A Novel SCO_2 Primary Cycle for Air-Combustible Fuels

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David is a twenty-nine (29) year veteran of aerospace engineering with expertise in turbo-machinery design and analysis (military and commercial, aero and power-gen), UAV aircraft design (both rotary and fixed wing), structural design and analysis including stress and life, vibration, and buckling analysis in a multitude of industries. He holds a BSME from the University of Colorado.

David began his career with GE Aircraft Engines in 1985 working on such projects as T700, GE36, CFE738, F412, GE90, and CF34-8C engines designing rotating engine structures (high and low pressure compressors, high and low pressure turbines). David also conducted the life re-certification analysis for the CT7-5 LPT and similar work for the CT7-9 LPT. In 1996 David founded Peregrine Consulting, Inc. and has served as an expert consultant to name brand turbomachinery OEMs since.

ABSTRACT

Current supercritical carbon dioxide Brayton cycles are recuperated cycles because of the large amount of heat left after expansion through the turbine. The high level of efficiency of SCO_2 cycles is largely owing to the use of recuperators to recapture this heat. However this characteristic has made it difficult to incorporate closed-cycle SCO_2 designs in applications where air-combustible fuels such as fossil fuels are the source of heat.

For air-combustible heat sources the ability to transfer the heat of combustion into the cycle is greatly inhibited by the high air-to-SCO_2 heat exchanger SCO_2 inlet temperature which has been substantially raised by preheating within the recuperator. The results are fairly high flue exhaust gas temperatures and low cycle efficiencies that are not very competitive with existing gas turbine technologies. This problem doesn't exist for solar and nuclear type heat sources; applications where SCO_2 has been embraced enthusiastically.

In this paper, Peregrine Turbine Technologies, LLC presents a novel proprietary cycle designed to employ air-combustible fuels with a closed SCO_2 Brayton cycle design, while achieving high thermal efficiencies. The ability to generate shaft power using fossil and other air-combustible fuels including biomass in primary cycle will facilitate unprecedented fuel efficiency in applications ranging from power generation, naval propulsion, oil and gas industry compressor drives, locomotive drives, automotive and eighteen wheeler drives, and more. The ability to achieve high thermal efficiencies in smaller powerplants using air-combustible fuels will greatly impact life-cycle costs in all of these applications while substantially reducing emissions.

INTRODUCTION

The challenge of using fossil-fuel combustion as a heat source for closed cycle turbines regardless of the thermodynamic fluid is generally the difficulty of exchanging the heat of combustion into a closed system where the compressor discharge temperature is already fairly high. This is true of ideal gas type fluids due to the isentropic temperature rise caused by compression, but it is a greater problem still for systems that employ recuperation.

SCO_2 cycles characteristically have low compressor discharge temperature when compared to an ideal gas, owing to the low enthalpy rise to achieve a given pressure ratio. This characteristic, together with the long slender nature of the T-s diagram as shown in Figure 1, make SCO_2 an ideal candidate for recuperation.

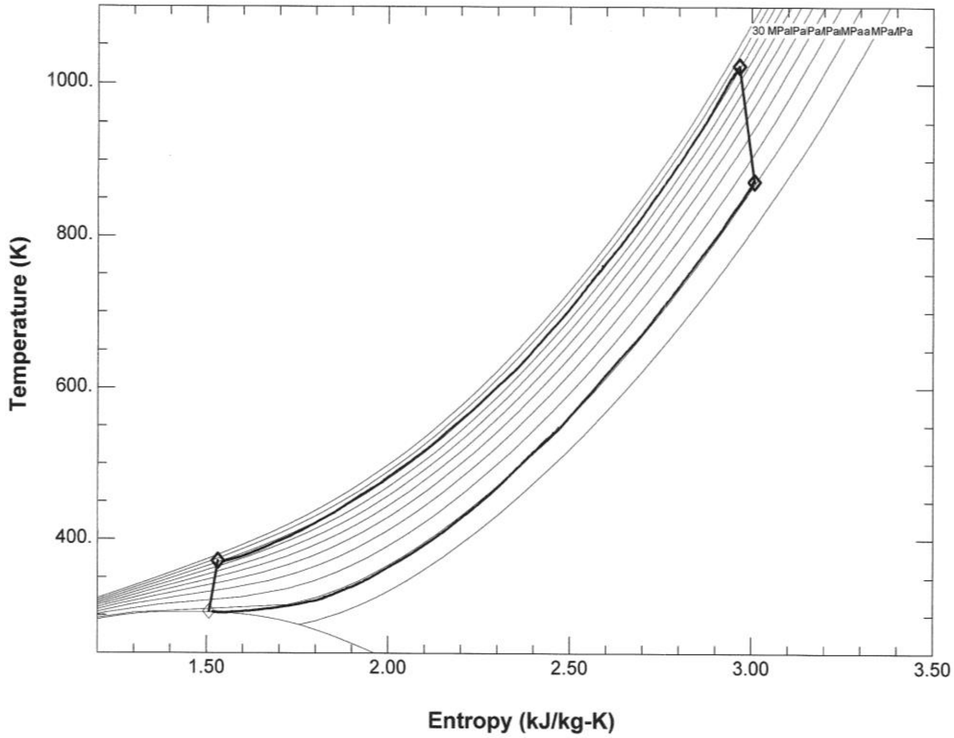

Fig. 1 T-s Diagram for Simple Brayton Cycle from NIST REFPROP

The Peregrine Turbine cycle presented here takes a different approach to exploiting the heat recovery opportunity inherent in the SCO_2 T-s diagram. Consider a simple unrecuperated Brayton cycle employing SCO_2 as the working fluid. Compressor discharge temperatures on the order of 370°K – together with high-effectiveness counter flow heat exchangers - would make it possible in this simple cycle for sufficient heat to be transferred to the SCO_2 to drop the post-heat-exchanger flue gas temperatures to roughly 380°K indicating successful capture of the majority of the available heat. Unfortunately in spite of the improved heat capture, the total thermal efficiency for that simple Brayton cycle would only be on the order of 13% because of the large amount of heat rejected at the SCO_2 chiller after turbine expansion.

Peregrine developed a cycle that alleviates this problem by employing two Brayton cycles that work in tandem, exchanging heat back and forth in a way which results in substantially higher thermal efficiency. Figure 2 illustrates the Peregrine Turbine cycle diagram. It can be seen that heat rejected by the closed-cycle SCO_2 turbine at the low pressure heat exchanger is used to preheat the compressor discharge air of the open-cycle air turbine. The temperature rise required at the burner is then only about 187°K. The heated, pressurized air is then sent to the gas generator turbine which drives the airside compressor. From there the flow goes to the high pressure heat exchanger where most of the heat is passed to the compressor discharge flow of the SCO_2 compressor. The result is a sort of circular flow of heat – or thermal flywheel - within the system with small amounts of heat added at the burner and small amounts of work taken out at the SCO_2 turbine. The advantage here is very high thermal efficiency in simple cycle. This cycle was analyzed using a commercial code NPSS, a cycle analysis tool developed by a consortium of turbine engine manufacturers led by NASA Glenn Research. Assuming heat exchanger effectiveness of 98%, the cycle shown in Figure 2 achieves an analytical thermal efficiency of approximately 49% for a turbine inlet temperature of 1023°K and a pressure ratio 3.5.

The intercooler, shown in Figure 2 at the air compressor discharge, functions primarily as a means to increase the approach temperature at the low pressure heat exchanger inlet and reduce the tendency toward a pinch point problem there. This could also be accomplished with a similar cooler or chiller between the low pressure heat exchanger and the SCO_2 compressor inlet. Either way, only a small amount of heat must be rejected at the intercooler, on the order of 2.5% of the heat exchanged at the low pressure heat exchanger. This can be accomplished with an evaporative cooler or other means. For operations below standard day temperatures the need for the intercooler diminishes and becomes unnecessary below about 273°K.

In order to sustain the cycle, about 12 kW of total heat must be exchanged (heat in and heat out combined) for each kW of power output. Therefore the cycle is sensitive to heat exchanger effectiveness in much the same way that an energy storage flywheel is sensitive to bearing losses.

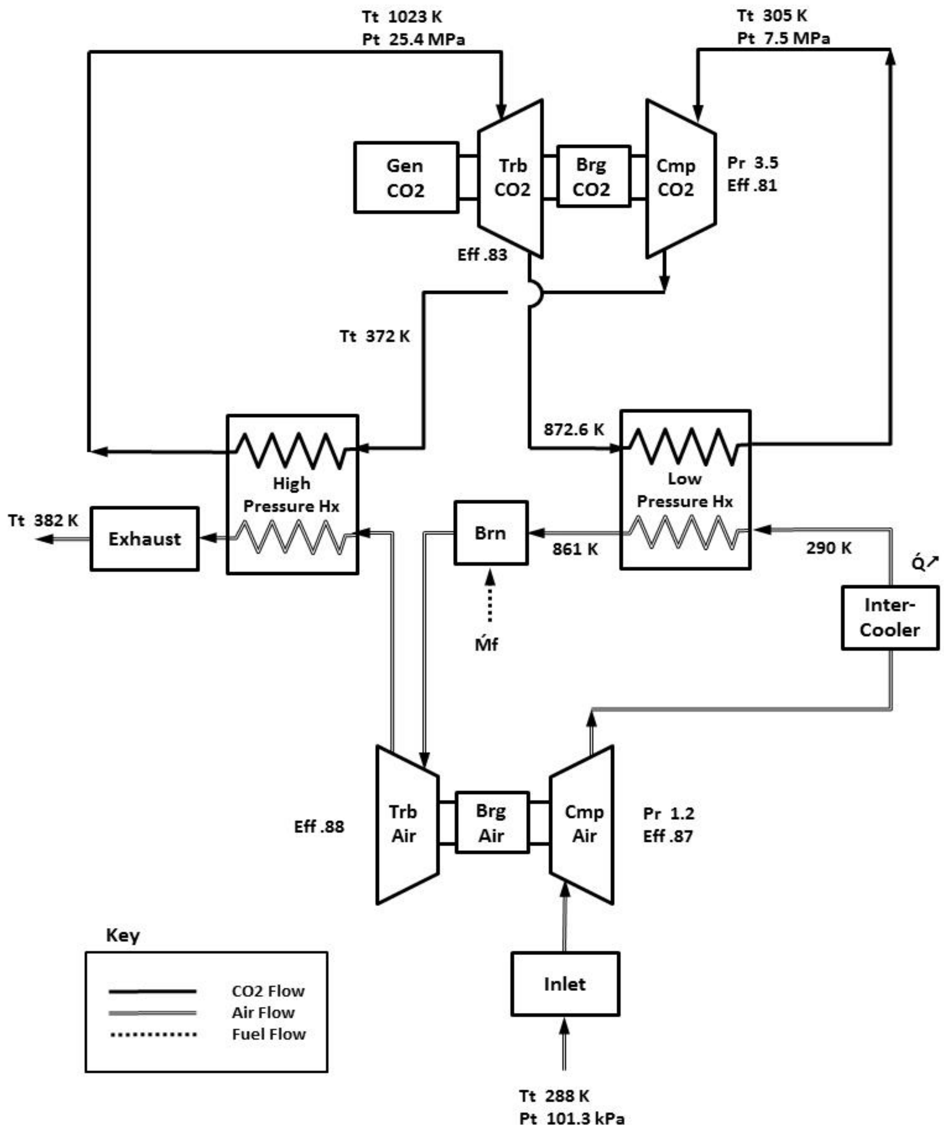

Fig. 2 Peregrine Turbine Proprietary Cycle

All of the heat exchange occurs between air and SCO_2 which is challenging due to the large disparity between convective heat transfer coefficients inherent to each. Likewise, there are challenges to matching the heat capacity rates as shown in Figure 3 and 4.

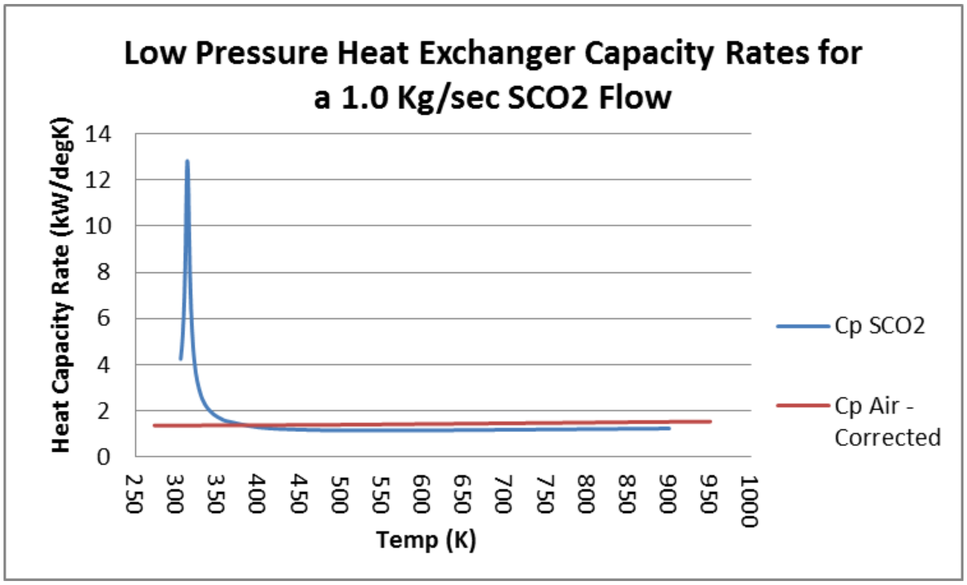

Fig. 3 Heat Capacity Rates for Low Pressure HX

For the low pressure heat exchanger the characteristic pseudo-critical point is evident from the spike in the heat capacity rate for 1 Kg/sec of SCO_2 at the exit of the heat exchanger. This spike results in the much-discussed pinch point inherent in this kind of heat exchanger scenario. This mismatch can be partly alleviated by controlling the air inlet temperature to the heat exchanger which is the function of the intercooler.

It is common to see Printed Circuit Heat Exchangers (PCHE) employed as recuperators for SCO_2 cycle applications due to their compact nature and the high effectiveness values that are achievable. The author has noted effectiveness values as high as 98.7% in published materials. Naturally, the higher the effectiveness the higher the cost and weight of a given heat exchanger and generally the higher the associated pressure drops. The monolithic nature of a PCHE presents challenges to employing it where large temperature swings are anticipated as is the case in the Peregrine Turbine. Likewise, air-to-SCO_2 heat exchange and the substantial mismatch in density of the two fluids present still greater challenges. It is evident that advancement in heat exchanger technology is necessary to achieve cost, weight and durability goals while preserving high heat exchanger effectiveness and low approach temperatures. Peregrine Turbine Technologies has developed proprietary heat exchanger technology designed to address those issues.

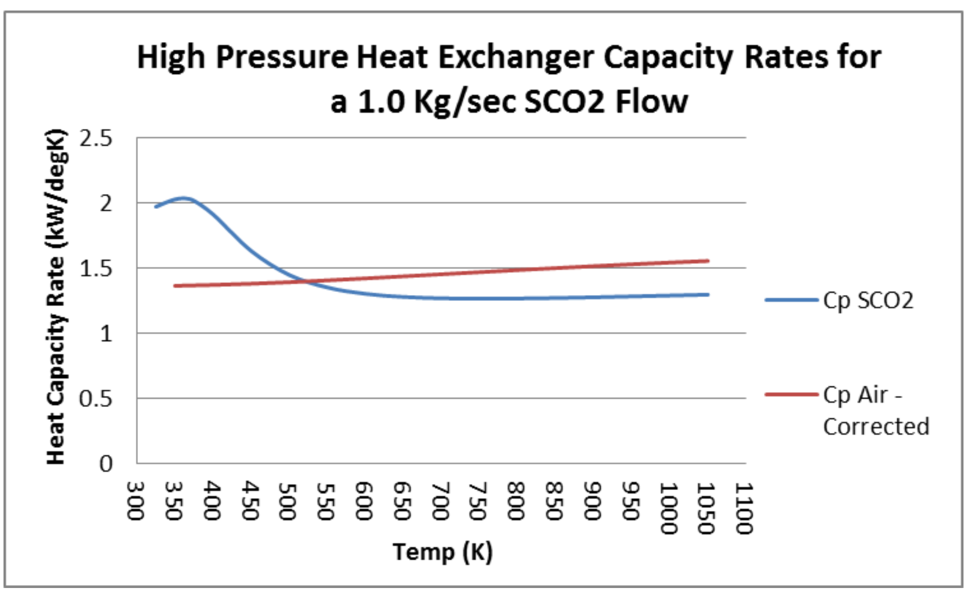

Fig. 4 Heat Capacity Rates for High Pressure HX

Figure 4 shows a much lower characteristic spike in heat capacity rate due to the fact that the abrupt changes in thermal capacity for SCO_2 are much less pronounced at higher pressure.

From analyses of various heat exchanger designs using an internally developed code called SCI-HEX, we found that the pinch point on the low pressure heat exchanger was significant when using air as the heat sink medium. Various results can be seen in Figure 5. HX1 and 2 are both high pressure heat exchangers and achieve a targeted performance of 98% at a length of 70 to 80 rows of heat exchanger fins. HX1 however, which is the low pressure heat exchanger, does not achieve the desired 98% effectiveness at any reasonable size because of the pinch point phenomenon. The thermal efficiency of the cycle shown in Figure 2 is seriously compromised by reduced heat exchanger effectiveness and makes this cycle unlikely to perform well at a reasonable cost.

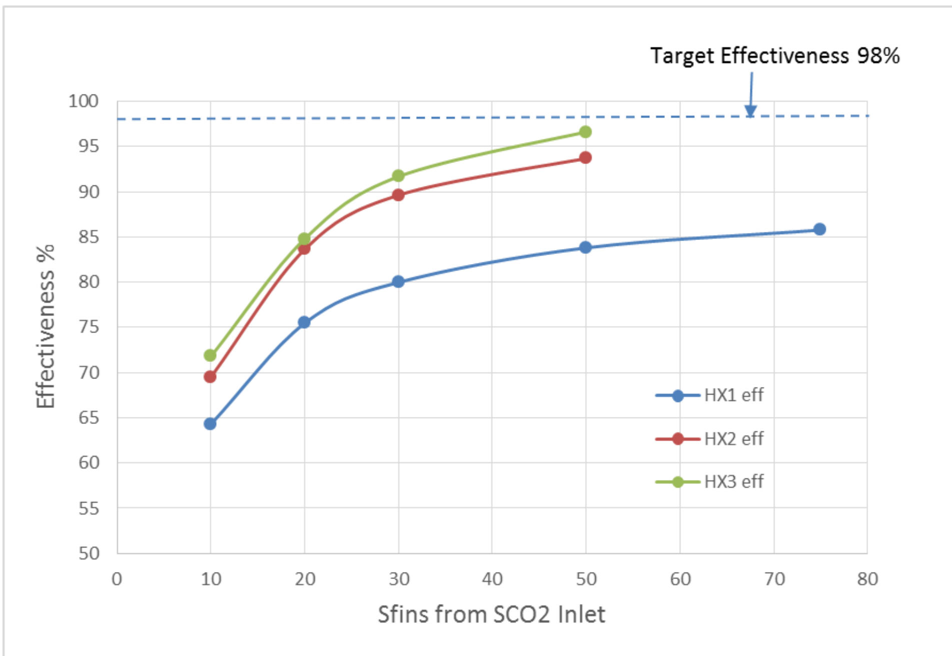

Fig. 5 Heat Exchanger Effectiveness by No. of Rows of Fins

The cycle shown in Figure 2 (which uses air as the heat sink and the heat source while maintaining low approach temperatures) also requires a larger mass flow rate on the air side than the SCO_2 side. Generally a ratio of about 1.3:1 is required in order to properly match the heat capacity rates of the two fluids. The airside compressor work required to drive that much mass flow is substantial. Based on these findings, a proprieatary partially recuperated cycle was designed and analyzed which alleviated the pinchpoint problem another way. By reducing the air mass flow rate and by using larger approach temperatures at each heat exchanger, the result is a substantial reduction in heat exchanger size for the same heat load. The partially recuperated cycle is shown in Figure 6. A notable difference is the inclusion of a recuperator which is designed to pass only a fraction of the full SCO_2 mass flow rate by adjusting valves C and D.

The total SCO_2 mass flow will pass through HX0 and HX2 but since the air mass flow rate in these heat exchangers is set to match heat capacity rates at HX1 and HX3, the result is a significant mismatch in thermal capacity rates at HX0 and HX2. This is desirable for two reasons. In HX0 this mismatch results in a very large approach temperature at the air inlet (as shown in Figure 7) in spite of the very close approach temperature at the air outlet. This mitigates the effect of the spike in Cp of the SCO_2 as it approaches its critical point near the compressor inlet conditions. A by-product of that is the elimination of the pinch point for HX1 and HX3 as shown in Figure 8.

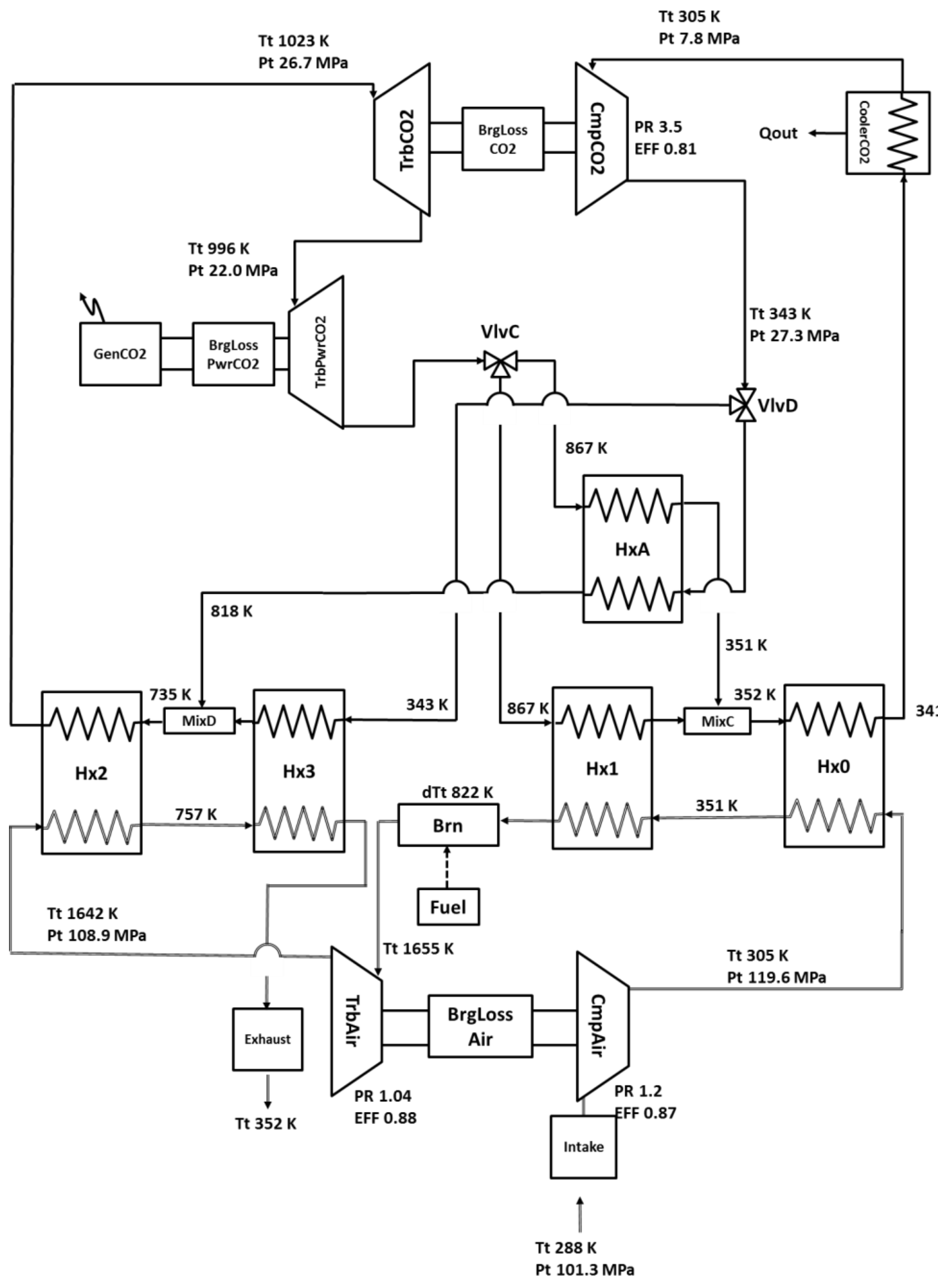

Fig. 6 Peregrine Proprietary Partially Recuperated SCO_2 Cycle

With this change of cycle configuration comes a modest reduction in achievable thermal efficiency at a given pressure ratio and turbine inlet temperature. However, the mass flow ratio between air and SCO_2 is approximately .33 for this configuration, resulting in more than a 75% reduction in air-to-SCO_2 heat exchanger size, weight and cost. In addition, the pinch point problem in the air-to-SCO_2 heat exchangers is alleviated. A design layout of this cycle configuration is shown in Figure 9.

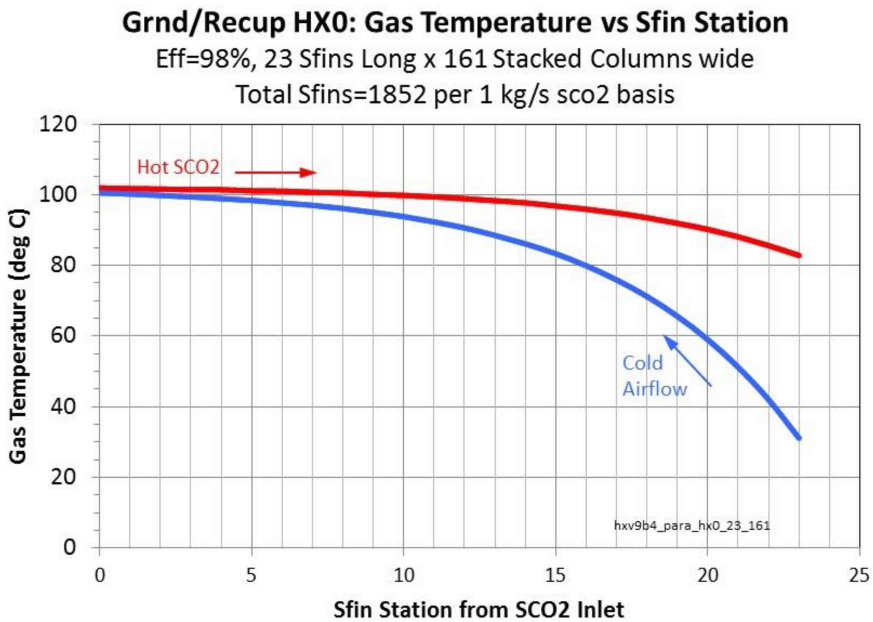

Fig. 7 Large Delta Temperature Alleviates Pinch Point

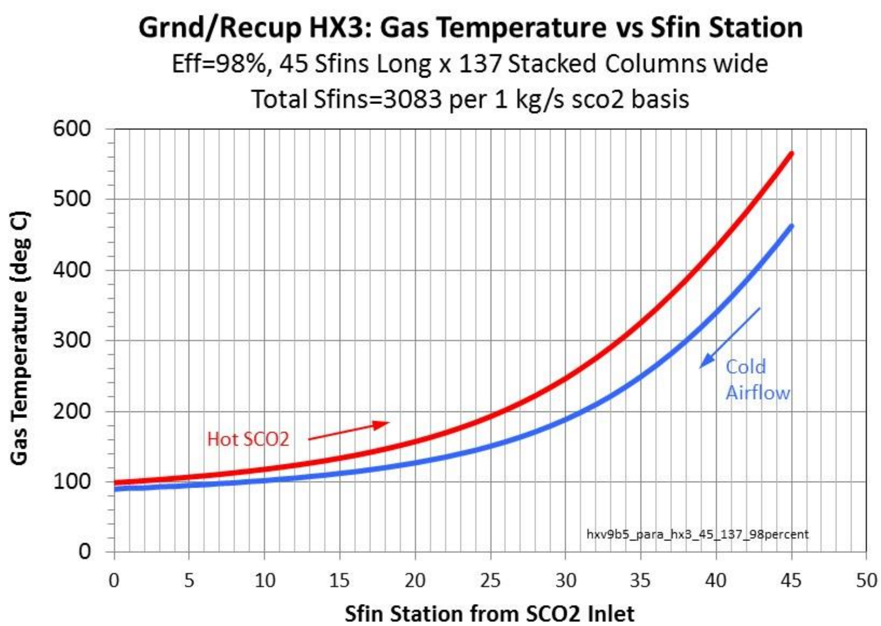

Fig. 8 Better Match of Heat Capacity Rate for HX3 also applies to HX1

The partially recuperated cycle was analyzed using NPSS and the statepoint conditions shown in Figure 6 and found to have a thermal efficiency of approximately 43%. The cycle does require additional SCO_2 heat rejection above what is rejected to the air heat sink at HX0 and HX1. Note the addition of an SCO_2 chiller before the compressor inlet. Very little heat is lost to the low-mass-flow exhaust gas which shows a temperature rise above ambient of just 64° C. Instead, the majority of rejected heat is transferred at the SCO_2 chiller which creates a substantial opportunity to recover heat via a

SCO_2-to-water heat exchanger, providing waste heat recovery in a form suitable for district heating or other combined heat and power (CHP) application.

Fig. 9 Peregrine Turbine Engine Configuration

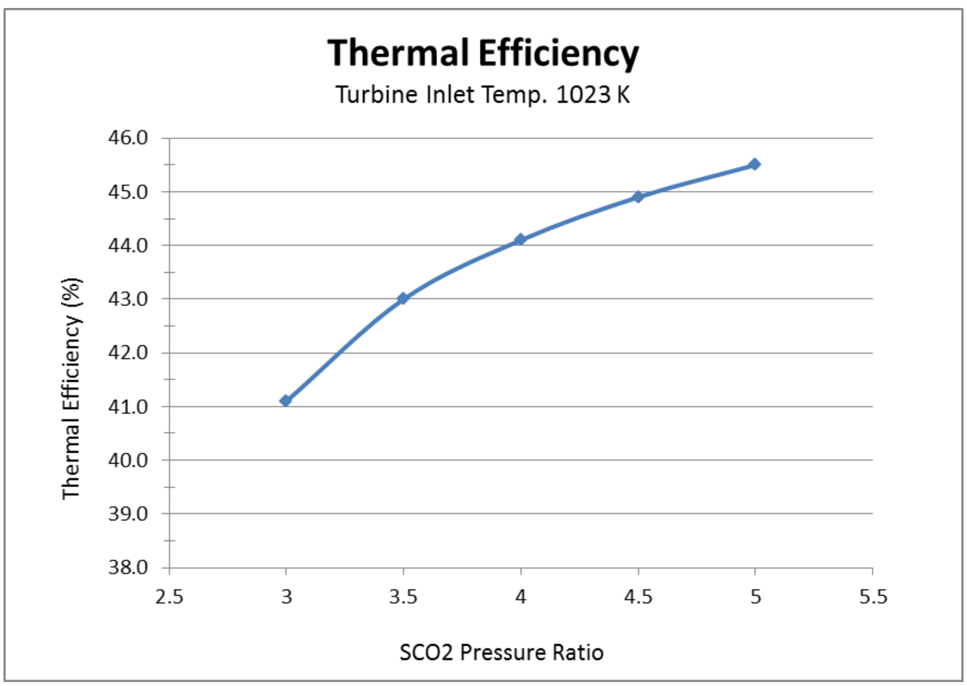

Fig. 10 Thermal Efficiency of the Partially Recuperated Cycle vs. Machine Pressure Ratio

Performance of the cycle as a function of SCO_2 pressure ratio is shown in Figure 10 for a turbine inlet temperature of 750° C. Improved thermal efficiency and specific power results from higher pressure ratios and turbine inlet temperatures both of which are limited by heat exchanger material creep life capability.

Peregrine Turbine Technologies, LLC is currently in the detail design phase of a privately funded program to produce a production prototype slated for installation in an existing industrial application.

CONCLUSION

The ability to use conventional fossil fuels, biofuels and any other source of heat to drive closed-cycle SCO_2 turbines creates opportunities to exploit the highly-favorable characteristics of SCO_2 including its low compressibility factor, favorable heat transfer and viscosity characteristics as well as the high specific power which drives dramatic reductions in turbo-machinery size, weight, cost, and part count.

In addition to the above, since high efficiency is possible at very low turbine inlet temperatures (i.e. 1000 K), turbine components can employ lower cost equiaxed grain materials without the need for blade cooling. As a result, expensive casting cores and the associated manufacturing steps are not necessary.

Since the purpose of the air-breathing side of the cycle is simply to generate hot gas, very low air pressure ratios are possible without compromising efficiency. Single stage axial compressors and turbines will suffice and will operate at lower speeds. The efficiency and specific power of the overall system will be largely unaffected by atmospheric pressures and temperatures such that high altitude or hot day operations will not be greatly compromised.

The cycle is fuel agnostic in that it matters little what the combustible fuel is. That being the case, a proprietary biomass version of the cycle has been proposed and is also under consideration.

The Peregrine Turbine cycle is patent pending and promises to substantially improve fuel efficiency in all applications where fossil fuels and biofuels are employed.

NOMENCLATURE

CHP = Combined Heat and Power
SCO_2 = Super-critical Carbon Dioxide

REFERENCES

NIST REFPROP Standard Reference Database, Ver 9.0, National Institute of Standards and Technology