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Thermal Refrigeration Cycles for Small Gas Turbine Power Augmentation

By

Rob Brandon P.Eng
Bryan Halliday Eng
Daniel Bolinu
Of
CANMET Energy Technology Centre
Natural Resources Canada
1 Haanel Drive, Ottawa, K1S 3S2

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THERMAL REFRIGERATION CYCLES
FOR SMALL GAS TURBINE POWER AUGMENTATION

BIOGRAPHIES

Rob Brandon, C.Eng P.Eng

Rob Brandon is a professional engineer who has worked in energy engineering over the last twenty years for a number of public and private organizations, including Rolls Royce, the AES Corporation and the Prince Edward Island Energy Corporation. He graduated from the University of Southampton in 1972. He joined the Community Energy Systems group of the CANMET Energy Technology Centre in 1994. He was the founding president of the Canadian District Energy Association, and has served as Secretary of the Canadian Wind Energy Association. He currently manages the Distributed Generation program out of the CANMET Energy Technology Centre.

Bryan Halliday, Eng.

Bryan Halliday, a 1982 graduate of McGill University, is a professional engineer who worked for ten years as a project engineer for private firms such as Domtar, Forbo and Peerless Clothing, and for five years as an energy engineer for Bombardier, JL deBall Canada and Sofame-Tech. He joined the Federal Industrial Boiler Program of the CANMET Energy Technology Centre in 2001, and since 2003 has been project engineer in the Distributed Generation program at CANMET.

Daniel Bolinu

Daniel Bolinu graduated in 1984 from a five year power engineering program at the Polytechnic Institute of Bucharest. He has worked for 11 years in power generation, mostly in power plant performance for the National Electricity Company of Romania (RENEL). He also worked in the energy field for Chiyoda Corp. Japan and City of Hull (Quebec). He joined the Community Energy Systems group of the CANMET Energy Technology Centre in 2000 where he has been working on district energy system projects and on microturbine applications.

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ABSTRACT

The CANMET Energy Technology Centre in Ottawa has been working with microturbine technology for over four years. Field experience is being obtained at a number of site; the applications are focused on combined heat and power as well as the use of waste fuel.

A recent CETC initiative has been to examine and rig test unconventional thermal cycles using low grade waste heat from gas turbine or reciprocating engine plants to produce refrigeration effects. The premise of the work was to re examine simple thermal refrigeration cycles that had the potential for low installed capital cost and might be economic for smaller power generation with surplus summer exhaust heat.

The paper used the availability of hourly power price and temperature data in the Toronto area for the summer of 2002 to investigate the economic performance of chilling systems using waste heat. The study determined the installed capital cost of any chilling system that would be economic using a 2.5 MW CHP gas turbine with a LP steam HRSG installed in an industrial plant in a southern Ontario location. The analysis showed that space cooling capacity was more valuable than inlet air cooling for most situations. Descriptions of the thermal cooling cycles that are being investigated by CETC for micro and mini turbines less than 500 kW are discussed.

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1.0 Background

The CANMET Energy Technology Centre (CETC) has been working with small gas turbines, micro turbines since 1998. It was thought that the technology offered a more attractive option than reciprocating engines for small distributed generation projects due to its potential for lower NOx, noise levels and reduced vibration. The first area of work for the program was to assist Canadian companies in the development of heat recovery technologies for these turbines. Two systems were developed, one by the Mariah Energy Corporation of Calgary and the other by Unifin International of London Ontario. Examples of these technologies are shown in Figures 1 and 2. The Mariah system involves the installation of a fin tube heat exchanger that is located on the top of the Capstone turbine. The heat exchanger is used to heat water for DHW or space heating needs. The Unifin concept was a separate heat exchanger that is connected to the microturbine by exhaust ducting. The economics of small distributed generation requires that the exhaust heat be used to provide base load requirements for space heating or cooling needs. In fact the CETC mission statement for its Distributed Generation Program is:

"To assist in the development of a packaged microturbine Combined Heat and Power unit that can be installed by a HVAC contractor with little or no consulting engineering requirements."or

" A boiler or chiller that provides electricity as a by product."

The importance of using the waste heat can be illustrated by the performance of the turbine installation in Figure 2, which is a Cummins/Capstone 60 kW turbine providing power and heat to a large Health Canada laboratory in Toronto. This unit is currently monitored and dispatched remotely using inputs from the Ontario pool price, the gas cost, the electrical demand of the building and the thermal demand of the building. If the building can use the heat, the pool price has to be over a 3.6 c/kWh threshold before the turbine is dispatched; on the other hand if the building does not need heat the pool price has to be over 13.8 c/kWh before the power value overcomes the cost of gas. This value difference in power price indicates the importance of getting value for the waste heat.

Recognizing that the thermal demands in buildings during the summer are low, CETC carried out a comprehensive review of thermodynamic cycles that could use the waste heat from gas turbines to produce chilling. The most common cycle in use today for larger systems is the lithium bromide absorption cycle. What we are trying to determine at CETC is whether there might be other thermal cycles which if re examined with the benefit of modern materials and fabrication technologies might be used. The current problem with thermally activated systems is that they are bulky, heavy and expensive compared with electrical compressor technology. Our objective was to determine if other cycles might be less costly, accepting that a lower efficiency or coefficient of performance might be achieved. In refrigeration the coefficient of performance (COP) is defined as the useful quantity of cooling energy supplied from the system divided by the input energy required to operate the system. In other words would there be a simpler and less expensive system that could be used recognizing that its efficiency might be significantly lower. At this point it should be noted that it is important in the analysis to recognize the importance of parasitic energy consumption in pumps and cooling fans as they can become quite significant in the economic justification for any technical option.

2.0 Examination of the potential for inlet air-cooling using thermal chilling systems.

In addition to the requirement for building cooling using exhaust heat from turbines it was thought that there might be an opportunity to cool inlet air using novel thermally activated systems. A study was commissioned that examined what performance improvement could be achieve for gas turbines using exhaust gas assuming that a chilling system was available with a low COP of 0.3. Two gas turbines were examined using available data. One was a recuperated 400 kW gas turbine and the other was a 4 MW unrecuperated gas turbine. The analysis assumptions are summarized in Table 1. The results are shown in Figures 3 and 4. There is enough energy in the waste gases to flat line the gas turbine performance and the performance of the turbine remains high even in air temperatures up to and exceeding 30°C. Of course this is a theoretical situation since any thermally driven system will need to reject heat and this becomes more difficult and expensive at higher ambient air temperatures. However in situations where there is no other use for the gas turbine waste heat and there are significant hours at high temperature or the plant is at a high elevation this concept might be worthwhile if the chilling system were simple and reliable.

Table 1
Evaluation of performance increase with inlet air-cooling. Assumptions used

Parameter	**Assumption**
Minimum Exhaust Temperature	160°C
Chilling System COP	0.3
Minimum inlet air temperature	4° C
Maximum allowable Relative Humidity	80%
Power Output	At Generator terminals
Inlet air cooling coil Pressure Loss	0.8 " WG
Exhaust Gas Heat recovery Pressure Loss	2.5 " WG

The conclusion can be drawn that the thermal energy in the exhaust of even a recuperated gas turbine can be used to flat line the output of a 400 kW gas turbine using waste heat available in the exhaust even if that cooling system had a low coefficient of performance of 0.3.

Figure 1 Two Mariah 30 kWe/60 kWt. Southern Alberta Institute of Technology, Calgary

Figure 2 Cummins 60 kWe with Unifin heat recovery unit, Health Canada, Toronto

Figure 3
Performance of a 400 kW recuperated gas turbine at various ambient temperatures

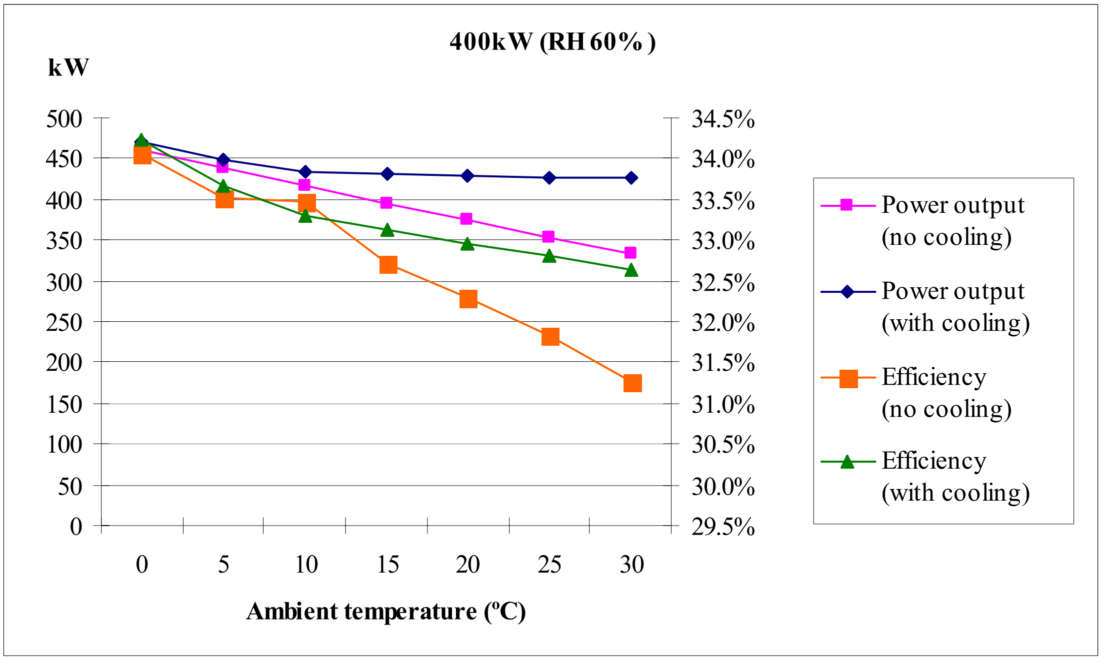
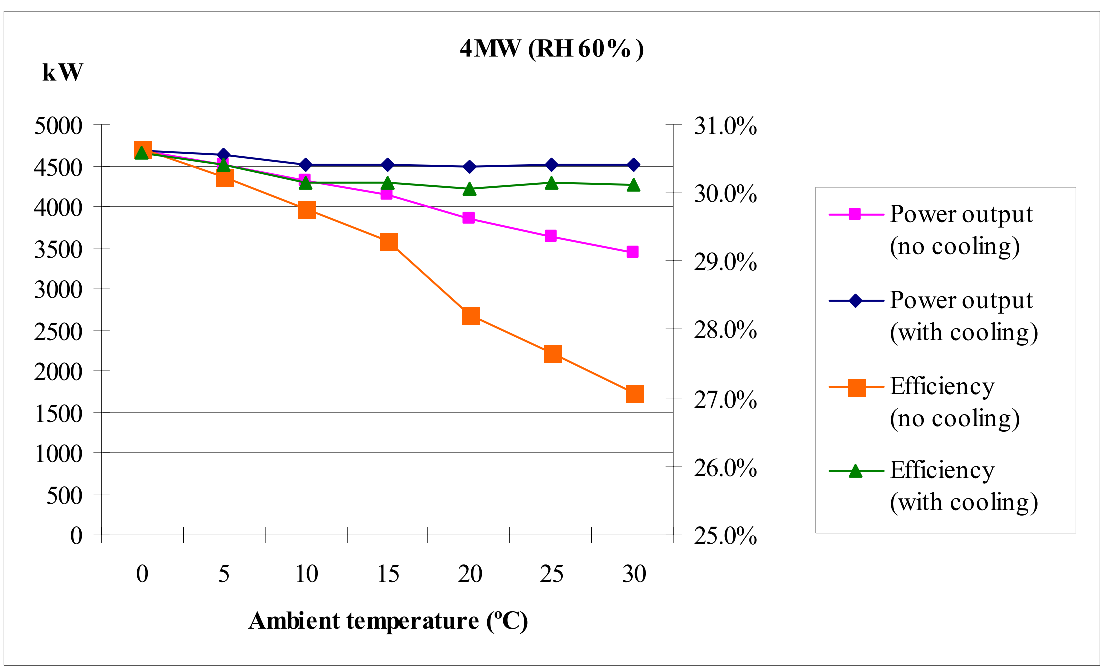


Figure 4
Performance of a 4 MW gas turbine at various ambient temperatures



3.0 Review of Thermally Activated Systems

A good review of these cycles was found in Reference 1 and there were some other cycles that were investigated (2). The results of this survey of thermal chilling cycles are shown in Table 1. Note that those systems that can operate using a low input temperature will also have a lower COP due to the lower Carnot efficiency. The systems taken from Reference 1 are modeled with an ambient temperature of 27.8°C. The COP's will reduce as the heat rejection temperature rises. An arbitrary judgment has been made in the table on system complexity and the quality of thermal input. The temperature level between high and low temperature has been set at 120° C.

Table 2
Coefficient of Performance (COP) of Thermally Driven Cooling Systems

Technology	**COP (Modeled at an ambient T of 82 °F/ 27.8°C)**	**Quality of Thermal Input**	**System Complexity**	**Source**
				Reference 1
Adsorption Zeolite/water	0.4-0.5	Low Temperature	High	Reference 1
	1.2 (with regeneration)	Low temperature	High	Reference 1
Vueilleumier	0.24	High Temperature	High	Reference 1
Ammonia carbon Adsorption	0.7-1.0	Low Temperature	High	Reference 1
Metal Hydride	0.45- 0.7	Low Temperature	High	Reference 1
Ammonia-Organic Compounds	1.0	Low Temperature	Medium (2 beds)	Reference 1
Duplex Stirling	1.41	High temperature	High	Reference 1
Ejector	0.25	Low Temperature	Low	Reference 1
Single Effect Absorption Ammonia	0.58	Low Temperature	Medium	Reference 1
Ammonia Adsorption GAX	0.64	Low Temperature	High	Reference 1
Lithium Bromide Double Effect Absorption	0.94	High Temperature	High	Reference 1
Einstein-Szilard	0.2	Low Temperature	High	Internal CETC

It appeared that there were two systems, other than the single effect absorption chiller, that were worth investigating based on the criteria of low complexity and hence opportunity for cost reduction. These were the ejector concept and the two-bed ammonia adsorption system. These systems also had the ability to producing chilling below freezing while the competing lithium bromide absorption system could not. This feature might be of importance if thermal storage using ice is to be considered.

3.1 Jet Ejector Chiller

Further investigation of the ejector concept has taken place with a co-operative project between CETC and the University of Nottingham, UK under the direction of Dr Ian Eames. Dr's Eames and Petrenko have studied the fluid dynamic issues associated with the ejector refrigeration cycle for several years and have published extensively on this concept. The ejector cycle is shown in Figure 5. The ejector has been in use for many years in industries where waste steam is available as a driving medium to provide low pressure in process systems. The problem has been that steam has a large specific volume at low pressures, which leads to large equipment dimensions.

CETC and the University have identified a recently introduced refrigerant that has good thermodynamic properties as well as being non-flammable, non-toxic and has no ozone depletion potential. In addition CETC and the University of Nottingham believe that with some new consideration of the ejector internal shape profile the COP of the cycle can be raised from 0.25 to perhaps 0.45 or higher. At present the University of Nottingham is building an ejector chiller with a capacity of 2 kW of cooling. CETC is in the process of building a rig with a cooling capacity of 14 kW in its laboratories in Ottawa, which will be used to test the new ejector designs. Early results with a standard ejector design show a COP of around 0.25 in line with expectations. The rig is shown in Figure 6.

Figure 5

Schematic of ejector chilling system for inlet air cooling using a recuperated gas turbine

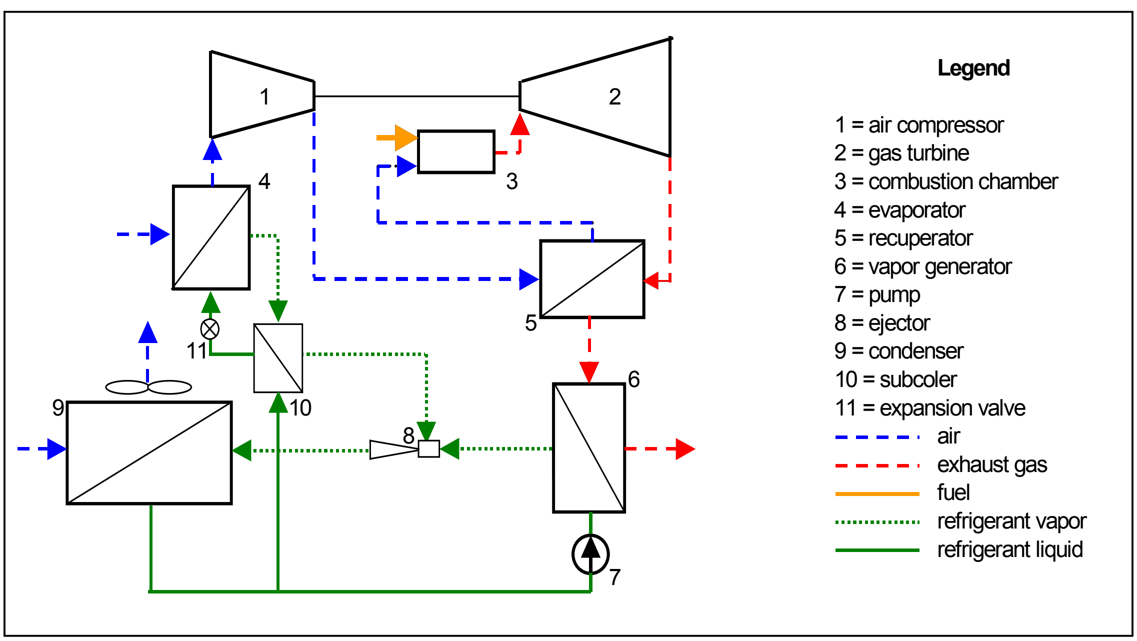
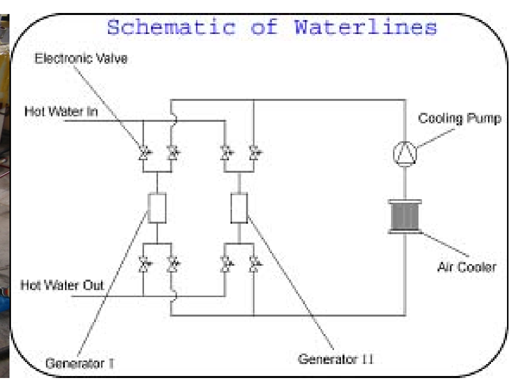


Figure 6
Ejector test rig in the laboratories of CETC, Bells Corners, Ottawa

Figure 7
0.7 kWc DY Chiller Prototype
HW driven system

Figure 8
Water Flow Schematic of DY Unit



3.2 Adsorption Chiller

During the exploratory part of the technology scan contact had been made with the Chinese company DY Refrigeration in Changsha China. Professor Li Dingyu working at the University of Changsha has developed an ammonia salt-based adsorption chilling system. Working with the Japanese company Meidensha a prototype chiller had been built using the exhaust heat of a Capstone microturbine. This prototype is shown in Figure 7 and was based on early DY projects that used the exhaust from fishing boat diesel engines to produce flake ice to keep the fish catch fresh. The Meidensha concept used multiple beds and used the microturbine exhaust heat directly. The problem with this concept was that the direct use of exhaust gas heat made the refrigeration system very large, heavy and costly. A development by DY now enables the heating and cooling of the adsorber beds to be done using water loops. A schematic is shown in Figure 8. The cycle works as follows. Hot water in the range 90 to 120° C is used to heat one of the beds. At the same time the other bed is being cooled using water that is rejecting heat to the ambient using an air cooler. Above 85°C ammonia is desorbed from the bed at a pressure of 2.3 MPa. The ammonia gas is cooled using an air or water loop cooler and then discharged through an expansion valve. The cooled bed has an affinity for the ammonia and this is adsorbed in the second bed. After about 3-4 minutes valving is changed and the cooled bed is heated and the heated bed is cooled. Prototypes have shown the ability to get a COP of 0.3 with evaporator temperatures of –5°C. CETC has a prototype unit of 0.7 kW_c capacity that will be tested over the summer of 2003 and a larger unit of 14 kW_c capacity that will be tested during the spring of 2004.

3.3 Advantages/Disadvantages compared with the Lithium Bromide Absorption Chiller

Table 3 shows some of the advantages and disadvantages of the two systems compared to the standard lithium bromide chiller. This system is available in sizes down to 35 kW_c (10 tons) of cooling capacity from Japanese companies such as Yakazi, and their size and cost effectiveness appear marginal at small sizes. The two alternate systems under consideration appear to offer greater simplicity and the ability to be scaled down in capacity. Whether they will be more cost effective remains to be determined. The benign nature of the ejector refrigerant is seen to be an advantage compared to the ammonia gas used in the DY system. On the other hand the DY concept was the only one found that offered the ability to provide thermal ice storage using inlet water temperatures down to as low as 90°C.

Figure 9
Meidensha 6 bed exhaust gas driven chiller using DY generators, Osaka Japan

Table 3
Relative Advantages of the two novel concepts compared with
a standard lithium bromide chiller

Technology	**Complexity**	**Toxicity**	**Minimum Driving Temperature**	**Comments**
LiBr Chiller	Moderate, can crystallize, operation below atmospheric pressure	Non Toxic	90 ° C	Difficult to make for small systems < 35 kW_c
Ejector	Simpler, Pressure operation	Non Toxic	130° C	Can be scaled down to 1-2 kW_c
Adsorber	Simpler, Pressure operation	Toxic (but no liquids)	90 °C	Can be scaled down to 1-2 kW_c. Pressure vessels scale up has limits

4.0 Economic Analysis

For a thermally driven chiller using the exhaust gas heat as a driver there are two ways by which the chiller can be used to generate revenue to support its capital cost. As shown in Section 2.0 the chilling effect can be used to lower turbine inlet air temperature to the compressor and thus increase the output from the gas turbine. Since this analysis is considering relatively small turbines it is assumed that the gas turbine installation will be associated with a building connected to a distribution utility and therefore onsite power generation will have the ability to save demand charges which in the summer will likely coincide with hot ambient conditions. Note however that in this case the turbine would use additional gas to produce the extra power and this would have to be taken into account.

An alternative would be to use the cooling energy from the thermally activated chiller to reduce the operation of any existing electrical chiller in the associated building. Again electrical savings would be achieved and if the thermal chiller were operating at the time of monthly peak there would be demand savings also.

The approach that was taken to compare the relative values of these two options was to take the opportunity presented to use the first summer of hourly pool power prices in Ontario combined with the availability of hourly temperatures for the Toronto Airport. The turbine was assumed to be a small industrial gas turbine of 2.5 MW_e with an existing HRSG that was used to provide space heating in the winter but had surplus steam during the period from May to the end of September. Using a jet ejector chiller the amount of cooling was determined and the revenue for summer operation calculated for each case. This revenue was then compared with an estimate for the capital cost of such a chilling system.

4.1 Assumptions

An industrial plant is assumed to have installed a 2.5 MW gas turbine with a steam HRSG producing 6,800 kg/hr of steam at 0.8 MPa. Only 50% of the steaming capacity of the HRSG was assumed to be available during May and September but all the steam was assumed to be available during the rest of the summer months. The location of the plant was assumed to be in Toronto and hourly ambient temperature at the Toronto airport was obtained from the Atmospheric Services of Environment Canada for the period 1st May to 30 September 2002. The hourly pool price was also obtained for the same period from the Ontario Independent Market Operator. For ease of analysis the hours were separated into four temperature ranges. The pool power prices for hours having these temperatures were then averaged. The results for this analysis are shown in Table 4.

Table 4
Air Temperatures and average pool power prices May 1st- September 30th 2002

Air Temperature Range (°C)	Number of Hours	Ontario Pool Price 2002 ($/MWh)
>30	197	114.45
25-29	596	79.76
20-24	1059	56
15-19	917	39.78

The chilling system was assumed to be a three ejector jet pump and the condenser was sized for the operation of only two ejectors for temperatures greater than 30°C. This was to avoid the condenser getting very large and expensive for cooling needed for only a short period of time. An evaporative condenser was assumed and the cost of water and associated treatment was taken at 0.8 $/m^3$. The ejector system COP was taken to be 0.45 based on expectations from current research work.

Two operational concepts were investigated. The first used the chiller to cool the inlet air; the second used the chiller to supply space cooling.

4.2 Inlet Air Cooling

It should be noted that the limitation here was that when the air is cooled below the dew point most of the chilling effect is used to condense water rather than further cool the air. Thus inlet air chilling has less impact when the relative humidity is high. As the air temperature falls the chiller evaporator temperature has to fall, reducing COP and capacity. It was found that for inlet air-cooling the lower ambient temperature ranges were impractical due to economic reasons.

Table 5
Inlet Air Cooling Physical Parameters

Air Temperature Range/*Hours* °C / ***Hrs***	**Cooling Power Available** MW_c	**Inlet Air Temperature Reduction** °C	**Increased Gas Turbine Power** MW_e	**Chiller Parasitic Power** MW_e	**Net Extra Power** MW_e	**Thermal Energy Rejected** MW_t
>30 / *197*	0.9	21	0.53	0.13	0.40	3.95
25-29 / *596*	0.5	19	0.47	0.15	0.32	5.15
20-24 / *1059*	0.4	14	0.36	0.14	0.22	5.05
15-19 / *917*	NA					

Table 6
Inlet Air Cooling Economic Parameters

Air Temperature Range/*Hours* °C / ***Hrs***	**Extra Gas Cost** $	**Cooling System Water Cost** $	**Pool Power Price Savings** $	**Transmission & Distribution Savings** $	**Net Savings** $
>30 / *197*	6,398	1,167	9,091	3,158	4,612
25-29 / *596*	17,066	4,604	15,212	4,212	-2,246
20-24 / *1059*	29,954	5,348	13,047	4,240	-18,015

The conclusion from this analysis is that it becomes less profitable to cool the inlet air at cooler temperatures for the following reasons.

- The power price is lower and the spark spread between gas and electricity reduced.
- The COP of the ejector falls with lower ambient air temperature, as the evaporator temperature has to be lowered.

4.3 Replacement of Electric Chiller for Space Cooling

The other use for the thermal cooling is to supply space cooling assumed to be currently provided by an electrical chiller. The amount of cooling supplied by the ejector system is greater in this case because the chiller evaporator temperature can be keep at 8°C rather than the lower temperatures needed for inlet air cooling. The electric space chiller COP including all cooling tower and associated pumping parasitic power, is assumed to be 4.63. It is assumed that the water costs for the electric chiller and the thermal chiller are the same and are therefore not considered.

Table 7
Space Cooling Physical Parameters

Air Temperature Range/*Hours* °C / ***Hrs***	**Cooling Power Available** MW_c	**Saved Power from Electric Chiller** MW_e	**Thermal Chiller Parasitic Power** MW_e	**Net Saved Power** MW_e	**Thermal Energy Rejected** MW_t
>30 / *197*	1.4	0.30	0.13	0.17	5.95
25-29 / *596*	2.1	0.45	0.15	0.32	6.64
20-24 / *1059*	2.1	0.45	0.14	0.22	6.64
15-19 / *917*	1.0	0.23	0.06	0.17	3.32

Table 8
Space Cooling Economic Parameters

Air Temperature Range/ Hours **°C / *Hrs***	**Pool Power Price Savings** **$**	**T&D Savings** **$**	**Net Savings** **$**
>30 / *197*	6,852	1,359	8,211
25-29 / *596*	13,587	3,990	17,577
20-24 / *1059*	17,050	6,143	23,193
15-19 / *917*	6,307	2,960	9,267
Total	43,796	14,452	58,248

In this case the savings from avoided power purchases can be taken as $58,000 per year.

4.3.1 Avoided conventional chiller cost.

At the time of building peak cooling load the ejector chiller can deliver 1.4 MW_c. In more usually understood HVAC terms this is 400 tons of cooling. A cost analysis was done on the installed cost of a standard 600 ton electric chiller. These costs are given in Table 9 and an installed cost of $810/ton could therefore be credited towards the cost of installation, or $324,000 for the ejector chiller's cooling capacity.

Table 9.
Installed cost of 600 ton electrical chiller

Complete 600 ton Chiller and Install (excluding Cooling Tower)	$396,000
Purchase and Install Cooling Tower	$90,000
Total	$486,000
Cost/Ton	$810

Using a three year simple payback the capital cost of the ejector chiller project should not exceed $324,000 + 3x $58,000=$498,000.

A preliminary estimate of the equipment cost of the ejector chiller system was also carried out and is tabulated in Table 10.

Table 10
Cost estimate for Ejector Chiller

Element of Ejector Chiller	
Refrigerant Pumps	$45,000
Refrigerant	$50,000
Refrigerant Vaporizer	$40,000
3 Ejectors	$20,000
Evaporator Subcooler	$40,000
Evaporator Coil	$80,000
Evaporative Condenser	$150,000
Total	**$425,000**

5.0 Conclusions

There is a need to find a cost effective system to provide cooling services using gas turbine exhaust heat. Small gas turbines that are used to provide winter space-heating needs will have surplus waste heat capacity during summer months, which might be used to reduce the break even power price at which the value of power exceeds the cost of gas. The short summer season means that the capital cost of such a chilling system has to be kept low.
Space cooling needs are to be preferred compared to inlet air-cooling since the evaporator temperature can be kept higher hence improving the COP of the thermally activated cooling system. The avoided capacity cost of the alternative electric chiller needed for space cooling is also significant in the economic analysis so the economic case is likely only attractive when new chiller capacity is required or in a new build application. Based on a simple analysis it would appear that a thermally activated chiller would have a three year or less payback using excess steam in the current Ontario power market. It should be noted that the spark spread between gas and electricity also affects the operation of the gas turbine itself and this has not been considered in the analysis.

It is possible that for gas pipeline operations in hot and elevated climates that the ejector chiller concept with a direct exhaust gas evaporator and dry condenser might be found to be cost effective for inlet air cooling. The economics will be dependent on the cost of gas, the value of power capacity and the installed cost of equipment. The ejector concept is very simple with the pump as the only moving part. More work needs to be done however, at smaller sizes to determine the viability of the chilling concepts described. Work at CETC will concentrate on supercharging as a more cost effective method of increasing gas turbine power for Canadian summer conditions.

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