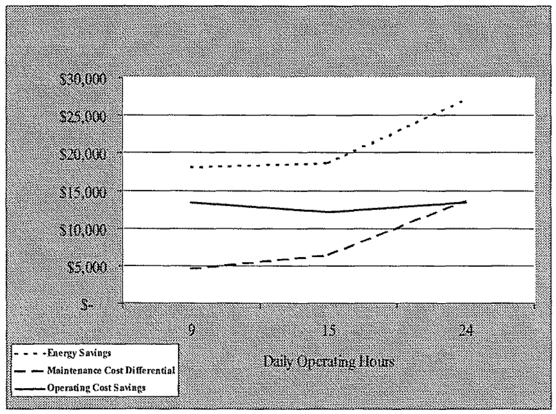

Figure 12. System Operating Cost Comparison with Heat Recovery

The future financial implications of achieving CHP optimum primary energy will depend upon:

1. Improving durability and reducing maintenance cost and risk for 24-hour per day engine operation.
2. Improving thermal energy recovery efficiency.
3. Linkage between primary energy and emissions being made for all end uses.

National Energy Impact

The impact of running the gas engine 24-hours per day on this four-sheet ice rink is a primary energy reduction of 2,500,000,000 Btuh. Over a normal 20-year life projection this would yield a primary energy reduction of 50 billion Btus. Extrapolating this one site across the country by retrofitting the population of 3,000 rinks over the course of the next 10 years would achieve annual primary energy savings 12.5 billion Btus (25 billion Btus for twenty years of operation). This modest amount of energy savings is significant when examining only 3,000 sites. Of course there are many other applications combining engine operation with heat recovery and dehumidification that can benefit from this case study. The lessons learned from this site are threefold:

1. The limited amount of time the engine/heat recovery systems were operated due to the utility rate structure and maintenance uncertainty.
2. The relatively small amount of energy actually recovered.
3. The cost of maintenance for on-site energy conversion remains a significant issue.

Rate Structure Design and Energy Efficiency

The natural question is: can rates be designed that improve the efficiency of CHP operation, improve economic payback for customers and increase gas and electric utility profitability – all at the same time? Moving this rink to a different utility region would have a significant effect on the economics of the rink operation and a profound effect on the financial incentive to move forward on such projects.

For example, running the ice rink model in New York City provides an interesting comparison and perhaps provides insight into a broader range of applicability for engines and heat recovery systems in ice rinks. The compressor schedule has been shifted to match the demand schedule of the Consolidated Edison SC9-3 utility rate. The engine operates weekdays starting at 8:00 am and runs to 6:00 PM in the 8-hour case, and 10:00 PM in the 14-hour case. Following engine operation, the 300 hp electric compressor operates up until midnight on weekdays (as required to meet the measured refrigeration load), and all day on weekends. The 150 hp electric compressor operates from midnight to 8:00 am weekdays.

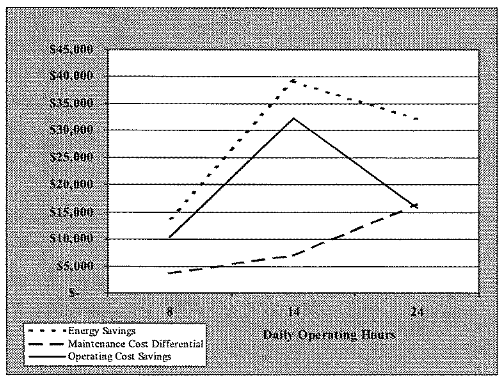

Figure 13 NYC Operating Cost w/o Heat Recovery

In this case energy savings mimicked the on-peak electric charges. Here the 14-hour operating parameter was the better operating match to the electric rate structure, showing a pronounced energy savings peak. Figure 13 reveals energy savings at the 14-hour per weekday mark at $33,491 which, when applied to the incremental engine driveline costs of $73,000, yields a simple payback of 2.2 years. Figure 14 shows that adding heat recovery improves the simple payback to $88,209/$42,319 = 2.1 years and, like the Minnesota case, the energy cost for the 24-hour gas case become the optimum energy cost point.

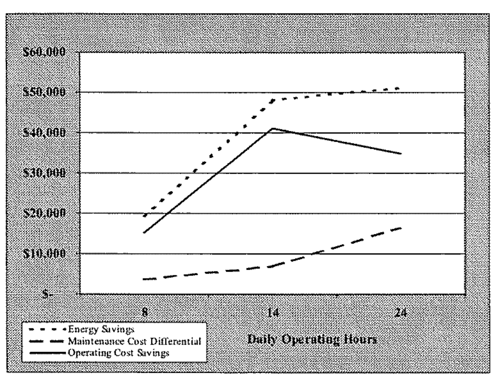

Figure 14 NYC Operating Cost with Heat Recovery

New York operation almost triples the annual savings shown in Minnesota, and would likely lead to increased utilization. Therefore, financially speaking, this leads one to conclude that this CHP approach would be more likely to succeed where commercial rate structures are more robust, such as in New England, California or elsewhere in the Midwest, and where it is structured to strike a balance between energy cost and engine maintenance, as seen in the operating cost curve of Figure 14. If peak power becomes more limited and restructuring produces cost-based pricing of electricity, then the scenario described above will yield an even higher payback to the customer, provide more profitability to the gas utility and improve load factor and profitability of the electricity utility.

Recovered Energy Improvement

The 24-hour engine operation successfully recovers 2,791 MMBtus annually. This is 26% of the total engine fuel input for the 24-hour-a-day operation or about 30% of the available heat energy. Increasing the recovery efficiency can be imagined through the use of systems modification to raise recovery temperature and thermal storage, with a potential of 1,452 to 3,150 MMBtu in additional annual energy savings (see Table 6 for details). This would yield $4,330 to $9,387 in additional annual savings.

Table 6. Energy Recovery

	Measured useful heat recovered for heating and dehumidification MMBtu	Assume 69.5 % Btuh of fuel input leaves engine as heat energy and 50% HR efficiency MMBtu	Assume 69.5 % Btuh of fuel input leaves engine as heat energy and 70% HR efficiency MMBtu
1/1/1999	161	308	431
2/1/1999	157	290	406
3/1/1999	210	339	475
4/1/1999	201	349	488
5/1/1999	214	372	520
6/1/1999	279	381	533
7/1/1999	331	406	568
8/1/1999	332	408	572
9/1/1999	280	367	513
10/1/1999	207	366	513
11/1/1999	231	338	473
12/1/1999	189	320	448
Total 1999	**2,791**	**4,244**	**5,941**
% Increase Available	**Base**	**52%**	**112%**

Maintenance Cost Improvement

The increase in maintenance cost is directly related to full-load engine run hours. More research into improvement of engine routine maintenance could yield important cost improvements such as: improved plug life, extended wire and belt life and longer oil service. New maintenance paradigms, like increased use of electronics and the Internet to anticipate and dispatch service on a just-in-time basis, will significantly enhance customer economics and reduce risk. Maintenance will significantly improve the drive toward primary energy optimization.

Implications to Other CHP Systems

It is clear from the results of this site that future CHP projects will benefit from stronger systems integration. For example, building modular desiccant systems to include engine-driven chillers and pumps could provide close-coupled systems for heat recovery, as well as, providing important redundancy (having one compressor ~ 40 HP per system) supplying chilled brine to service one sheet of ice. Having each chiller feed into a common supply header and drawing from a common return header would yield good turndown and adequate redundancy. These units could be factory-made for high quality and low cost, and be roof-mounted to eliminate the need for a mechanical equipment room.

An ideal example of the future potential of such an integrated device would be a microturbine packaged within a desiccant system where the exhaust of the microturbine directly regenerates the desiccant wheel, further improving the efficiency and reducing the integration costs.

Nevertheless, it is important to find closer links between components and controls. A great improvement would be to use desiccant materials that can adsorb or absorb at lower temperatures. Engine jacket water typically is in the range of 180 - 200 °F and exhaust heat is usually recovered ~ 210 °F in non-pressurized systems, and 260 to 270 °F is easily achieved in pressurized systems. Using silica gel that is optimally regenerated at 250 °F air temperature generally requires 260 – 270 fluid temperature. The limiting factor at this site was not the ability of the heat recovery system to achieve 270 °F supply temperature. The limiting factor was a temperature limit on the gas boilers. This could have been avoided by piping the engine dump radiator between the desiccant units and the boilers and setting the dump radiator at the 180 °F boiler temperature limit. This change could have a significant impact on energy recovery efficiency. Also, having no thermal storage also means operating the desiccant burner when the engine is not operating.

Improved retrofit strategies using electric motors close-coupled to on-site electric power might increase market penetration. There may be an optimum amount of power to constantly generate for the rink, which is probably on the order of 200 kW. The maximum demand for the refrigeration system is about 215 kW, and the building uses another 400 kW on-peak for lighting, fans and other electric devices. By generating electric power 24/7, then heat recovery is always available. With new low temperature desiccants, or using high temperature exhaust from microturbines or fuel cells, in the future significant additional energy savings are possible.

The implications of these results to other applications are simple and direct. The success of CHP will depend on optimal matching of certain elements, which may or may not be within the preview of the designer.

1. If at all possible, base load the energy conversion device.
2. Match recovered thermal energy temperatures to end-use requirements and consider carefully design limitations that hinder full use of the exergy available.
3. Prime mover maintenance is a key parameter in the purchase decision and maintenance contract, and performance contracts are the likely tools for success.
4. Factory integration of CHP systems is essential for significant market penetration.

References

[IC] Industrial Center (IC). June 2000. Monitoring Report, Schwan's Super Rink, Blaine, Minnesota.