Second-Law Analysis of a Hybrid Reverse Brayton Stirling Cryocooler

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ABSTRACT

Multistage cryocoolers are often used to cool infrared sensors for space applications. Using a Reverse Brayton cryocooler for the lower stage ensures low vibration due to only high frequency moving parts and allows for remote cooling separated from the compressor and Stirling upper stage. A second-law analysis of a Reverse Brayton cryocooler as the lowest stage in a multistage cryocooler is presented. Parametric studies were done on the losses in the lower stage. Parametric studies were also done on variations of the supported cooling load. The effects on hybrid system performance of variations in the operation of the Stirling cycle upper stage were modeled. Finally, losses due to the upper stage are studied.

INTRODUCTION

Prior work [1] has described the conceptual basis for how a hybrid Brayton-Stirling cryocooler system might be designed. Figures 1 and 2 reiterate its basic layout. The model described below seeks to provide an open-sourced model to parametrically analyze the hybrid system.

REVERSE BRAYTON STAGE MODEL

The model is for a Reverse Brayton cycle at the lower stage in a multistage cryocooler. Thus, a hot reservoir temperature of 25 K is assumed. Table 1 shows some other assumptions for various parameters, where $\dot{Q}_{leak,\ recuperator}$ is the heat leak from the recuperator to the surroundings and f is the fraction of work recovered from the turbine to the compressor.

Table 2 shows some results for a cooling load of .03 W at 15 K, where η_2 is the second-law efficiency, $\varepsilon_{recuperator,\ effective}$ is the recuperator effectiveness accounting for the heat leak, $\dot{W}_{in}$ is the net work into the system, and $\dot{W}_{out}$ is the net work out of the system. This case requires 0.403 W of work into the system and is about 5 % efficient. Figure 3 shows a *T-s* diagram for this case.

Figure 4 shows the losses for this case, where the first six columns are the losses for each component and then the overall system, the seventh column is the loss associated with the products (in this case, the cooling load of .03 W), and the final column is the loss associated with the required

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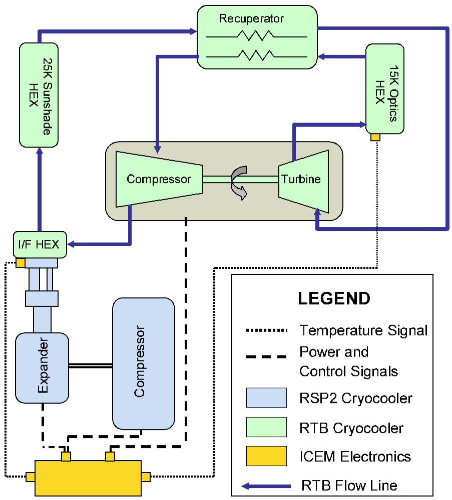
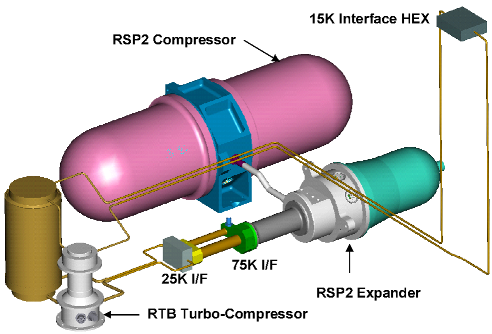


Figures 1. Functional diagram of hybrid system.



Figures 2. Physical layout of hybrid system.

Table 1. Assumptions.

Fluid	Helium
$\eta_{compresor}$	0.40
$\eta_{turbine}$	0.50
ε_{HHX}	0.99
ε_{CHX}	0.99
$\varepsilon_{recuperator}$	0.995
$\dot{Q}_{leak,\ recuperator}$ $[W]$	0.007
$P_{r,\ compressor}$	1.5
$P_{r,\ turbine}$	1.4
Pressure Drop in HX	0.01
Pressure Drop in Recuperator	0.02
f	0

Table 2. Results for cooling load of .03 W at 15 K.

COP	0.07435
η_2	0.04957
$\dot{m}$ [g/s]	0.007051
$\varepsilon_{recuperator,\ effective}$	0.9857
$\dot{Q}_H$ [W]	.398
$\dot{W}_{in}$ [W]	.403
$\dot{W}_{out}$ [W]	.035

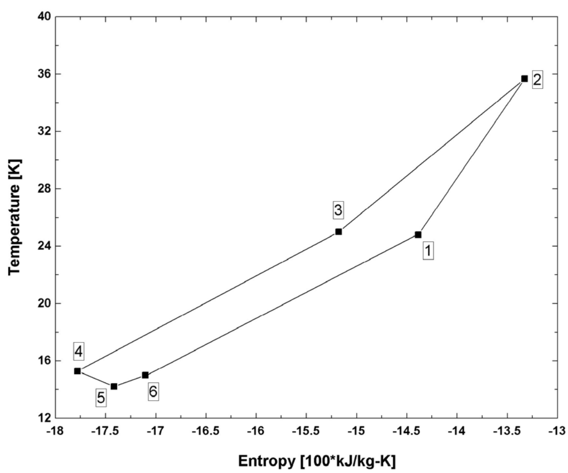


Figure 3. T-s diagram for cooling load of .03 W at 15 K.

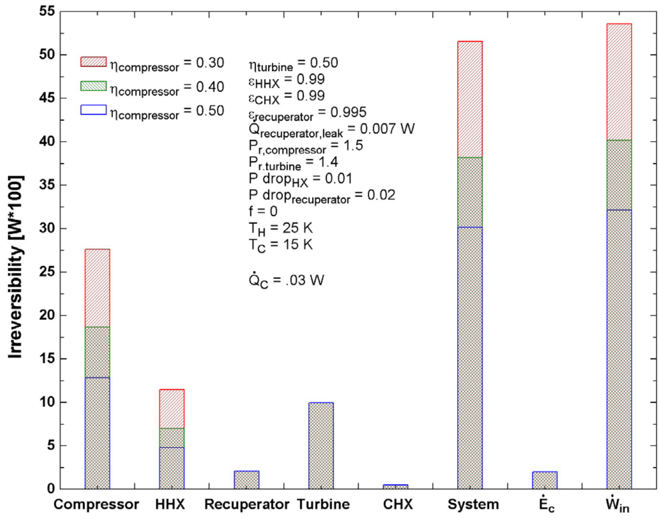


Figure 4. Losses varying compressor efficiency

Table 3. Percentages of the overall system loss for each component for cooling load of .03 W at 15 K

Compressor	48.86
HHX	18.31
Recuperator	5.424
Turbine	26.1
CHX	1.304

work into the system. The losses from the compressor and the turbine are the highest in the system because they are inefficient. The loss from the hot heat exchanger is higher than that from the cold heat exchanger because the rejected heat is being dumped, while the added heat is being reincorporated into the total energy available to the system. The overall system losses are about .38 W. The loss associated with the cooling load is .02 W and the loss associated with the required work into the system is about .4 W. Table 3 summarizes the percentages of the overall system loss for each component.

Parametric Studies on Second-Law Losses

Parametric studies were done to further investigate the losses in the system, varying compressor efficiency, turbine efficiency, recuperator effectiveness, heat leak from the recuperator to the surroundings, pressure ratio of the compressor, pressure drop in the cold heat exchanger and hot heat exchanger, pressure drop in the recuperator, the fraction of work recovered from the turbine, the hot temperature, the cold temperature, and the cooling load. Figures 5 to 9 show a subset of these losses.

Increasing the compressor and turbine efficiencies greatly decreases losses. Increasing the recuperator effectiveness by a small amount or reducing the recuperator heat leak also decreases losses. High pressure ratios produce lower losses than low pressure ratios. Lower pressure drops also result in lower losses. Increasing the fraction of work recovered from the turbine decreases the loss associated with the turbine and the work into the system. Higher hot temperatures and lower cold temperatures increase losses. Decreasing the cooling load by .02 W decreases all losses by about 30 %.

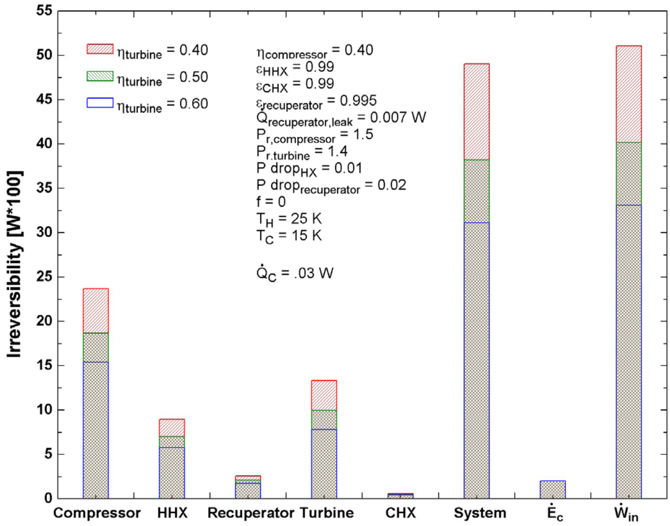


Figure 5. Losses varying turbine efficiency

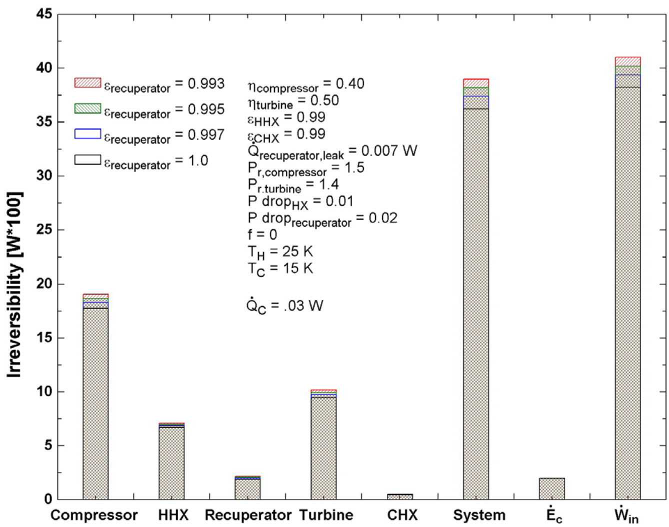


Figure 6. Losses varying recuperator effectiveness

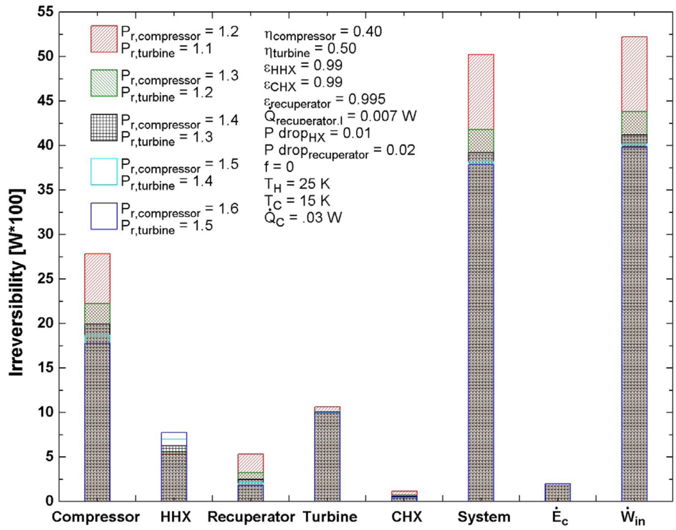


Figure 7. Losses varying compressor pressure ratio

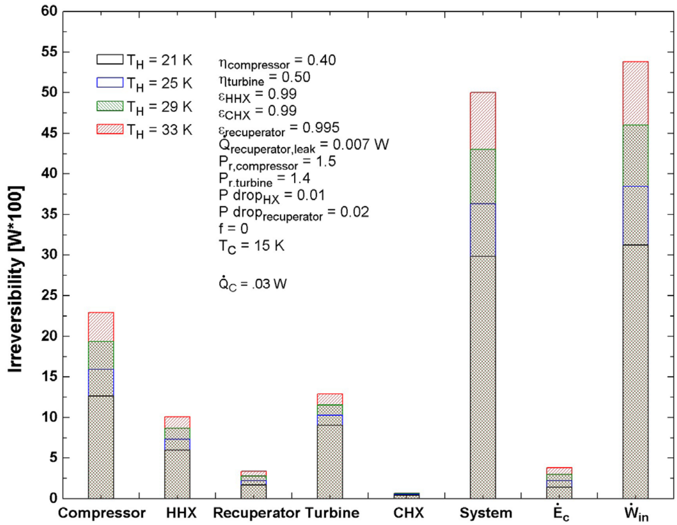


Figure 8. Losses varying hot side, reject temperature

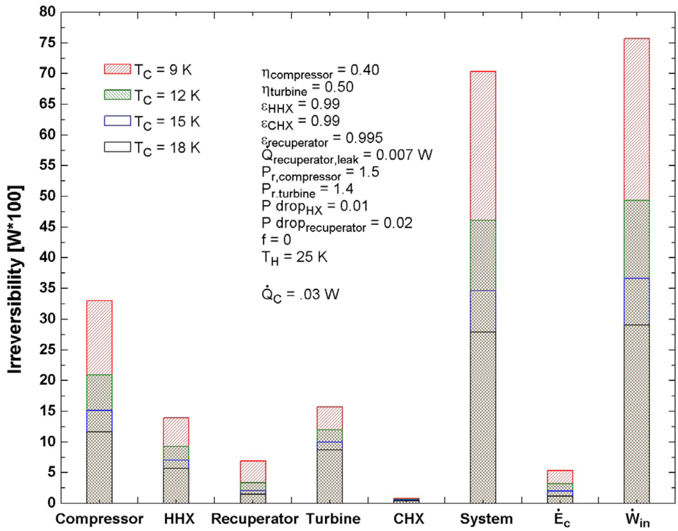


Figure 9. Losses varying cold temperature

Lower Stage Load Curves

To study the effects of changing various parameters on the cooler performance, load curves were generated varying compressor efficiency, turbine efficiency, HHX effectiveness, CHX effectiveness, recuperator effectiveness, heat leak from the recuperator to the surroundings, pressure ratio of the compressor, pressure drop in the HHX and CHX, pressure drop in the recuperator, the fraction of work recovered from the turbine, the hot temperature, and the work into the system. A subset of these are shown in Figures 10 to 16, respectively. Figures 17 and 18 illustrate the impact on efficiency of different cold end temperatures and heat rejection temperatures.

Increasing the compressor or turbine efficiency raises the load curve. Increasing the HHX effectiveness and recuperator effectiveness also raises the load curve. Changing the CHX effectiveness has little effect. An increasing heat leak from the recuperator to the surroundings causes a lower

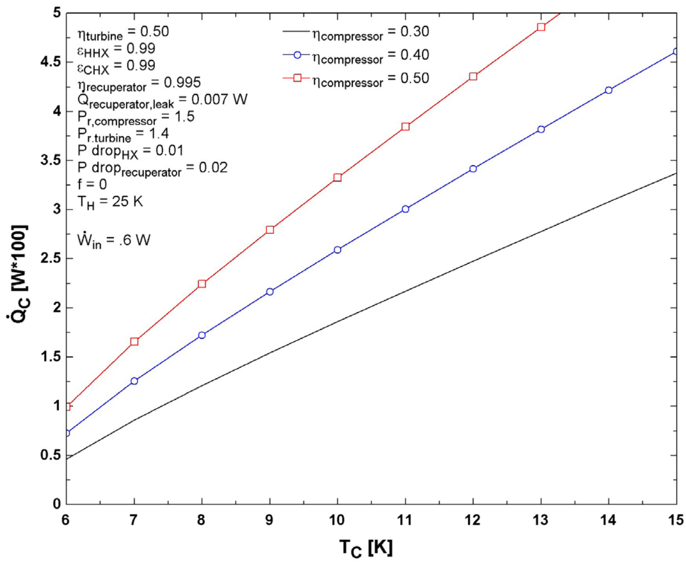


Figure 10. Load curves varying compressor efficiency

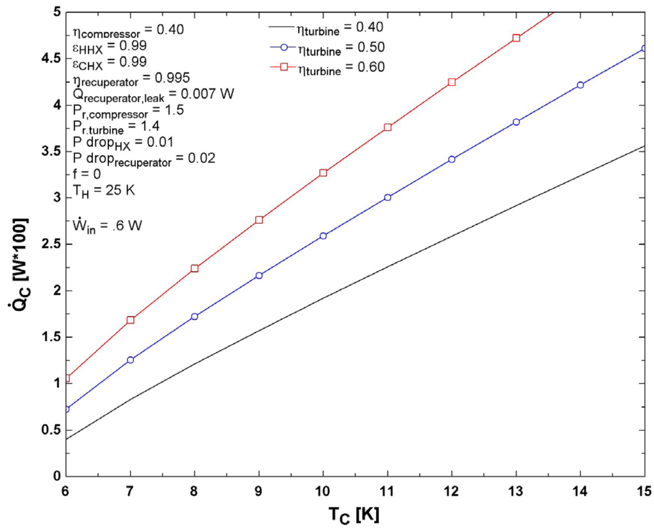


Figure 11. Load curves varying turbine efficiency

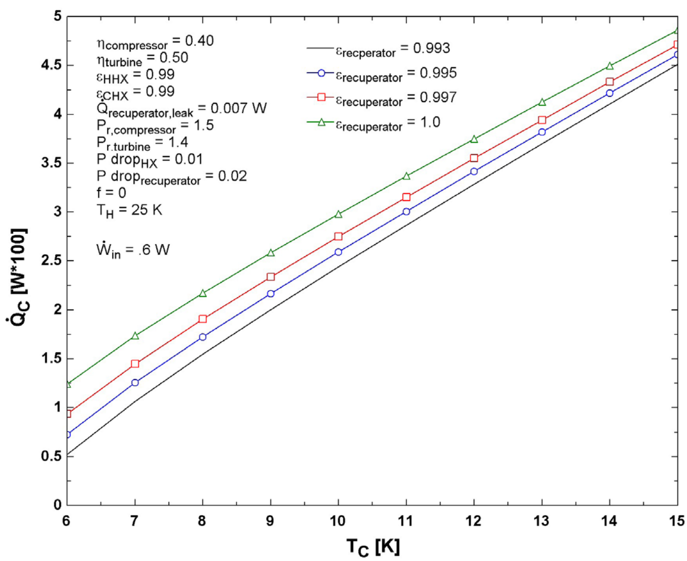


Figure 12. Load curves varying recuperator effectiveness

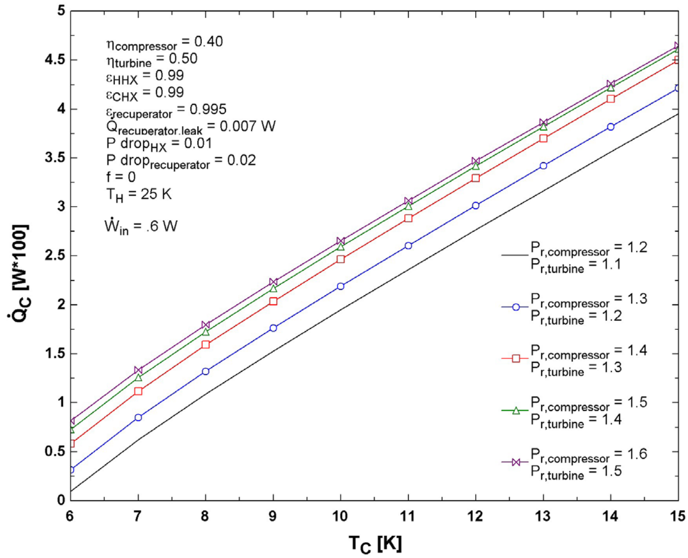


Figure 13. Load curves varying compressor pressure ratio

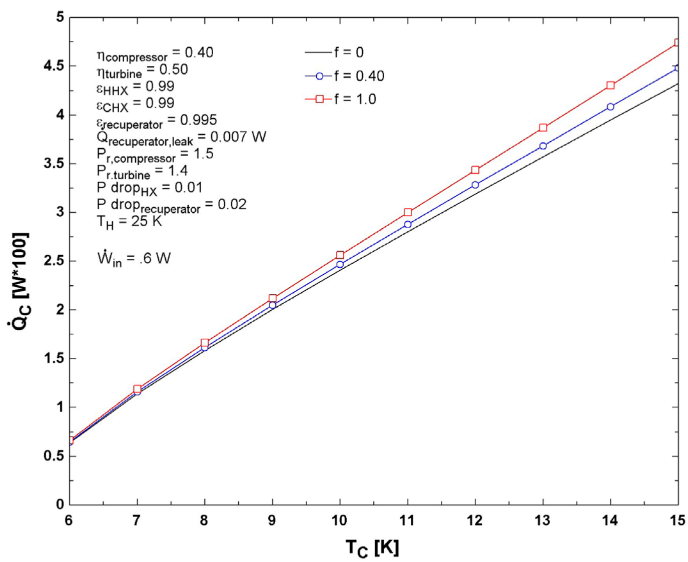


Figure 14. Load curves varying fraction of work recovered from turbine

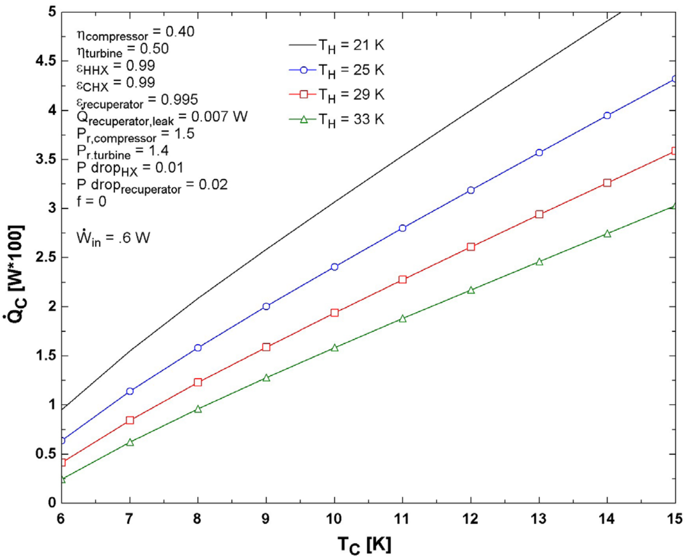


Figure 15. Load curves varying hot side, reject temperature

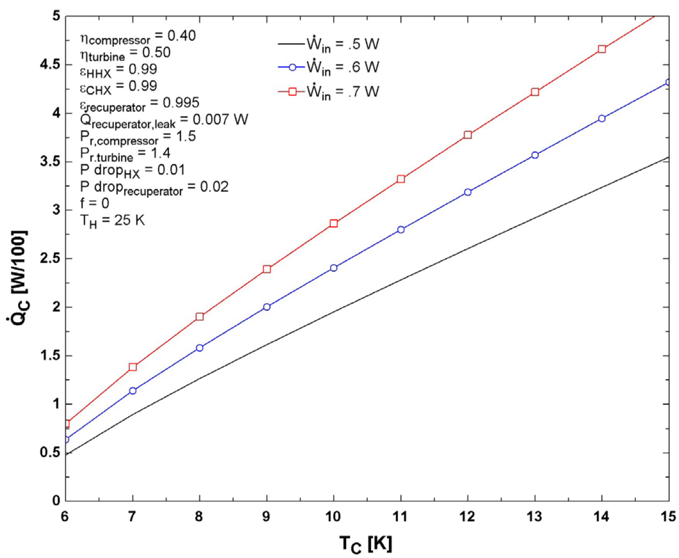


Figure 16. load curves varying work into system

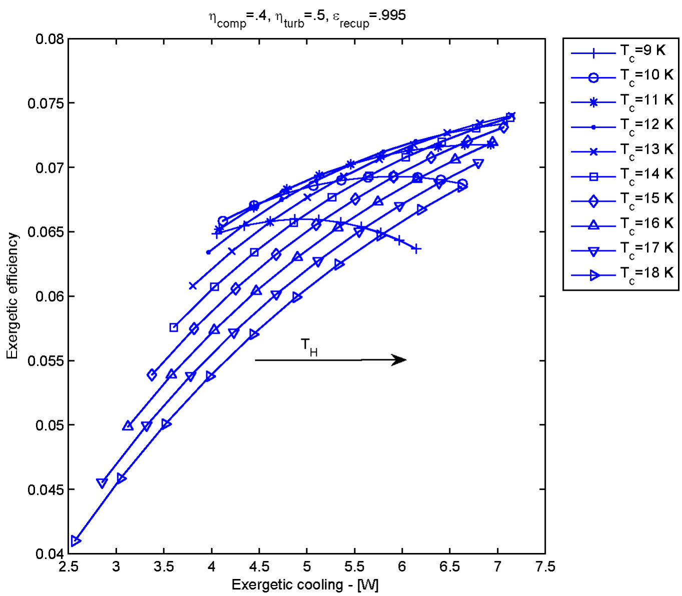


Figure 17. Efficiency at different cold end temperatures, massflow = 1.05 g/s

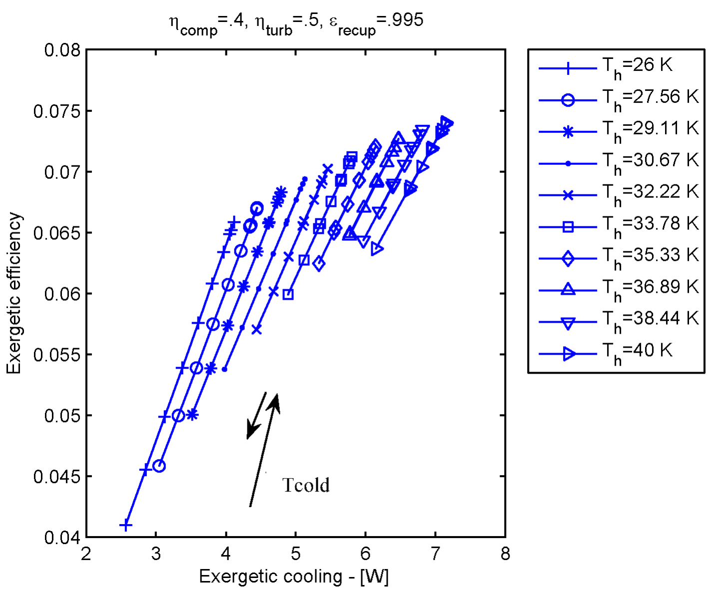


Figure 18. Efficiency at different rejection temperatures, massflow = 1.05 g/s

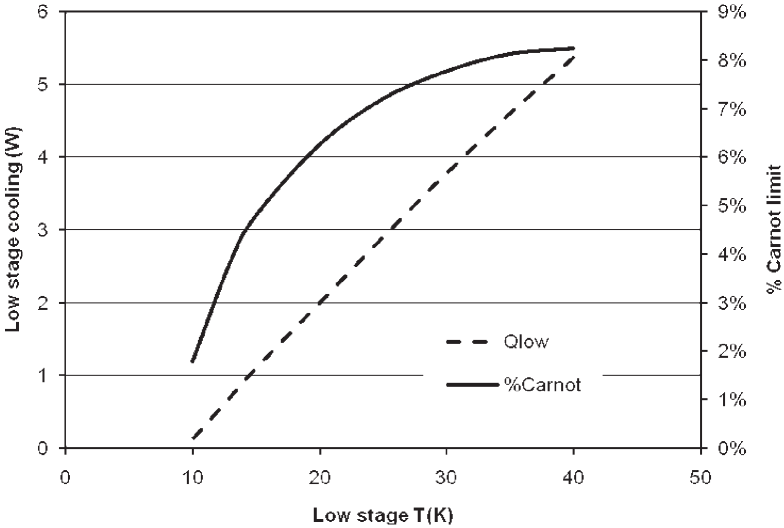


Figure 19. Raytheon RSP2 performance: reject T= 295K, variable stroke, 1.5 W upper stage load.

load curve. Higher pressure ratios in the compressor and turbine also cause a lower load curve. Increasing the pressure drops in the recuperator, HHX, and CHX cause the curve to lower. Recovering work from the turbine to the compressor is helpful at higher temperatures, but is less significant as the difference between reject and cooling temperatures increases, thereby increasing axial conduction. Lower high temperatures have higher load curves. Finally, more work into the system raises the load curve.

UPPER STAGES PERFORMANCE AND HYBRID PERFORMANCE ESTIMATES

Raytheon RSP2 Performance Estimate and Actual Performance Data

Raytheon SAS prepared performance estimates for the RSP2 Stirling-Pulse Tube two stage cryocooler, with 1.5W loading on the Stirling upper stage, shown in Figure 19. Notable in this study is the customary falloff in Stirling efficiency below 30 K.

Raytheon actual performance measurements, consistent with this model, have been published by Kirkconnell et al. [2].

Hybrid Performance Estimates

Combining the results of the lower stage Brayton model, scaled linearly for interface heat flows from the Brayton rejection to Stirling lower stage cooling, with the actual measured RSP2 performance[2] gives the following performance estimates shown in Table 4. The lower Stirling stage provides a rejection interface to the Brayton and supports a 1 W load at the indicated temperatures.

Using the modeled estimated performance yields the performance estimate shown in Figure 20 for varying interface-lower Stirling stage temperatures. The Stirling lower stage is again assumed to

Table 4. Hybrid performance estimate using RSP2 measured performance.

Qc (W) Brayton	T (K) interface	Qc (W) upper stage	T (K) upper stage	Power ttl (W)	%Carnot
0.038	31.9	12.48	89.9	510.3	7.53%
0.053	35.2	9.99	85.3	508.5	6.67%
0.043	30.3	7.54	81.5	510.0	5.90%
0.065	38.9	7.54	80.8	512.0	5.63%

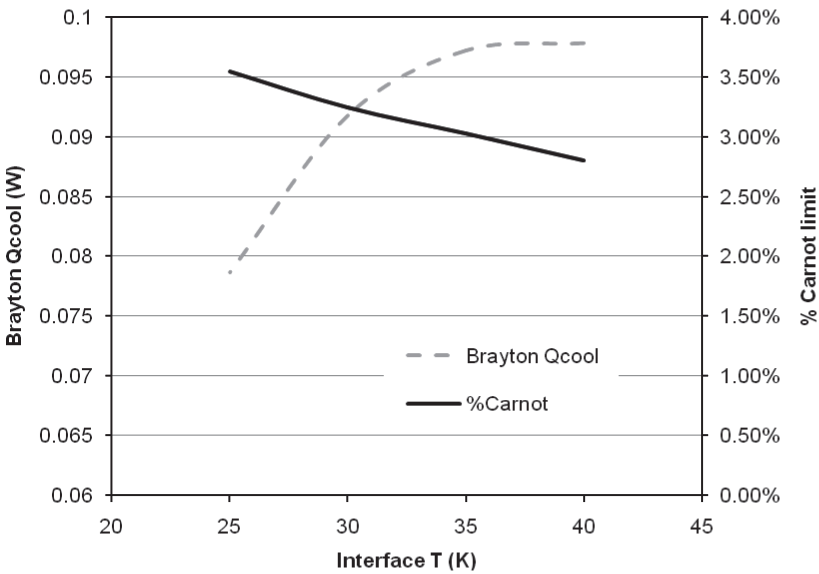


Figure 20. Hybrid performance: reject T= 295K, 1 W interface load, 1.5 W upper stage load.

have a 1 W cooling load available, while the upper stage supports a constant 1.5 W load. The Brayton stage always cools at 10 K and the interface heat transfer is assumed to be nearly isothermal for modeling simplicity.

CONCLUSIONS

The impact of using a recuperative Brayton cycle as a lower stage allows for the normal drop off in Stirling cycle performance below 30 K to be avoided. It is evident from this study that further optimization of the hybrid cycle can be conducted on a modeling basis. The general trend of performance shown indicates that there should be a distinct hybrid system total cooling versus efficiency path [3] from which optimal cooling outputs and efficiencies can be described for a specific operating parameter set. This further description of the hybrid system represents the subject of future modeling efforts.

REFERENCES

1. Kirkconnell, C. et al., "Hybrid Stirling / Reverse Brayton and Multi-Stage Brayton Cryocoolers for Space Applications," *Adv. in Cryogenic Engineering*, Vol. 51B, Amer. Institute of Physics, Melville, NY (2006), pp. 1489-1502.
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3. Razani, A. et al., "Second Law Based Thermodynamic Optimization Criteria for Pulse Tube Refrigerators," *Cryocoolers 14*, ICC Press, Boulder, CO (2007), pp. 285-292.