Bulk Energy Storage using a Supercritical CO_2 Waste Heat Recovery Power Plant

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ABSTRACT AND INTRODUCTION

This report describes a bulk energy storage and power peaking concept that is coupled to a Supercritical CO_2 (SCO_2) Waste Heat Recovery (WHR) power plant. The waste heat source could be the exhaust from a 25 MWe class gas turbine or hot gases from manufacturing process such as a metal smelter. The SCO_2 power system is configured either as a supercritical Brayton cycle (off-peak power production) or as a transcritical Rankine cycle (on-peak). Energy is stored as ice in the "charging cycle" and then discharged by melting the ice during the on-peak demand time period. The ice is produced in a transcritical CO_2 refrigeration cycle (heat pump) from CO_2 at –5 C. The power to run the CO_2 refrigeration cycle can come from either the grid or directly from the SCO_2 Brayton cycle. The charging cycle operates for approximately 8 hours during the night and early morning when the demand and the price of electricity are low. The stored thermal energy is recovered, or "discharged", over a period of about 4 hours during the early evening when peak power demands and the price of electricity are high.

During the discharge process the stored energy plus additional energy from the waste heat stream is recovered because the flow in the WHR plant is directed, through valves, to use the stored ice as the heat sink during the discharge cycle. During the discharge process an additional turbine stage and compressor stage are added to the turbo machinery to accommodate the required larger pressure ratio. In the discharge cycle the CO_2 is cooled to +5 C by melting the ice. This results in a lowering of the CO_2 cold side pressure thereby condensing the CO_2. For this reason, the CO_2 discharge cycle operates as a transcritical Rankine cycle. Because of the corresponding lower turbine discharge pressure and the lower heat rejection temperature the power is increased by up to 66% when compared to the off-peak Brayton power product process. In addition, the cycle net efficiency increases (34.5% versus 31.7%). Plus the plant makes more effective use of the waste heat (68.0% versus 44.7%).

The dispatchable round trip efficiency is the ratio of the energy produced in the discharge cycle to the energy purchased to make the ice (bulk energy storage). Early calculations indicate that for a waste heat flow stream, representative of a gas turbine (538 C), the dispatchable round trip efficiency is in the range from 148% - 183% depending on whether an expansion valve is used or if energy is recovered in a turbo-expander within the refrigeration cycle. The efficiency exceeds 100 percent because electricity is recovered from both the waste heat flow stream and from the stored energy. However, the excess dispatchable round trip efficiency is less than one. The excess efficiency is defined as the incremental electricity generated above that of the WHR plant without using stored energy. The excess dispatchable efficiency varies from 59% - 73% depending on how the SCO2 refrigeration plant is configured.

SCO2 POWER PRODUCTION AND REFRIGERATION CYCLES

POWER PRODUCTION CYCLE DESCRIPTION

The process flow diagram for the power production cycles is illustrated in Figure 1. This cycle has three unique attributes to it that make it well suited for this waste heat recovery process. The three attributes are split-flow with preheating, compressor inter-recuperation, and switching from a non-condensing Brayton cycle to a condensing Rankine cycle. Each is described briefly below.

1. <u>Split flow with preheating</u>: The cycle splits a fraction of the flow from the compressor and sends the high pressure CO_2 to a preheater. The split flow with preheating improves the performance for waste heat recovery and combustion based heat sources because it increases the utilization of waste heat or combustion heat by transferring more thermal power into the CO_2 thereby using more of the available heat. This is a process that was originally developed for Organic Rankine Cycles [1,2,3]. It also makes it easier to avoid pinching within the recuperator because of the reduced flow in the high pressure (high heat capacity) leg of the recuperator. (Note, for the model presented here the exit of the preheater is arranged to have the same temperature and pressure as the high pressure leg exit from the recuperator. However, this is not a requirement for all applications.)
2. <u>Compressor inter-recuperation:</u> The cycle uses a patented compressor inter-recuperation technique[4] to increase the cycle efficiency by recuperating the remaining thermal energy from the low pressure leg of the recuperator. To achieve an efficiency benefit, it is necessary to

perform the compression in two stages thereby allowing the recuperated energy to be added between the two stages. This feature looks similar to compressor inter-cooling; however, recuperated heat is added to the intermediate pressure flow stream rather than being removed. For smaller SCO_2 power cycles this has the added effect of increasing the volumetric flow rate through the compressors which lowers the compressor speed, allows for a larger compressor, and thus increases the compressor efficiency. The last major benefit is that it lowers the amount of heat that needs to be rejected to the environment because a large fraction is recuperated back into the loop. It therefore lowers the cost of the heat rejection hardware.

3. Cycle Switching from Non-Condensing Brayton to an ICE-condensing Rankine: The cycle uses valves to select the type of heat rejection (condensing/non-condensing). During off-peak demand times, the power production process uses a heat sink that is non-condensing with respect to the CO_2. For this power cycle the heat can be rejected to water via an evaporative cooler or it can be rejected to air. This results in the compressor inlet being above the critical temperature which makes the power system an SCO2 Brayton cycle. However, valves can be used to switch the CO_2 flow to an "Ice-on-Coils" bulk energy storage tank to melt the ice, and thus condense the CO_2. Because the CO_2 is cooled to +5C which is below the critical temperature (31 C), it therefore has a vapor pressure that is below the critical pressure resulting in CO_2 condensation. This Ice-melting heat rejection process is part of the "discharge" power cycle. It melts the ice that was previously generated in a "charging" process during low demand (late evening and early morning) when excess renewable wind energy is often available. Also note that to accommodate the larger pressure ratio in the ICE-Rankine discharge cycle, both the turbine and the compressor will require an additional stage that must be valved in or out when switching between the Brayton and the Ice Rankine cycle (not shown in the figure).

The thermodynamic cycle analysis presented in the process flow analysis was determined by using Microsoft Excel 2010 with the spreadsheet solver, and coupled to the NIST Refprop subroutines for the CO_2 equation of state. The waste heat source was selected to represent the combustion gas flow (~69.8 kg/s at 538 C) from a 25 MWe class gas turbine or flue gas from manufacturing process with greater than 40 MW_{th} of waste heat at a similar temperature. The process flow diagram is presented in Figure 1, while the results of the temperature and pressure state point analysis, plus the fluid flow rate and fluid type are provided in Table 1. A summary of the thermal or mechanical power in the heat exchanger and turbomachinery components is presented in

Table 2. Likewise the T-S curves for both the Ice-Rankine and the Brayton cycles processes are illustrated in Figure 2(a & b).

A summary of the expected performance for the off-peak Brayton cycle and the on-peak Ice-Rankine cycle is provided in

Table 2. The CO_2 mass flow rate through the discharging Ice-Rankine cycle was selected to reject all of the CO_2 waste heat by melting the ice over a 4 hour period. The refrigeration system size was selected to provide sufficient ice for these four hours of heat rejection (ice-melting), but was generated during an 8 hour period during the late evening and early morning hours. Two types of refrigeration systems were assumed for the analysis. One uses an expansion valve. The other uses a turbo-expander to recover some of the expansion energy. These cycles are described in the next section.

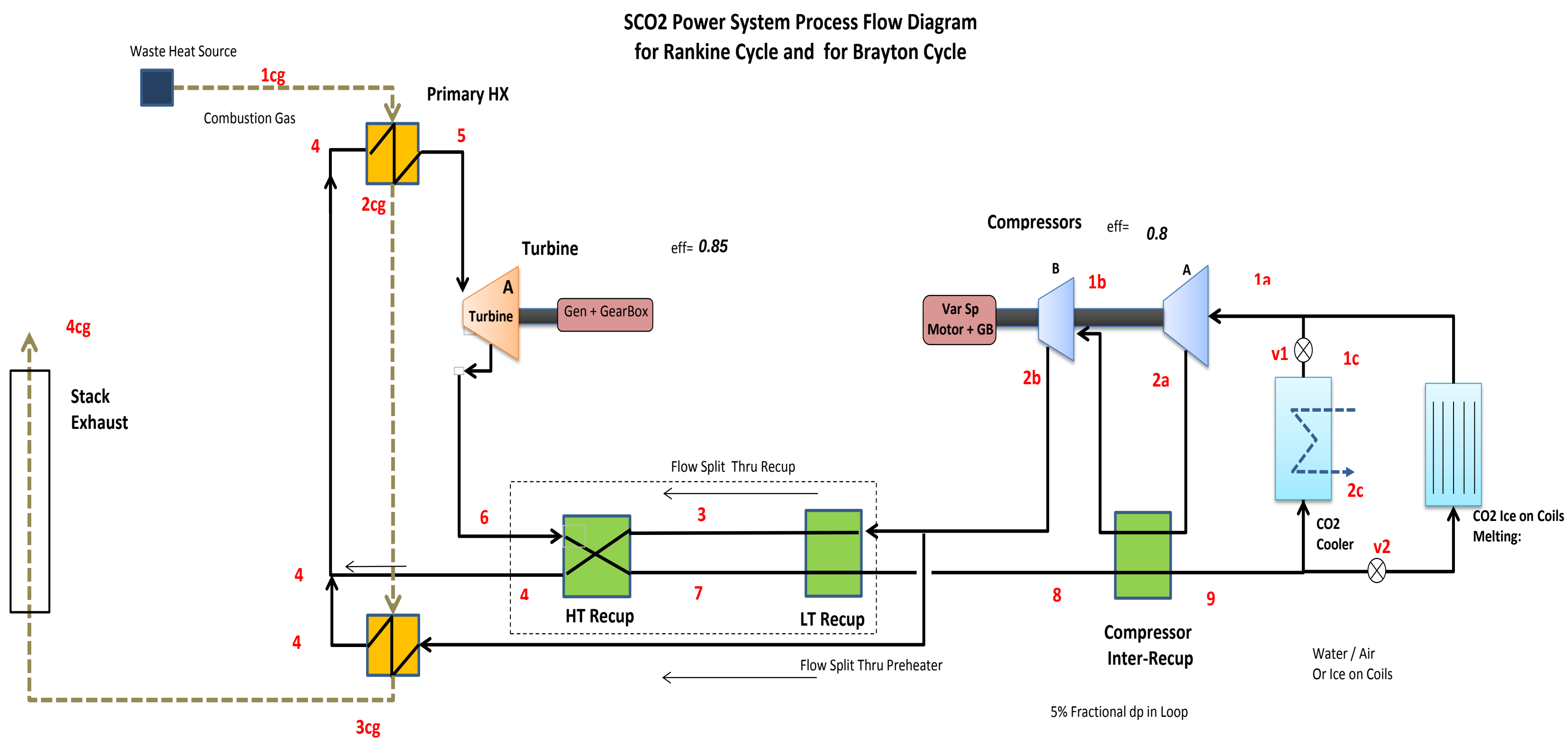

Figure 1: S CO_2 WHR process flow diagram using bulk ice-energy storage. This power production cycle is used for both off-peak and on-peak power production. The cycle splits a fraction of the flow from the compressor and feeds it directly to a preheater, it also uses a compressor inter-recuperation technique, and it can use either a non-condensing or a condensing heat sink. The condensing heat sink cools the CO_2 by melt ice. The heat sink is selectable via valves.

Table 1: Temperature and pressure state points for the S CO_2 Brayton cycle and the ICE-Rankine cycle. The fluid type and mass flow rate are also provided.

SCO2 POWER CYCLES	1a Comp A inlet	2a Comp A Out	1b Comb B Inlet	2b Comp B Outlet	2c Preheat Hx In	3 Integ. Recup HP Internal Loc	4 Primary HX Inlet	5 Prim. HX Outlet	6 Turbine Out	7 Integ. Recup LP Internal Loc	8 Inter-Recup Inlet	9 Heat Rejection Water/Air or Ice	cg1 Waste Heat Source	cg2 Prim HX Exit	cg3 Preheat HX Exit	cg4 Stack Exhaust	c1 Cooling Water Inlet	c2 Cooling Water Exit
BRAYTON																		
Temperature (K)	304.5	328.4	350.1	374.9	374.9	445.6	614.3	761.2	644.3	482.0	389.4	338.4	811.15	644.31	581.7	581.7	292.15	308.38
Pressure (kPa)	7500	15927	15795	23400	23400	23205	23012	22820	7691	7691	7627	7563	101.00	101.00	101.00	101.00	420.00	320.00
Fluid Type	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CG	CG	CG	CG	H2O	H2O
Mass Flow Rate (kg/s)	70.2	70.2	70.2	70.2	13.36	56.88	70.2	70.2	70.2	70.2	70.2	70.2	69.8	69.8	69.8	69.8	169.66	169.66
ICE-RANKINE																		
Temperautre (K)	278.0	289.6	350.1	374.9	374.9	467.1	554.9	761.2	584.9	529.2	450.3	299.6	811.2	584.9	462.2	462.2	273.15	273.15
Pressure (kPa)	3954	15927	15795	23400	23205	23205	23012	22820	4055	4055	4021	3987	101.0	101.0	101.0	101.0	101.0	101.0
Fluid Type	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CO2	CG	CG	CG	CG	ICE	ICE
Mass Flow Rate (kg/s)	67.5	67.5	67.5	67.5	33.12	34.34	67.5	67.5	67.5	67.5	67.5	67.5	69.8	69.8	69.8	69.8	-	-

Table 2: Table of power and heat transfer values for the ICE-Rankine Power system and the SCO2 Brayton power system. Both power system use a recuperated power cycle with compressor inter-recuperation (CIR). Also, both power cycles split some of the compressor exit high pressure flow through a CO2 preheater without recuperation.

Description	Name	Unit		ICE Rankine	Brayton
Preheater HX Therm Pwr	Q.PreHX	kW		9418.6	4804.6
Primary HX Therm Pwr	Q.PrimHX	kW		17372.5	12810.1
Total CO2 Hx Therm Pwr	**Q . Prim + Pre Hx**	**kW**		**26791.1**	**17614.8**
LT Recup Therm Pwr	Q LT Recup	kW		5670.9	7542.0
HT Recup	Q.HT.Recup	kW		4096.7	12909.5
Integ Recup	**Q.Integ Recup**	**kW**		**9767.6**	**20451.5**
Comp Inter-Recuperation	**Q.CIR**	**kW**		**11295.8**	**4931.7**
Total Recup	*Q.recup.total*	*kW*		*21063.43*	*25383.18*
Main Comp Pwr	P.Comp	kW		2329.1	2430.9
Comp Pwr A	P.Comp. A	kW		1100.1	1151.3
Comp Pwr B	P.Comp.B	kW		1228.9	1279.6
ReComp Pwr	P.ReComp	kW		0.0	0.0
Power Ratio.cycle				*162.2%*	
Power Losses					
Aux Power (water pump, purge gas, fans, elec		kW		**68.41**	**88.28**
CompShaft-Power Losses (Driven Wheel)		kW		**92.23**	**96.26**
Turbine Train Power Losses (Driving Wheel)	Sync Gen	kW		**484.28**	**337.90**
Total Power Train Losses		kW		**576.51**	**434.16**
Total Power Train Loss Fraction				**5.82%**	**7.12%**
Net Elect Power		kW		**9255.27**	**5579.40**
Power Cycle Ratio Net (Rankine/Brayton)				**166%**	
Eff Cycle				**0.370**	**0.346**
NET Elect Cycle Efficiency			**0.366**	**0.345**	**0.317**

The key operating parameters are that the net power produced by the Ice-Rankine cycle is 9255 kWe compared to 5579 kWe in the off-peak Brayton cycle power generation process. The remarkable observation is that the Ice-Rankine cycle (discharge process) produces 66% more power than the standard Brayton cycle for the same waste heat source, for the same waste heat mass flow rate, and for a similar CO_2 mass flow rates (67.5 kg/s in the Ice-Rankine cycle process versus 70.2 kg/s in the Brayton cycle). This increase in power occurs primarily because of the

1. decreased back pressure on the turbine due to the condensing the CO_2 at + 5 C within the ice-melting heat sink,
2. the better cycle efficiency due to the lower heat sink temperature 34.5% versus 31.7%, and
3. higher waste heat utilization fraction; defined as the fraction of waste heat that gets into the CO_2 . For the Ice-Rankine cycle it is 68% while it is 44.7% for the Brayton cycle. *(Note, the Excel solver was used to maximize the amount of electrical power produced for the proposed system configuration. Also, the performance for other system configurations was not fully examined, thus there may be other power cycle configurations that offer some marginal benefit.)*

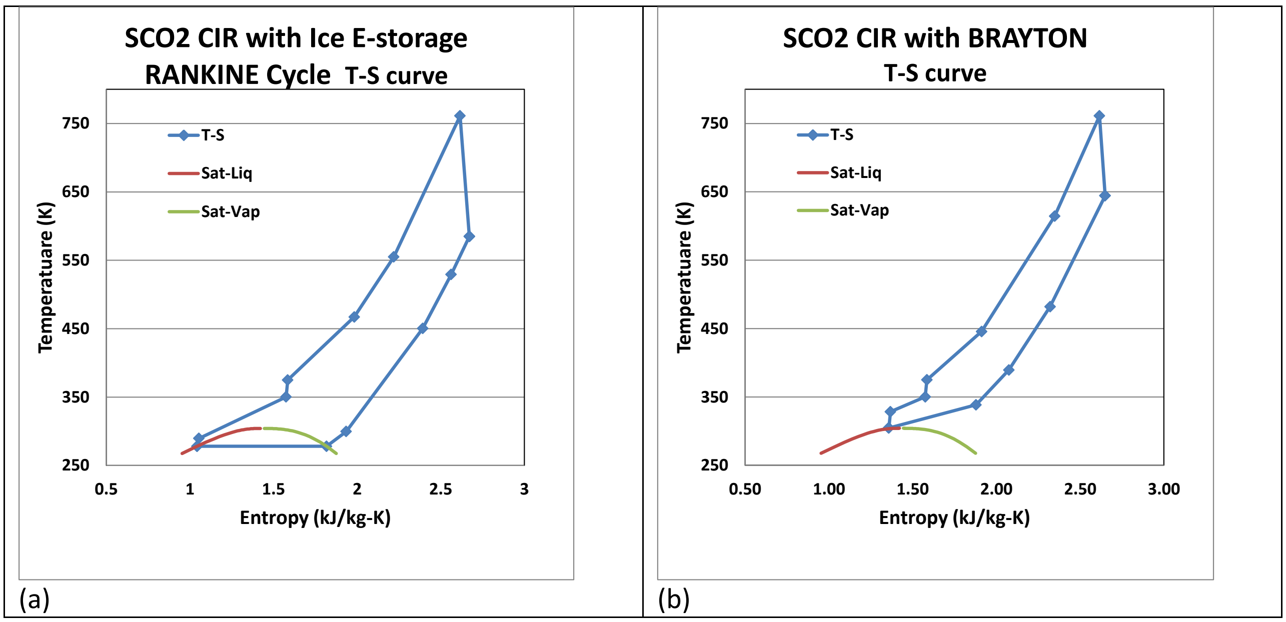

Figure 2: Temperature-Entropy diagrams for the Ice-Rankine cycle (a) and for the non-condensing Brayton S CO_2 power cycle (b).

SCO_2 REFRIGERATION CYCLE DESCRIPTION

As described in the preceding section, the power peaking plant uses the melting of bulk quantities of stored-ice to lower the heat rejection temperature and pressure which greatly increases the power generation capability of the ice-Rankine cycle compared to the Brayton cycle. The method of generating this ice is described in these paragraphs. It consists of an SCO_2 transcritical refrigeration heat pump plant that takes heat from water to form ice, and rejects high temperature waste heat to the environment. The process flow diagram for the ice-making SCO_2 refrigeration system is illustrated in Figure 3. The temperature entropy diagram for this cycle is shown in Figure 4. A summary of the important operating parameters are also provided in Table 3a and 3b.

The refrigeration system uses CO_2 at -5 C to remove heat from water to form ice. The ice is formed over coils of CO_2 flowing within a large ice-on-coils tank (~ 5m height x 20 m OD for 50 volume percent ice). Note that in the proposed peaking plant this ice is generated over an 8 hour time period, while the discharge process (ice-melting) takes place over 4 hours. Electricity is either purchased from the grid or from the SCO_2 Brayton cycle that continues to run during this time period. The same ice-on-coils tank is used in the charging cycle as is used in the discharging cycle (ice-melting). The CO_2pressure is ~30 bar within the coils of the storage tank. Ice-on-coils energy storage tanks are commercial components [(5)], but development is still required to use CO_2 to make ice, primarily because of the larger pressure of CO_2 in the coils. Most commercial ice-on-coil or tube-ice makers use either a refrigerant or a water-ethylene-propylene-glycol mixture at approximately -5C make the ice.

The performance of the ice-generating refrigeration loop can be improved by using a turbo-expander to lower the pressure and temperature of the CO_2 rather than an expansion valve. This type of turbo-expander is often used in Air Separation Units to recover some of the energy that is available during the expansion process. However, some amount of development and testing to verify the ability of the turbine to expand into the liquid side of the two phase region and to characterize the efficiency of this turbine. Because there is only a factor of about 10 between the liquid density and the vapor density, erosion in the turbine is expected to be greatly reduced compared to steam systems where this is problematic but can be accepted over a range of liquid fractions.

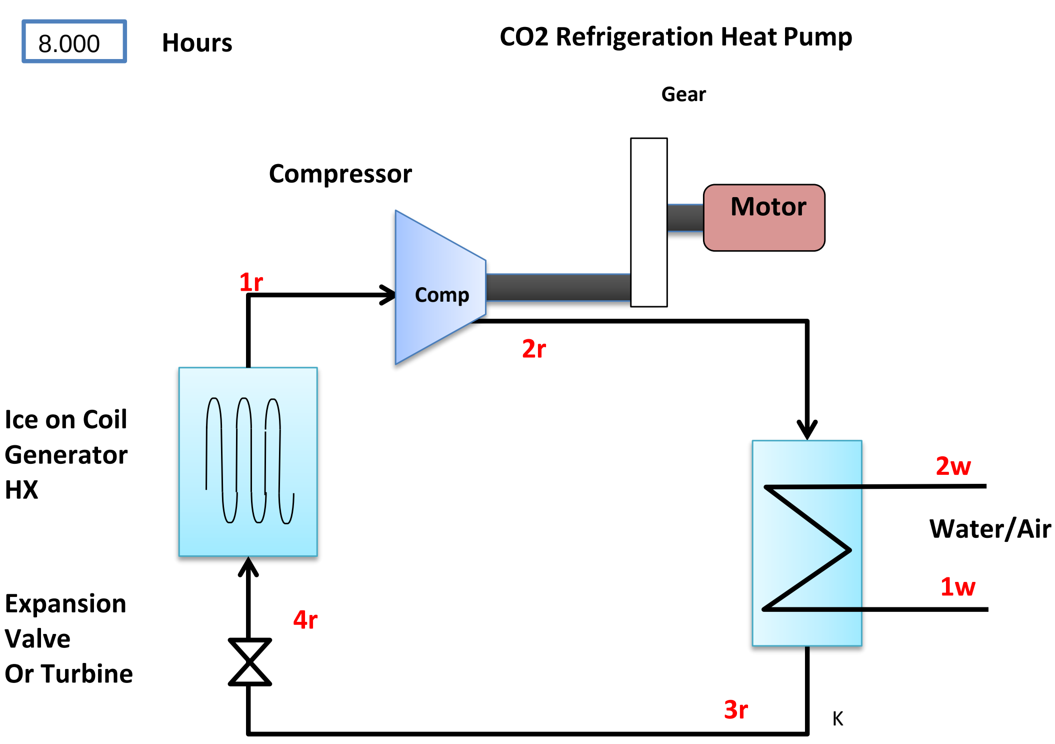

Figure 3: Process flow diagram for the SCO_2 refrigeration system. The illustration shows an expansion valve in the loop, however an alternative is to use a turbo-expander, which is capable or recovering some of the power required to run the compressor.

The proposed ice generating refrigeration system uses a pressure ratio of 3.0, with a flow rate varying from 47.7-50.1 kg/s of CO_2. The net Coefficient of Performance (COP) is estimated to vary from 2.97 to 3.34 depending on whether the expansion valve or the turbo-expander is used. The refrigeration system produces 8445 kW of cooling power or about 2401 tons of refrigeration. The corresponding refrigeration power rating is 1.05 – 1.3 kWe/ton-refrig to make ice.

The waste heat rejection in the refrigeration loop is performed at temperatures that vary from 300 K (26.8 C) up to 359.2 K (86 C). The high temperature of heat rejection means that most of the heat rejection can be performed with dry air, but the requirement to cool the CO_2 to 300 K (26.8 C) requires a cold climate, or water obtained from an evaporative cooler.

Table 3: Summary performance data for the S CO_2 transcritical refrigeration cycle.

CO2 REFRIGERATION	1r Comp Inlet	2r Comp Out	3r CO2 Hot Gas Chiller Exit	4r Expansion Valve Turbine Exit
EXPANSION VALVE				
Temperature (K)	268.2	359.2	300.0	268.5
Pressure (kPa)	3046	9138	8986	3071
Fluid Type	CO2	CO2	CO2	CO2
Mass Flow Rate (kg/s)	50.14	50.138	50.138	50.138
TURBINE EXPANDER				
Temperautre (K)	268.2	359.2	300.0	268.5
Pressure (kPa)	3046	9138	8986	3071
Fluid Type	CO2	CO2	CO2	CO2
Mass Flow Rate (kg/s)	47.7	47.7	47.7	47.7

a) Temperature and Pressure of CO2 Refrigeration: eff.comp=.80 , eff.turb=.85

CO2 REFRIGERATION	Expansion Valve	Turbo Expander
COP (Cycle)	2.973	3.683
COP (Net)	**2.694**	**3.345**
Compr Pwr (kW)	2841	2703.21
Turbine Pwr (kW)	na	409.95
Net Elect Pwr (kWe)	3135.05	2524.81
Heat Reject -Hot (kWth)	11323	10773
Evap Cooling ICE Gen (kWth)	**8445**	**8445.5**
Mass Flow Rate	50.14	47.70
Pressure Ratio	3.00	3.00

b) Summary performance parameters for SCO2 transcritical refrigeration cycles.

SCO_2 POWER PEAKING AND ENERGY STORAGE SUMMARY

The goal of the proposed waste heat recovery power cycle is to use ice-energy-storage to increase the amount of dispatchable energy that is available during periods of high demand and high costs. This is accomplished by using energy-storage to shift the time when excess renewable energy stored as ice (night and early morning hours) can be used to produce even more energy by recovering some fraction of the stored ice-energy, but also by increasing the heat utilization of the high temperature (538 C) waste heat source, and by increases in the SCO_2 power cycle efficiency during periods of peak demand (late day time and early evening).

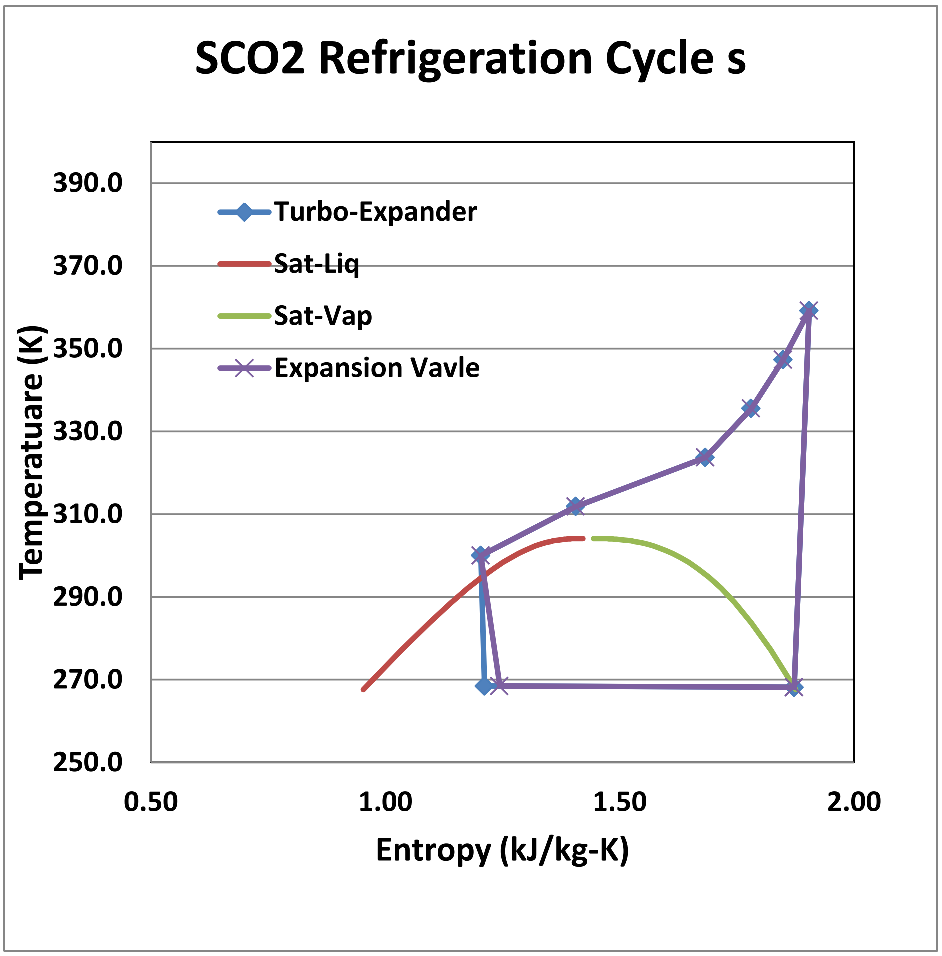

Figure 4: Temperature-Entropy Diagram for the SCO_2 Ice generating refrigeration system with either an expansion valve (purple) or a turbo-expander (blue). The net COP varies from 2.69 to 3.35 for the expansion valve and the turbo-expander respectively.

The ability to use ice-energy-storage to shift energy production to a later time is illustrated for this concept in Figure 5. This figure shows the rate of power production over a 24 hour period for this plant. As shown, there are three periods of different net power production. On-peak power that is produced over 4 hours is 9.3 MWe, off-peak power is 5.6 MWe for a 12 hour period, and the net power during the

8 hours of ice-making or charging cycle when the 5.6 MWe produced from the WHR Brayton cycle is reduced by 2.6 MWe to run the turbo-compressor in the turbo-expander refrigeration cycle. The resulting net power during the ice-production time period (8 hours) is therefore 3.0 MWe.

The corresponding electrical energy produced by these three different operating modes is illustrated in Figure 6 which shows the results assuming that a turbo-expander is used in the refrigeration system. Column A shows that 37 MWh of peak power is available for 4 hours (at 9.3 MWe). The value 37 MWh is the amount of dispatchable energy that is provided by this plant. Column B shows the gross energy that is available 111.6 MWh over a 20 hour period (12+8 hours) from the Brayton cycle, while Column C shows the electrical energy 20.2 MWh required to make ice by running the turbo-compressor in the turbo-expander refrigeration system. Thus 183% (37/20.2) of the electrical energy used to make the ice can be dispatched to the grid at any time after the ice is made. The available energy is greater than one because the recovered energy comes from the high temperature waste heat stream (not from the low temperature waste heat-of-rejection used in the refrigeration system). A summary table of these energy storage values is provided in Table 4. This table includes results for both the SCO_2 refrigeration system that uses an expansion valve and a turbo-expander.

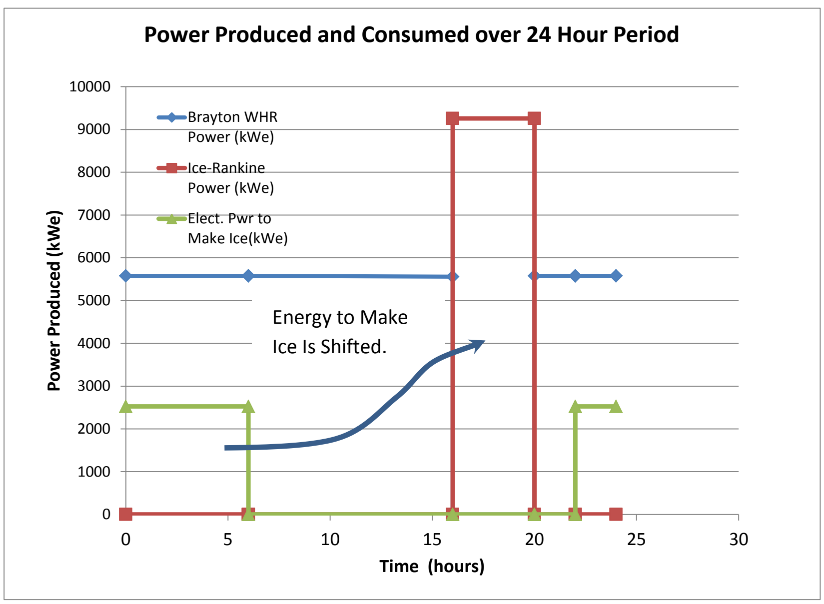

Figure 5: Power produced by the WHR bulk energy storage plant over a 24 hour period. Note the 9.2 MWe that are available/dispatchable for 4 hours compared to the 5.57 MWe that are available during 12 hours, and 3.05 MWe (5.57-2.5 MWe) available for 8 hours during the late night and early morning hours. Values are based on using a turbo-expander for the refrigeration cycle.

ROUND TRIP EFFICIENCY

The figure of merit for energy storage systems it the round trip efficiency. It should be noted that the system described in this report only uses ice-energy storage. The system as proposed does not store the hot thermal energy from the refrigeration process to be recovered as would be required in an ideal thermal energy storage plant [(6)]. Instead, a higher quality waste heat source at 538 C (from a gas turbine or industrial heat source) is used for power production, which means that the use of stored ice-energy allows better electricity recovery of the waste heat by increasing the waste heat utilization and by increasing the SCO_2 power system efficiency. Therefore, the definition of round trip efficiency needs to be clearly defined.

In this report two terms are used for round trip efficiency; they are *dispatchable round trip efficiency* and *excess dispatchable round trip efficiency*. Based on the preceding paragraph, the dispatchable round trip efficiency is the ratio of the energy produced (37 MWh) to that used (20.2MWh) or (37/20.2) = 183%. The dispatchable round trip efficiency exceeds 100% because the dispatchable energy includes energy recovered from the ice, but also additional energy due to better waste heat utilization, and better thermo-dynamic cycle efficiency.

Another way to define the round trip efficiency for this system is to look at the excess dispatchable energy. The excess dispatchable energy is the difference between the energy produced during ice-melting discharge cycle (column A in Figure 6, 37 MWh) minus the energy that would have been produced without ice storage (column D in Figure 6) 5.6 MW*8 hours=22.3 MWh). So this excess dispatchable energy is 37 MWh – 22.3 MWh, or 14.7 MWh which is shown in column E of Figure 6 . In this case the excess round trip dispatchable efficiency is the ratio of excess energy produced ÷ energy required for ice-energy-storage, or column E ÷ column C =14.7 MWh / 22.2 MWh which is 72.3%. In other words, the ice-energy storage concept allows this plant to shift 72.3% of the energy required to make the ice to a different time period and increase the power by 66% for the proposed ice-energy-storage concept using the turbo-expander SCO2 refrigeration system.

Regardless of which definition is used, all of the thermal energy comes from the external waste heat source. In an ideal energy storage system [6] the heat source would use the heat that was rejected from the refrigeration system. However, for the cycle proposed here the industrial waste heat source offers better quality heat so it is used instead, because more of it can be turned into electricity. So in essence the process proposed here is a hybrid type of energy storage. In the cycle discussed in this paper, waste heat from a high quality industrial heat source is converted to electricity; however, it is also possible to use heat from renewable fuels, solar energy, or even fossil or nuclear fuels. The important point is not; what is the proper definition of round trip efficiency, but can the system be used to provide some economic benefit to the utility and the consumer?

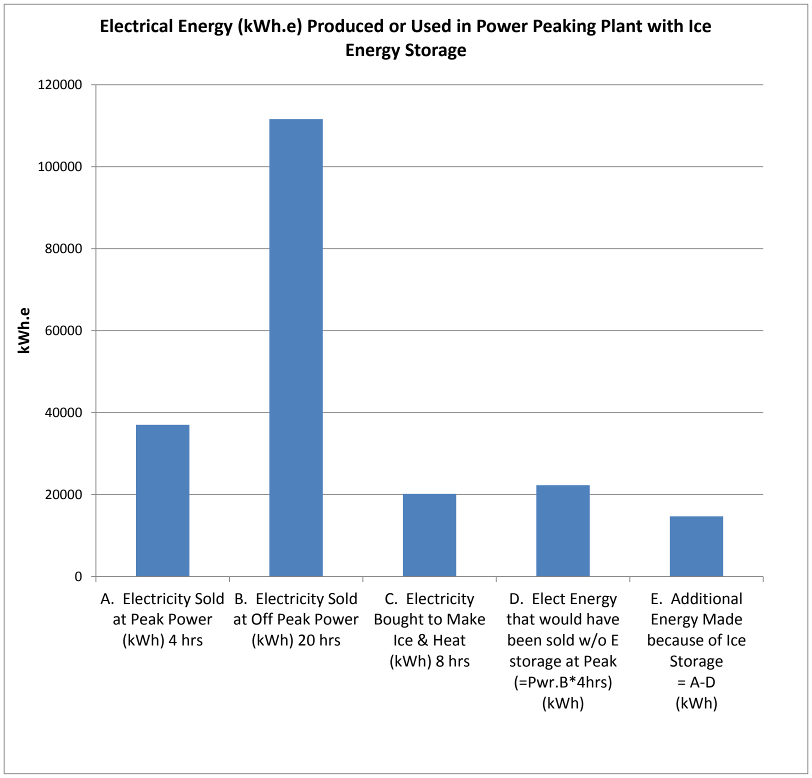

Figure 6: Electrical energy produced or used over a 24 hour period plant using uses bulk ice-energy-storage. Column A is the amount of dispatchable energy that is available at peak demand over a 4 hour period. Column B is the amount of energy that is available over a 20 hour period. Column C is the amount of energy purchased from B (or from the grid) over an 8 hr period. Values in column C are for the turbo-expander refrigeration system.

ECONOMICS

The time variability of electricity demand and the increasing supply of intermittent renewable power play a very important role on the price of electricity and stability of the grid. This paper describes a SCO_2 power system that uses waste heat, offers dispatchable power peaking capabilities, and bulk energy storage by making ice during low periods of demand.

A power plant of this nature offers a unique business approach to deal with the time variability of electrical demand because it offers a site independent plant that stores energy (as ice), it provides a way to increase power production by 66% over a four hour time period, it can dispatch the power at any time after the ice is made, it operates 24 hours per day; thus, it is always spinning, and because it uses waste heat as the heat source the dispatchable round trip efficiency is varying from 148% to 183% above the energy required to make and store ice, and requires no additional waste heat.

Table 4: Summary of energy storage and power peaking capabilities for CO_2 refrigeration with an expansion valve and with a turbo-expander. The dispatchable Round trip efficiency (ratio of dispatchable power to power purchased to make the ice) is 148% or 183% with respective to use of an expansion valve or turbo-expander.

Energy Storage Summary		
Duration of Discharge Cycle Operation w EvapRefrig	*hrs*	*4.000*
Q.th Energy Melting Ice	**kWh.ice**	**67563.7**
Waste Heat Temp	K	811.15
A. Electricity Sold at Peak Power	kWh	37021.08
Electricity Sold at Off Peak Power	kWh	111588.03
Electricity Bought to Make Ice & Heat	kWh	25080.38
Electrical Energy that would have been sold w/o E storage at Peak	kWh	22317.61
Additional Energy Made because of ice storage	kWh	14703.48
Round Trip Eff **(excess dispatchable Rnd Trip Eff)**		**58.6%**
Peak WH-Co2 Eff		68.0%
Off Peak WH-CO2 Eff		44.7%
Peak CO2 Net Eff		34.5%
Off-Peak CO2-Net Eff		31.7%
Peak Stack Exit Temp (K)	K	462.22
Off Peak Stack Exit Temp (K)	K	581.73
Waste Heat Q.th	kW	39399.66
Power Peak	**kW**	**9255.27**
Power Off-Peak	**kW**	**5579.40**
Power Refrig	**kW**	**3135.05**
Ratio of Energy Produced/Consumed		5.93
Ratio of Dipatchable Energy / Consumed **(Disp. RoundTrip Eff)**		**1.48**
Using TurboRefrigeration Cycle		
Duration of Discharge Cycle Operation w TurboRefrig	*hrs*	*4.000*
Net Power Required to make ice	kWe	2524.81
Q Energy Melting Ice	kWh.ice	**67563.7**
Electricity Bought to Make Ice & Heat	kWh	20198.51
Round Trip Eff **(Excess dispatchable Rnd Trip Eff)**		**72.8%**
Ratio of Energy Produced/Consumed		736%
Ratio of Dipatchable Energy / Consumed **(Disp. RoundTrip Eff)**		**183%**
Volume 50% ice	m^3	1458.43
Tank Diam for a height of 5 m (5 x 20 m)		19.27

A companion paper discusses more of the economics incentives and issues for the proposed SCO_2-WHR bulk energy storage plant, and it is presented in a separate report in this conference. This report shows three things:

1. It shows that the ability of this SCO_2 power cycle concept to use a high temperature waste source such as from an industrial process such as metal smelting or the exhaust from a gas turbine, is very economic and has very short time periods for the return on investment (< 3 years).
2. The unique ability to store and dispatch large quantities of electricity (10's of MW_e) for 4 hours or more offers many substantial benefits over existing power peaking and energy storage technologies provided the time variability of power peaking is adequately factored into the price of electricity. In the companion paper it was shown that the SCO2 WHR ice-energy storage system can increase the return on investment by about 17% over a plant that offers no storage.
3. Further, because the plant is always operating (spinning) it operates at higher average efficiency continuously is not an under-utilized capital asset for the utility.

CONCLUSIONS

The economics of the total SCO_2-Waste Heat Recovery using ice-energy-storage plant are improved because the plant operates continuously. Thus, revenue is continuously created either by the WHR Brayton cycle during normal operations (12 hours), during the charging cycle (8 hours), and during the discharge cycle (4 hours). During these three time periods 40 MW_{th} of waste heat produces 5.7 MW_e for 20 hours, consumes 2.6-3.2 MW_e during charging, and produces 9.4 MWe during discharging. The additional 20 hours of revenue generation (that occur during normal and charging operations) mean that the SCO_2-WHR energy storage plant is not underutilized over a 24hour period. The economics are further improved because the colder heat rejection temperature during the discharge cycle greatly improves the waste heat utilization, the SCO_2 cycle efficiency, plus the increased electric power. Additionally, the plant also provides “spinning reserve” capabilities. And lastly, because the plant is always running, it is hot. This means the plant can be switched from normal operations, to charging or discharging simply with the use of valves and alternative flow paths to operate as needed.

REFERENCES

1) Personal communication, Robert Fuller, bfuller@barber-nichols.com, Barber Nichols Inc. Arvada, Colorado, 2012.
2) Personal communication, Mario Gaia, mario.gaia@turboden.it, 2011.
3) David Montgomery, Caterpillar ARES Program Manager, “An Otto Rankine Combined Cycle for High Efficiency Distributed Power Generation”, https://www.erc.wisc.edu/documents/symp09-Montgomery.pdf, June 10, 2009.
4) C. Davidson, S.A. Wright, R. Fuller, “Compressor Inter-recuperation, Patent Allowance”, SuperCritical Technologies Inc., May, 2014.
5) Evapco Inc., “Thermal Ice Storage: Application & Design Guide”, Taneytown, MD 21787, 2007.
6) S. A. Wright, A. Z’Graggen, J. Hemrle, “Control of a Supercritical CO2 Electro-Thermal Energy Storage System”, Proceedings of ASME Turbo Expo 2013, GT2013-95326, San Antonio, Texas, USA, June 3-7, 2013.