Commercial Building Applications for Fuel Cell Gas Turbine Hybrids:
Analysis of Absorption Cooling Potential Using Exhaust Heat

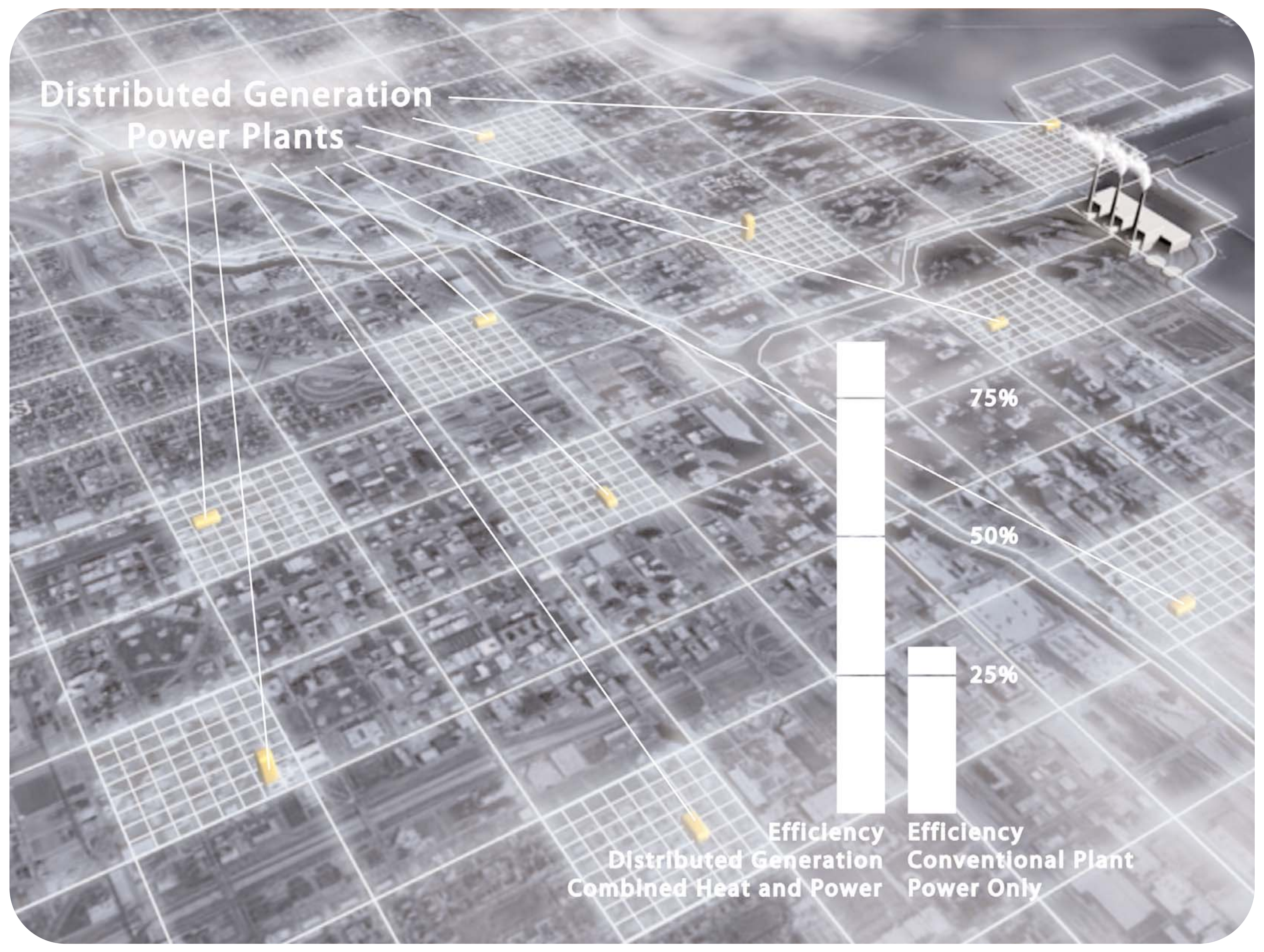
2005 Fuel Cell Seminar

Poster Session 1: Hybrids, MCFC, PAFC, SOFC and Miscellaneous; Poster #1 Presented by Ronald R Engleman Jr, LTI

Overview: The potential of fuel cell gas turbine (FCGT) hybrids to generate clean, efficient, and reliable electricity make them excellent candidates for future distributed generation (DG) installations. DG is an increasingly attractive option for providing electricity, because it improves reliability, reduces transmission losses, lowers costs associated with distribution, and provides the opportunity to capture exhaust heat for cogeneration in combined heat and power (CHP) applications. This analysis considers the potential to recover exhaust heat from a FCGT hybrid to supply thermal energy for commercial building heating, ventilation, and air-conditioning (HVAC). Although the primary role of a FCGT hybrid would be to provide electricity to the local utility grid, capturing and using the exhaust heat could potentially improve the overall efficiency of these systems in future DG installations. Although the FCGT hybrid exhaust gas is no longer of use to the power system, it still contains energy in the form of low-grade heat (LGH). Capturing this LGH and making use of its remaining energy content takes advantage of an otherwise discarded resource. The recovered heat can be used to replace other energy sources such as natural gas or electricity. This reduces the amount of fuel or electricity that must be purchased and used, which lowers operating costs as well as total emissions. Improving the overall environmental performance through integrated cogeneration can further increase potential applications for FCGT hybrids.

Previous on-site fuel cell (FC) installations have shown that finding a beneficial use for FC exhaust heat could greatly boost overall system efficiency. Architects and building engineers who have experience with FC systems in commercial buildings have suggested the use of absorption technology, because absorption chillers can use thermal energy to produce chilled water for cooling. Commercial buildings such as offices generally have large cooling demands even requiring cooling during cooler months, which makes finding a beneficial use for large quantities of waste heat a challenge. Although the absorption refrigeration cycle is generally less efficient than the compressive refrigeration cycle, its potential to use LGH sources can make it economically advantageous. Absorption equipment is also quieter, because it has fewer moving parts; and it does not use chlorofluorocarbons (CFC's) or hexachlorofluorocarbons (HCFC's). Advancements in absorption technology continue to improve efficiency and reduce costs making it an attractive alternative to more conventional cooling systems. A FCGT hybrid, however, is inherently more efficient than the stand alone fuel cell, because the system is optimized to maximize electricity output. This reduces the amount of thermal energy available in the hybrid exhaust heat for cogeneration. Initial calculations determined that FCGT hybrid exhaust may still contain enough thermal energy to provide input to an HVAC system based on absorption chiller technology. The amount of steam that can be generated in a heat recovery boiler from the exhaust of a prototypical FCGT hybrid system is calculated from data provided by the National Energy Technology Center (NETL).

Today: Most electricity comes from central-station power plants that generate power for thousands of residential, commercial, and industrial customers dispersed over a wide area. This power is delivered through a complex network of transmission wires that make up the electricity grid. During the generation of this power a great deal of excess heat is also created. This heat, however, is typically produced too far away from where it can be efficiently used.



Tomorrow: Expanded use of Distributed Generation (DG) power plants generate electricity on site or in a facility located near customers. Since the power is being produced in close proximity to customers, the excess heat created is also available to meet local thermal energy needs. This type of cogeneration or Combined Heat and Power (CHP) application would supply both electricity and thermal energy directly where it is needed.

Opportunities

Challenges

Practical Limitations of Heat Recovery

Heat Recovery Boiler: The proposed heat recovery boiler system would consist of a large duct through which the hybrid exhaust travels over a bank of steam tubes. The overall heat transfer coefficient of this configuration is estimated to be approximately 10 Btu/°F-ft^2-hr (based on gas side heat transfer). This would result in a required heat exchanger surface area of about 6,000 ft^2 to supply 115psig steam to a two-stage absorption chiller. The limiting factor, however, is not necessarily the size and subsequent cost of a heat recovery boiler, but the fact that the hybrid exhaust exits at ambient pressure. This means there is no driving force to overcome the pressure drop through the boiler and provide a high enough velocity to promote good mixing and thermal transfer. The other serious concern would be potential back pressure on the turbine, which would result in decreased hybrid efficiency. One manufacturer of heat exchangers and boilers had an application involving a small boiler used to produce 110psig steam. The temperature of the low-pressure exhaust needed to raise this steam was 1500°F. When the exhaust gas dropped below 1200°F, they found it difficult to provide the required 110psig steam. These temperatures are significantly higher than the 442°F temperature of the hybrid exhaust. Because raising 115psig steam was found to be impractical, the application of a single-stage absorption chiller using 12psig steam was considered as a possible alternative.

Preliminary Heat Recovery Analysis

Single-Stage Absorption Chiller

Provide 40°F to 60°F chilled water for comfort or process cooling applications. Operate using nominal 12psig or 270°F hot water and having a .71 COP.
System Analyzer 5.0 assumes the input for a single-stage absorption chiller to be 18.8 lb_{steam}/ton-hr @ 12psig.

TRANE Horizon Absorption Series, Single-Stage Steam-fired or Hot Water Absorption Water Chillers 500-1350 Tons, ABS-PRC001-EN, TRANE, May 2001.

Two-Stage Absorption Chiller

Provide 40°F to 60°F chilled water for comfort or process cooling applications. Operate using nominal 115psig steam or 370°F hot water and having a 1.21 COP.
System Analyzer 5.0 assumes the input for a two-stage absorption chiller to be 9.83 lb_{steam}/ton-hr @ 115psig.

TRANE Horizon Absorption Series, Two-Stage Steam-Fired or Hot Water Absorption Water Chillers 380-1650 Tons, ABS-PRC004-EN, TRANE, January 2001.

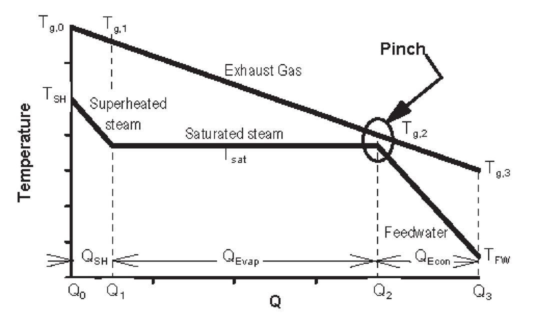
Initial Heat Recovery Estimate: A two-stage steam-fired absorption water chiller for building cooling was first considered. The following section provides a preliminary calculation for how much 115psig (approximately 130psia) saturated steam can be produced from the exhaust gas of a 15.9 MW FCGT hybrid. This provided an estimate of how much of building HVAC energy demand (cooling demand is the limiting factor) could potentially be supplied by the thermal energy in the exhaust gas. The 42 MWe FCGT hybrid was also considered, but the exhaust temperature of 157°F was determined to be too low. Review of the heat recovery boiler calculations suggested that the FCGT hybrid exhaust gas temperature must be at least 378°F in order to produce any useable steam. The 399 kW systems also appeared to have a high enough temperature to generate steam, but the 15.9 MW system offered more available thermal energy due to its larger size and corresponding higher mass flow rate.

Fuel Cell-Gas Turbine Hybrid Exhaust Data

System Size	Pressure	Temperature	Flow Rate
42 MW	1 atm	157°F	370,000 lb/hr
15.9 MW	1 atm	442°F	166,000 lb/hr
4.1 MW	1 atm	335°F	54,000 lb/hr
399 kW	1 atm	444°F	3,890 lb/hr

FCGT Hybrid Exhaust Data, NETL

Heat Recovery Boiler T-Q Diagram



Fuel Cell Handbook, 6th Edition, November 2002, pg. 9-20

Heat Recovery Assumptions (115psig steam):

15.9 MW (hybrid system); $T_{g,1}$ = 442°F; m_{gas} = 166,000 lb/hr; c_p = .25 Btu/lb-°F (hybrid exhaust)
P_{steam} = 130psia + 10psia = 140psia (add 10psia to account for boiler pressure losses)
T_{sat} = 353°F (140psia saturated steam); $T_{g,2}$ = 378°F (includes 25°F pinch temperature for boiler)
h_f = 324.8 Btu/lb (140psia saturated water); h_g = 1193.0 Btu/lb (140psia saturated steam); h_{fg} = 868.2 Btu/lb
$T_{feedwater}$ = 60°F; $h_{feedwater}$ = 28.1 Btu/lb

Discussions with engineers at TRANE regarding absorption chiller technology determined that steam superheat reduces the performance of absorption chillers, so superheat was not included in the preliminary heat recovery analysis.

Heat Transfer + Cooling Potential:

Calculate Heat Transfer and Mass Flow Rate

Q_{gas} (boiler) = $m_{gas}c_p(T_{g,1} - T_{g,2})$

Q_{gas} (boiler) = (166,000 lb/hr)(.25 Btu/lb-°F)(442°F - 378°F)

Q_{gas} (boiler) = 2,656,000 Btu/hr

Q_{steam} (boiler) = $m_{steam}(h_g - h_f) = m_{steam}h_{fg}$

Q_{steam} (boiler) = m_{steam}(868.2 Btu/lb)

Q_{gas} (boiler) = Q_{steam} (boiler)

2,656,000 Btu/hr = m_{steam}(868.2 Btu/lb)

m_{steam} = 3,059.2 lb/hr

Calculate Total Heat Recovery Boiler Duty

Q_{steam} (total) = $m_{steam}(h_g - h_{feedwater})$

Q_{steam} (total) = 3,059.2 lb/hr(1193.0 Btu/lb - 28.1 Btu/lb)

Q_{steam} (total) = 3,563,784.4 Btu/hr

Calculate Final Exhaust Gas Temperature

Q_{gas} (total) = $m_{gas}c_p(T_{g,1} - T_{g,3})$

3,563,784.4 Btu/hr = (166,000 Btu/hr)(.25 Btu/lb-°F)(442°F - $T_{g,3}$)

$T_{g,3}$ = 356°F

Calculate Maximum Potential Cooling (two-stage)

$Cooling_{max}$ = m_{steam}/9.83 lb_{steam}/ton-hr

$Cooling_{max}$ = 3059.2 lb/hr/9.83 lb_{steam}/ton-hr

$Cooling_{max}$ = **311 tons**

Fuel Cell Handbook, NETL, 6th Edition, November 2002, pp. 9-19 - 9-22 for heat recovery calculations.

Heat Recovery Assumptions (12psig steam):

15.9 MW (hybrid system); $T_{g,1}$ = 442°F; m_{gas} = 166,000 lb/hr
c_p = .25 Btu/lb-°F (hybrid exhaust)
P_{steam} = 27psia + 3psia = 30psia (add 3psia to account for boiler pressure losses)
T_{sat} = 250°F (30psia saturated steam)
$T_{g,2}$ = 275°F (includes 25°F pinch temperature for boiler)
h_{fg} = 945.3 Btu/lb

Boiler Specifications:

The vendor proposed a single-pass, firetube boiler (ASME Code Design, Section (IV) to provide the 12psig steam.
Boiler Size: 10 ft diameter pressure vessel 12 ft in length
Thermal Surface Area = 5,200 ft^2
Weight = 66,000 lbs
Boiler Design Pressure = 15psig
Boiler Operating Pressure = 12psig

Heat Transfer + Cooling Potential:

Calculate Heat Transfer and Mass Flow Rate

Q_{gas} (boiler) = $m_{gas}c_p(T_{g,1} - T_{g,2})$

Q_{gas} (boiler) = (166,000 lb/hr)(.25 Btu/lb-°F)(442°F - 275°F)

Q_{gas} (boiler) = 6,930,500 Btu/hr

Q_{steam} (boiler) = $m_{steam}h_{fg}$

Q_{steam} (boiler) = m_{steam}(945.3 Btu/lb)

6,930,500 Btu/hr = m_{steam}(945.3 Btu/lb)

m_{steam} = 7,331.5 lb/hr

Calculate Maximum Potential Cooling (single-stage)

$Cooling_{max}$ = m_{steam}/18.8 lb_{steam}/ton-hr

$Cooling_{max}$ = 7331.5 lb/hr/18.8 lb_{steam}/ton-hr

$Cooling_{max}$ = **390 tons**

Boiler Design + Cooling Potential:

Expected Output of 12psig Steam

The expected steam output from the proposed design is 7,130 lb/hr, which is only slightly less (2.7%) then the 7,331.5 lb/hr calculated using heat and mass balance. The vendor has noted that some draw would be needed to overcome the 1.5" H_2O pressure drop through the boiler.

Calculate Maximum Potential Cooling (single-stage)

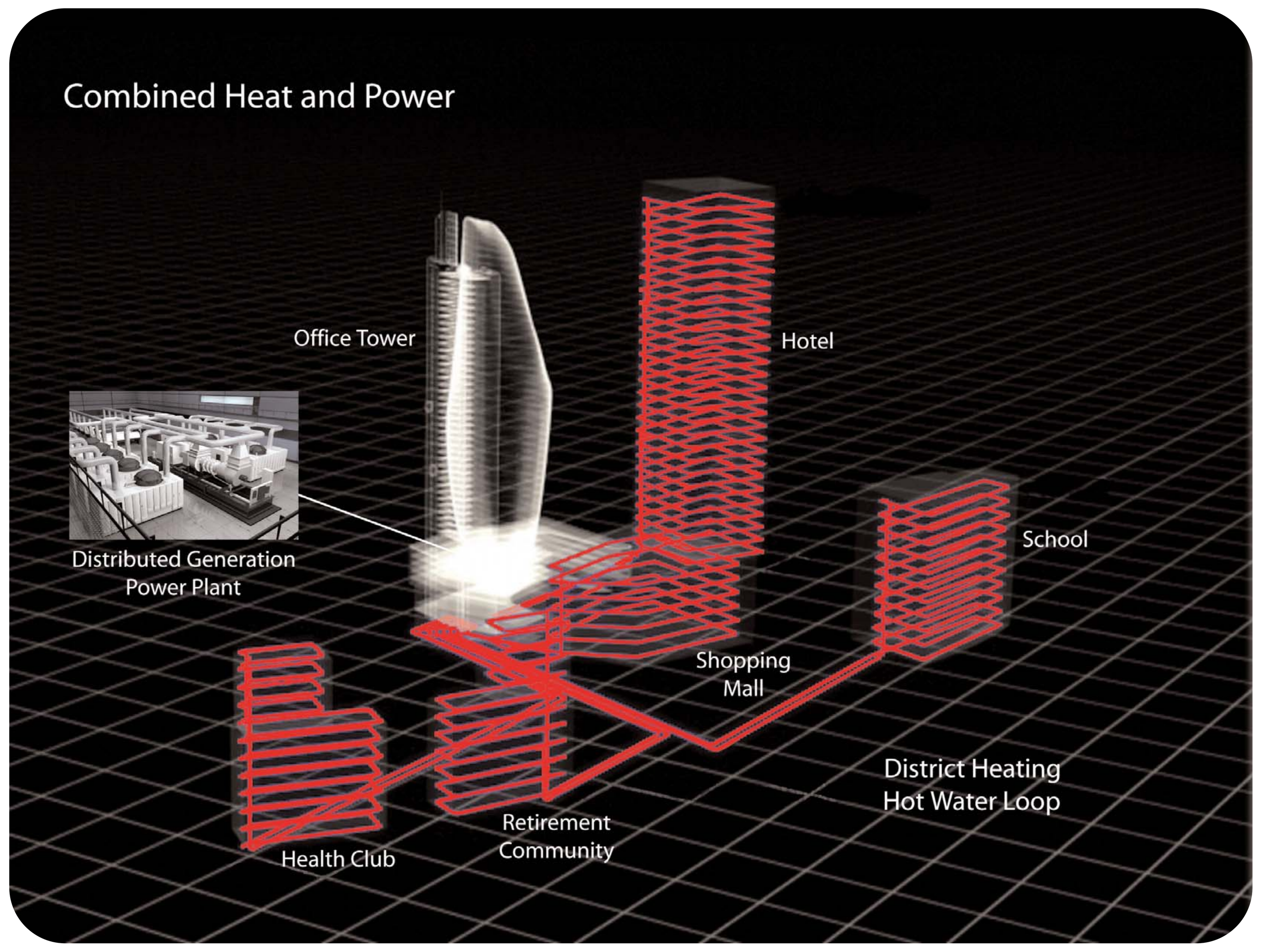
m_{steam} = 7,130 lb/hr

$Cooling_{max}$ = m_{steam}/18.8 lb_{steam}/ton-hr

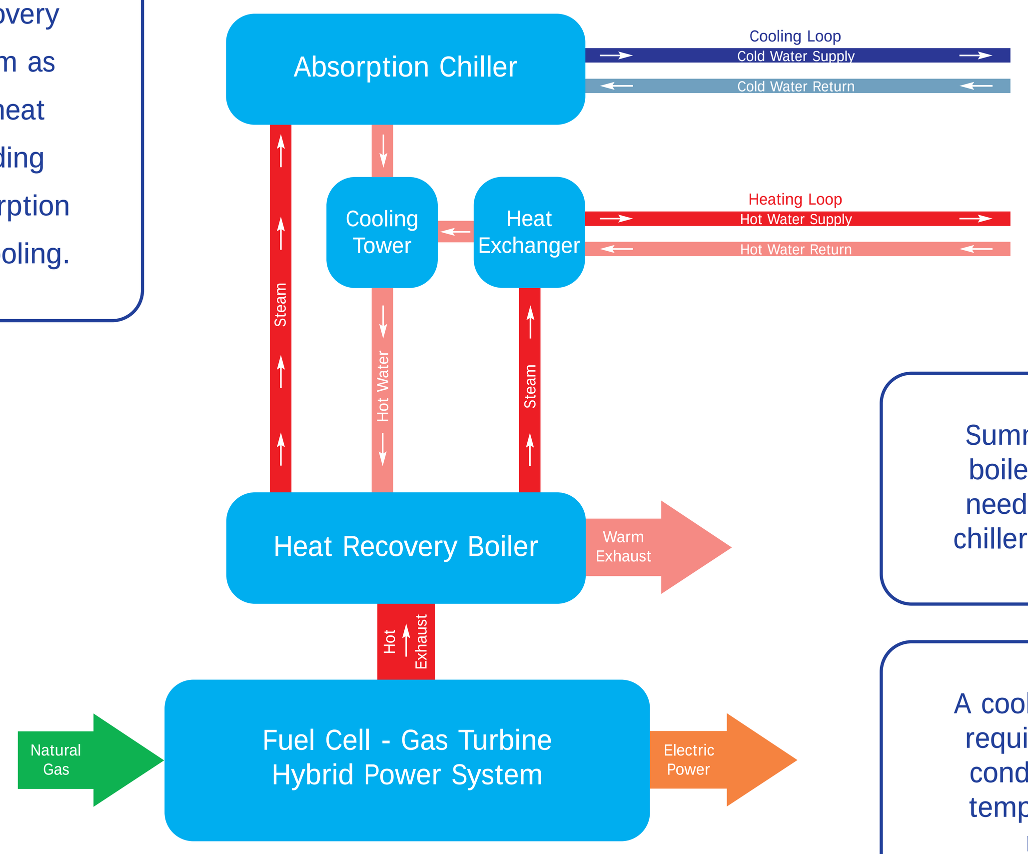
$Cooling_{max}$ = 7,130 lb/hr/18.8 lb_{steam}/ton-hr

$Cooling_{max}$ = **379 tons**

Maximum available cooling from the hybrid exhaust using a single-stage absorption chiller is assumed to be 379 tons.

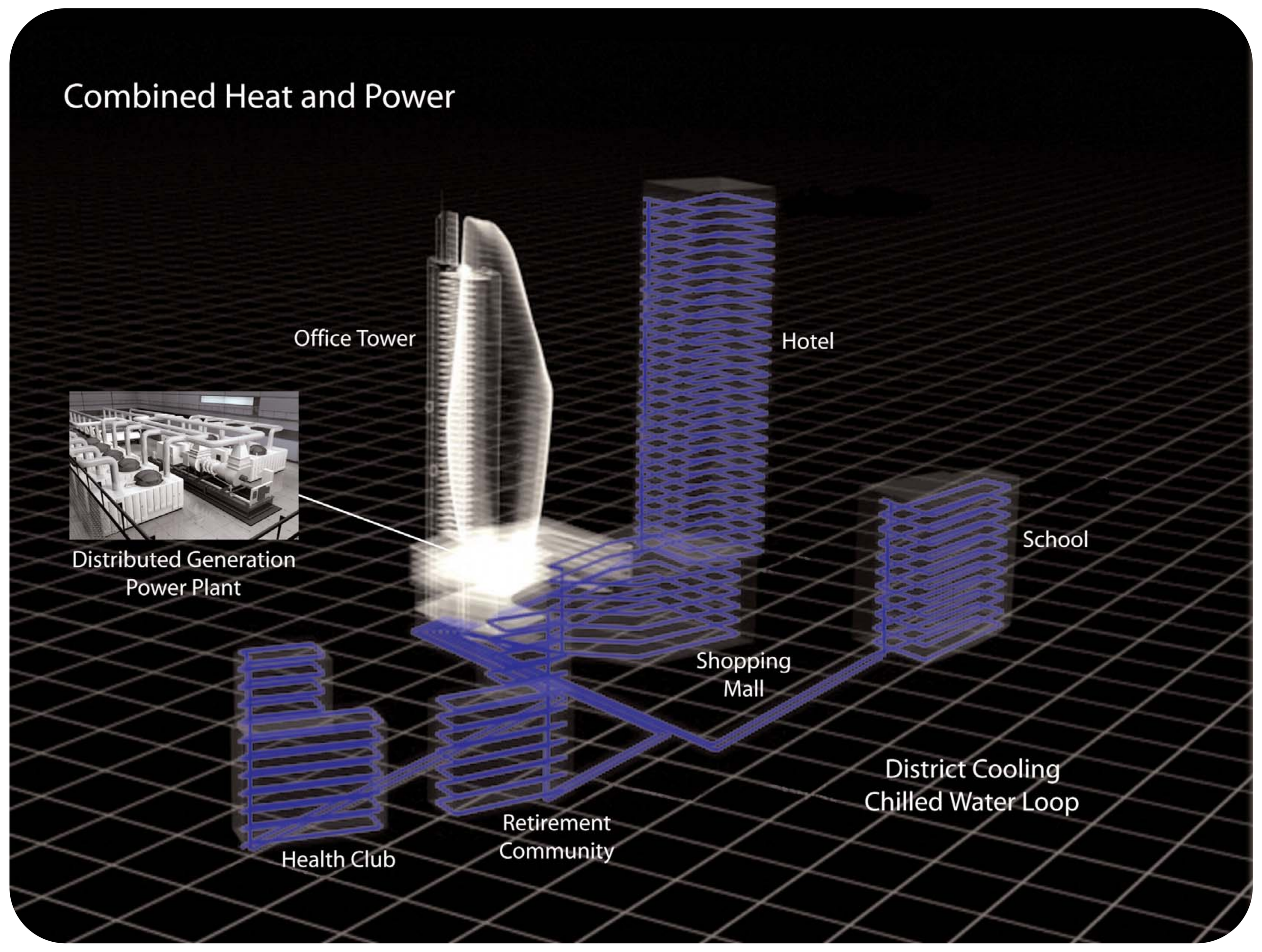


Winter: A heat recovery boiler supplies steam as needed to both a heat exchanger for building heating and an absorption chiller for building cooling.



Summer: Heat recovery boiler supplies steam as needed to an absorption chiller for building cooling.

A cooling tower is used as required to return steam condensate to the input temperature of the heat recovery boiler.



District Heating: The example depicted above involves a DG power plant supplying electricity to local customers as well as heat to a district energy system. DG reduces the efficiency losses associated with transporting electricity over long distances and improves electricity grid reliability by decreasing the load on existing transmission infrastructure. A heat recovery system captures the exhaust heat from power generation to supply such things as building space heating, domestic hot water, dehumidification, and steam production for a variety of industrial processes. In this example thermal energy is delivered to a hot water loop that supplies heat to a series of interconnected buildings. This would be typical in winter or whenever heat is needed throughout the year.

HVAC System Schematic: In the proposed scenario a FCGT hybrid is configured to supply exhaust heat to a central heating and cooling plant for a series of buildings. The central plant would include a heat recovery boiler to generate steam from the FCGT hybrid exhaust. The steam in turn would supply thermal input as needed to an absorption chiller for chilled water distribution and to a steam to hot water heat exchanger for high-temperature water (HTW) distribution. Operating the absorption chiller would primarily take place in summer when cooling demand is high, but cooling is often required throughout the year because of internal thermal loading from people, lighting, and equipment. Hot water and chilled water distribution among a group of buildings is widely used at universities, airports, hospitals, and military bases.

District Cooling: The surplus heat can also be used to operate absorption chillers, which convert thermal energy into chilled water to meet building cooling needs. As an alternative to the electric powered centrifugal chillers typically found in buildings, absorption chiller technology offers an efficient way to make use of excess heat while reducing electricity demand. This example shows a DG power plant sending excess thermal energy to absorption chillers that in turn supply an interconnected chilled water loop. The chilled water loop is parallel to the hot water loop and delivers cooling in summer or whenever additional cooling is required.

Building Energy Use Analysis

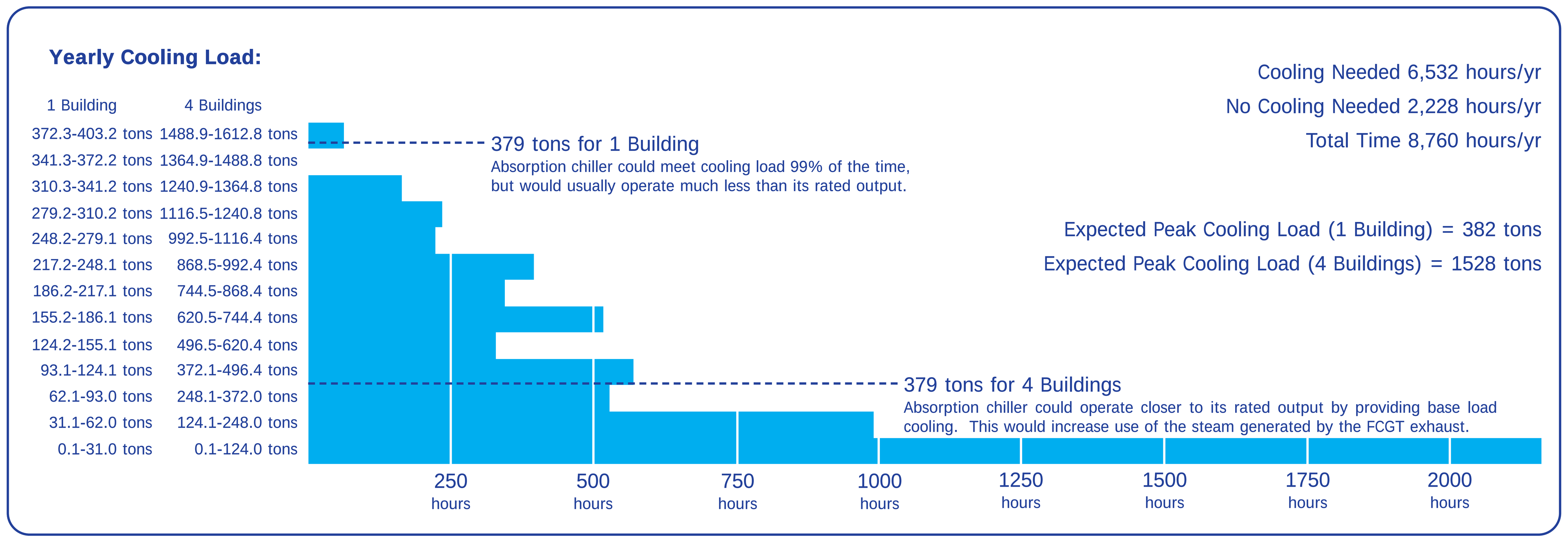
Building Design: System Analyzer 5.0 software developed by TRANE was used to conduct the building HVAC system analysis. A representative urban apartment/office high-rise (25 stories) using standard commercial construction was modeled to determine the heating and cooling load characteristics of a typical building in the Northeast. A building size of 250,000 ft^2 was taken as a standard module. Review of daily energy data from the System Analyzer run shows that the highest space heating demand (2,517,000 Btu/hr) corresponds with the lowest cooling demand (0 tons) in the month of December. The lowest space heating demand (0 Btu/hr) corresponds with the highest cooling demand (382 tons) in the month of August. System Analyzer calculated the total yearly heating load to be 4,748,395 kBtu/yr and the total yearly cooling load to be 677,464 ton-hr/yr. The System Analyzer run did not include hot water needs in this heating demand. DHW requirements were set to be met by supplementary gas heating.

Single Building Data:

Location = Northeast
Building Type = 25 story apartment/office high-rise
Number of Occupants = 1,000
Floor Area = 250,000 ft^2
Wall Area = 150,000 ft^2
Glass Area = 75,000 ft^2
Roof Area = 10,000 ft^2
Weekly Operation = 168 hours
Peak Cooling Load (August) = 382 tons
Maximum Building Cooling Design Load = 620 tons
Peak Heating Load (December) = 2,517,000 Btu/hr
Maximum Building Heating Design Load = 3,685,724 Btu/hr
Yearly Cooling Load = 677,464 ton-hr/yr
Yearly Heating Load = 4,748,395 kBtu/yr

Total Monthly Energy Loads

Month	Heating Loads (kBtu)	Cooling Loads (ton-hr)
Jan	921,681	6,593
Feb	736,401	9,301
Mar	888,896	12,770
Apr	299,554	44,888
May	149,442	72,313
Jun	28,351	110,202
Jul	7,724	129,414
Aug	9,529	131,219
Sep	67,968	91,389
Oct	201,733	40,042
Nov	394,193	23,275
Dec	1,042,924	6,057
Total	4,748,395	677,464



Energy Demand: If all of the steam generated in the heat recovery boiler is directed to the absorption chiller, a maximum total cooling of 379 tons can be produced. This is enough to meet a single building's cooling requirements at least 99% of the time. System Analyzer calculates the maximum cooling design load to be 620 tons, which proposes that the cooling system should be conservatively overdesigned by 62.3% above the peak cooling load of 382 tons. This overdesign is suggested in order to meet extreme temperature days in the summer beyond what is normally experienced. These extreme conditions, however, would only be expected for a few days during a heat wave. A more efficient system would most likely use the absorption chiller up to its rated output to deliver the bulk of the cooling combined with a small centrifugal chiller to deliver peak cooling under extreme demand conditions.

If all of the steam generated in the heat recovery boiler is directed to heating, a maximum of 6,930,500 Btu/hr can be delivered by the steam to a hot water heat exchanger to meet building space heating demand. This would be sufficient to meet the building's predicted winter peak heating requirements of 2,517,000 Btu/hr and is nearly twice the maximum heating design load of 3,685,724 Btu/hr. This does not include DHW energy requirements, which are assumed to be supplied by supplementary gas heating. These DHW thermal needs, which are estimated to average 480,370 Btu/hr and peak at approximately 827,925 Btu/hr, however, could also be provided by the generated steam during the cooler months of the year. Since the highest space heating demand corresponds with the lowest cooling demand, there is enough steam to always meet building space heating demand throughout the year. DHW needs could be met most of the year in addition to the space heating requirements, but not during peak summer cooling demand when all of the steam would be needed to supply the absorption chiller. Cooling demand is the limiting factor of how much of the HVAC energy load can be furnished by the steam generated from the FCGT exhaust.

An improved configuration would use the 379 tons of cooling to provide base load cooling for a group of buildings integrated into a district cooling loop. The 379 tons is an amount great enough to meet the cooling needs of up to four 250,000 ft^2 buildings (a total of 1,000,000 ft^2 of space) at least 56% of the time. A centrifugal chiller could then be used to deliver peak cooling needs during the summer months. Use of a combined absorption and centrifugal chiller connected in series allows more flexibility to the overall system. Under the series chiller design, the chilled water path goes first through the absorption chiller and then through the centrifugal chiller. The decision on which chiller provides base load is usually determined by prevailing energy prices. In this case, since the exhaust heat is assumed to have a zero incremental cost, the absorption chiller would provide the base load. As load on the absorption chiller approaches 90%, the centrifugal chiller would be brought on line and the loading on both chillers is optimized for total system efficiency. The efficiency of a chiller is highest when operated near its rated capacity, so an optimized system would most likely use two or more smaller absorption chillers connected in parallel for base cooling load with an additional centrifugal chiller connected in series for peak cooling.

If district heating and cooling were used to supply hot and chilled water to a group of buildings, these buildings could include other building types such as schools, offices, shops, recreational facilities, and hotels. Mixing building types helps to balance the composite energy use and provide a more stable heating and cooling load throughout the year, because different building categories have unique energy demand profiles. Offices require cooling even during cooler months, but relatively low heating. Recreational facilities and hotels typically have larger heating demands due to laundry facilities, pool heating, large support spaces, and other services. Integrating a variety of building types also helps to equalize the energy loads throughout the day. Schools and offices would have high energy demands during the day, but little demand during the evening. Residential buildings would tend to have peak energy demand in the early morning and during the evening. System Analyzer is not capable of modeling a group of different buildings tied together into a district system, but interconnected buildings can be modeled using TRACE 700 from TRANE or EnergyPlus software available through DOE.

Life Cycle Analysis

Savings Estimate: If the FCGT exhaust is used to operate an absorption chiller for a single 250,000ft^2 building, the cost savings is approximately 47,074 USD/yr with a projected payback period of approximately of 3.1 years. If all the available FCGT exhaust is used to operate an absorption chiller to its maximum potential, the cost savings could be as much as 230,823 USD/yr resulting in a payback period of only 0.6 years.

Economic Assumptions:
Capital Cost (Single-Stage Absorption Chiller 300-400 tons) = 350 USD/ton
Capital Cost (Centrifugal Chiller 300-400 tons) = 250 USD/ton
Average Energy Use (Centrifugal) = .52 kW/ton
Yearly Cooling Load (Single Building) = 677,464 ton-hr/yr
Yearly Cooling (Maximum Potential) = 379 tons x 8760 hours/yr = 3,320,040 ton-hr/yr
Cost of Electricity = .12 USD/kW-hr
Monthly Demand Charges = 10 USD/kW-month
Capital Cost (Heat Recovery Boiler) = 106,000 USD
Discussions with engineers at TRANE regarding costs suggested that improvements in absorption technology has reduced maintenance costs to only slightly higher than centrifugal chillers, so a maintenance cost differential was not included in the cost estimate.

Capital Investment:
Total Capital Cost Difference (Absorption) = Size x Capital Cost Difference (Absorption)
Capital Cost Difference (Absorption) = 350 USD/ton - 250 USD/ton = 100 USD/ton
Total Capital Cost Difference (Absorption) = 379 tons x 100 USD/ton
Total Capital Cost Difference (Absorption) = 37,900 USD
Total Investment Cost = Total Capital Cost Difference (Absorption) + Capital Cost (Boiler)
Total Investment Cost = 37,900 USD + 106,000 USD = 143,900 USD

Single Building Savings (Low Estimate):
Electricity Consumed = .52 kW/ton x 677,464 ton-hr/yr = 352,281 kW-hr/yr
Avoided Electricity Cost = Electricity Consumed x Cost of Electricity
Avoided Electricity Cost = 352,281 kW-hr/yr x .12 USD/kW-hr = 42,274 USD/yr
Average Electricity Demand = .52 kW/ton x (677,464 ton-hr/yr / 8760 hours/yr) = 40 kW
Avoided Demand Cost = 40 kW x 10 USD/kW-month x 12 month/yr = 4,800 USD/yr
Total Yearly Avoided Cost = 42,274 USD/yr + 4,800 USD/yr = 47,074 USD/yr
Payback Period = 143,900 USD / 47,074 USD/yr = 3.1 yr

Maximum Potential Savings (High Estimate):
Electricity Consumed (Maximum) = .52 kW/ton x 3,320,040 ton-hr/yr = 1,726,421 kW-hr/yr
Avoided Electricity Cost = 1,726,421 kW-hr/yr x .12 USD/kW-hr = 207,171 USD/yr
Peak Electricity Demand = .52 kW/ton x 379 tons = 197.1 kW
Avoided Demand Cost = 197.1 kW x 10 USD/kW-month x 12 month/yr = 23,652 USD/yr
Total Yearly Avoided Cost = 207,171 USD/yr + 23,652 USD/yr = 230,823 USD/yr
Payback Period = 143,900 USD / 230,823 USD/yr = 0.6 yr

Centrifugal Water Chillers, 400-1000 Tons, CTV-PRC001-E4, March 2003.
Earthwise CenTraVac Water-Cooled Liquid Chillers, 170-3950 Tons, CTV-PRC007, TRANE, June 2005.

Thanks to the United States Department of Energy for their support and assistance in the completion of this study. Special thanks also to Hurst Boiler & Welding Company, Inc. for their input on boiler design and manufacturing and to TRANE for their guidance on the use of System Analyzer software and the technical information they provided regarding absorption chiller technology.