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Technical Note

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Model Fidelity Requirements for Closed-Brayton-Cycle Space Power Systems

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I. Introduction

CLOSED-BRAYTON-CYCLE (CBC) power conversion is one technology that can be used in space power systems for spacecraft, surface bases, and planetary rovers. More than 270 works on Brayton space power system topics appear in the literature over the last 30 years. Brayton system conceptual designs for milliwatt to megawatt power converters have been developed [1–9]. When optimizing a CBC power conversion system (PCS), numerous factors affect overall system performance; a partial list includes turbomachinery efficiency, heat exchanger effectiveness, working-fluid composition, and cycle temperatures and pressures. The present work also addresses the significance of compressor bleed flows and different internal loss mechanisms. Unfortunately, in much of the literature, CBC modeling fidelity is unclear. If items such as the aforementioned factors are not specifically addressed, a reader cannot confidently rely on the results. Effecting CBC model improvements, the present work has two objectives: to demonstrate system-level impacts of fidelity differences in CBC models and to recommend a minimum CBC modeling fidelity for conceptual design studies.

II. Fidelity Necessity

System models aid in answering engineering design questions. Requisite model fidelity depends on the questions being considered. In the conceptual design and sizing of CBC power systems, steady-state models generate performance, mass, and volume estimates. For any model, there exists a minimum set of component and subsystem models that are needed to adequately characterize the system.

Figure 1 shows a recuperated CBC system. This oversimplified configuration includes basic CBC elements but omits bearings, compressor bleed flow (used to cool bearings and the alternator rotor), heat exchanger details, and external subsystem elements that directly influence performance. Figure 2 shows a more realistic CBC diagram that includes heat rejection subsystem (HRS) information; the HRS is added to understand gas cooler performance and auxiliary loads such as pumping power. For convenience, the heat source subsystem is shown generically because vastly different models are required for solar, chemical, or nuclear sources. Some heat source details are mandatory to complete a thorough power system analysis. However, because we only seek to illustrate CBC modeling issues, the generic source will suffice. To evaluate systems such as the one shown in Fig. 2, we use a pedigreed high-fidelity CBC modeling code: the NASA closed-cycle engine program (CCEP) [8–12]. The CCEP code [and its successor: closed-cycle system simulation (CCSS)] was experimentally validated under steady-state and transient conditions; the most recent validation results are presented by Johnson and Hervol [13]. Table 1 summarizes the fidelity effects for cases A through F. Check marks show the increased fidelity of each case. Although the simplest case may appear to be a reasonable CBC model, over 10 points of cycle efficiency are lost as the model becomes more realistic.

Detailed data from all cases are summarized in Table 2. Each case represents a 100-kWe, two-engine configuration, as shown in Fig. 2. Case A is an oversimplified case that neglects compressor bleed flows, mechanical losses (bearings and windage), and electrical (EM) losses. Optimistic turbomachinery efficiencies are also selected. The resulting 50-kWe engine runs at 31.9% converter efficiency:

$$\eta_{\text{converter}} = \text{alternator electrical output/cycle heat input} \tag{1}$$

In case B, we add 2% compressor bleed flow for bearing and alternator cooling. The bleed flow reduces the turbine inlet temperature and degrades converter efficiency to 30.1%. Case C introduces design performance maps to estimate turbomachinery efficiencies. (Mean-line design codes are even better estimates of compressor and turbine performance and are preferred to generic conceptual design maps.) Even though the original polytropic efficiencies were less than 90% ($e_{\text{compressor}} = 87\%$ and $e_{\text{turbine}} = 89\%$), the design performance maps reduce each isentropic efficiency by three points or more (85 to 82% for the compressor and 90 to 86% for the turbine). The converter efficiency drops to 25.1%. Cases D through F add bearing, windage, and electromagnetic losses that yield progressively more realistic performance estimates. The

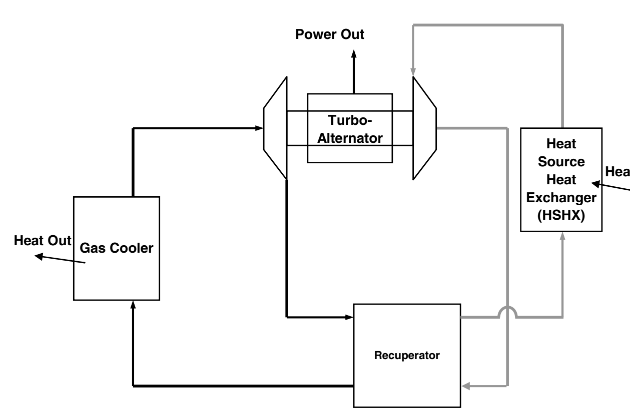

Fig. 1 Simple recuperated Brayton cycle.

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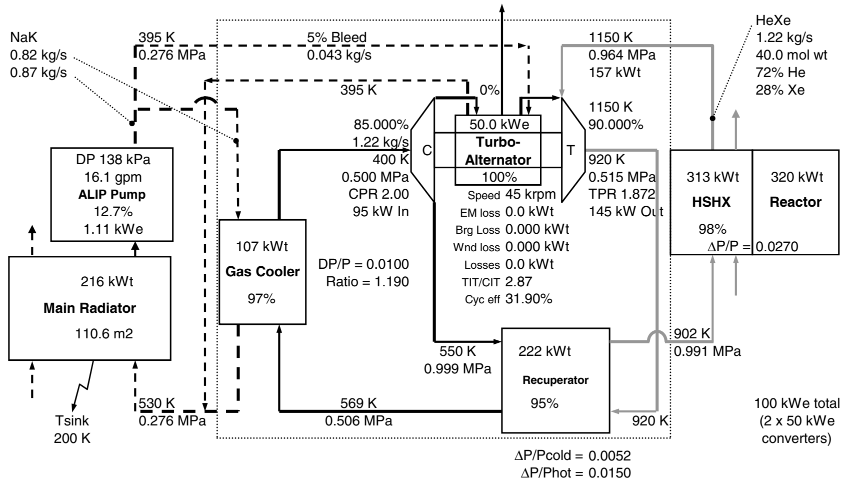

Fig. 2 CBC with optimistic turbomachinery η, 0% compressor bleed, no bearing, windage, or EM losses.

fidelity increase from case A to case F revises the efficiency prediction from 31.9 to 21.7%, a relative reduction of 32%. As converter efficiency falls, heat input and heat rejection increase, cycle state points shift, auxiliary loads change, and system mass estimates increase. From case A to case F, the combined CBC and HRS system mass estimate increases from 1899 to 2308 kg, a jump of nearly 22%. In the conceptual design phase, a 22% underestimation of mass could deplete an entire subsystem mass margin allocation. Additionally, in even the earliest conceptual design of a space power system, 30% error in calculated efficiency is unacceptable. Conducting CBC analyses without realistic turbomachinery efficiencies, mechanical losses, and electromagnetic losses is, at best, troublesome and, at worst, seriously misleading.

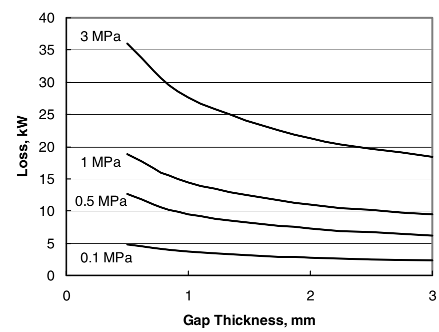

Fig. 3 Windage loss example.

III. Performance Sensitivity

Performance of a CBC converter is strongly affected by individual component characteristics. Turbomachinery efficiencies and heat exchanger effectiveness are commonly known to influence cycle performance. However, sensitivity to parameters such as bearing-film thickness or alternator-gap size is less recognized.

Film thickness and gap size affect frictional losses associated with shaft rotation. These viscous losses rob the converter of useful shaft work. For a converter that runs at 25% efficiency, each kilowatt of viscous loss requires up to 1 kW of heat rejection and 4 kWt of additional supply heat. (A fraction of viscous heating can be recovered as useful work if the heated fluid travels through the recuperator or turbine.) Selecting a design-point bearing-film thickness requires an extremely complicated trade study involving items such as bearing load capacity, bearing and shaft geometry, off-design expectations, thermal management, and rotordynamics [14]. Likewise, trades in electromagnetic efficiency, rotordynamics, material properties, and operating stresses are needed to set the operating "air" gap in the alternator. Because the working fluid is sheared in the bearing-film and alternator-gap cavities, mechanical losses are significantly impacted by separation distances and operating pressures. For thrust and journal bearings, effective film thickness and operating pressure cannot be treated as independent parameters; bearing load capacity couples the two parameters and requires load estimates to optimize the thickness setting. However, for windage losses, we can discern independent effects. To illustrate, Fig. 3 presents windage loss estimates as gap thickness and peak operating pressure vary on a 100-kWe CBC unit. The data indicate that loss estimates can be significant percentages of total power output, particularly at higher pressures. Loss models in CCEP [12] allow us to evaluate performance sensitivities to loss estimates. Comparing cases C through E uncovers cycle impacts from mechanical losses. At 1-MPa peak pressure, bearing and windage losses reduce the converter efficiency from 25.1 to 22.4%. Cases F and G specifically highlight mechanical loss sensitivity to pressure. For a peak operating pressure of 1 MPa, case F shows total mechanical losses of approximately 9 kW. At a 3-MPa peak, case G totals 23 kW of mechanical losses, a 156% increase. Unfortunately, significant variation appears in different loss models. Case H (3-MPa peak) gives converged cycle conditions similar to that of case G but using a different empirical loss estimator [9]. Mechanical losses shown in case H are less than 14 kW, still 56% above those in case F but 40% less than case G. Mechanical loss values in cases F, G, and H contribute to the respective 21.7, 14.7, and 16.1% converter efficiencies noted. In addition to converter efficiency, other major design factors are also influenced by mechanical losses. For example, startup auxiliary power requirements are heavily dependent on mechanical loss and total motoring time estimates [15].

Table 1 Cycle efficiency decrease as model fidelity increases

Model element	Case A	Case B	Case C	Case D	Case E	Case F
Compressor bleed flow		✓	✓	✓	✓	✓
Map-based turbomachinery efficiencies			✓	✓	✓	✓
Bearing losses				✓	✓	✓
Windage losses					✓	✓
Electromagnetic losses						✓
Cycle efficiency, %	31.9	30.1	25.1	23.8	22.4	21.7

Table 2 Data summary

Parameter	Case A	Case B	Case C	Case D	Case E	Case F	Case G	Case H
Compressor								
T in, K	400	400	400	400	400	400	400	400
P in, MPa	0.500	0.500	0.500	0.500	0.500	0.500	1.50	1.50
T out, K	550	550	556	556	555	555	559	560
P out, MPa	1.00	1.00	1.00	1.00	1.00	1.00	3.00	3.00
Mass flow, kg/s	1.22	1.27	1.58	1.67	1.79	1.85	2.83	2.59
Bleed flow	0.0%	2.0%	2.0%	2.0%	2.0%	2.0%	2.0%	2.0%
Power, kW	95.0	99.6	128	135	144	149	234	215
Efficiency	85.0%	85.0%	82.1%	82.2%	82.3%	82.3%	80.3%	79.9%
Turbine								
T in, K	1150	1143	1143	1144	1145	1145	1147	1146
P in, MPa	0.964	0.963	0.962	0.962	0.962	0.962	2.89	2.89
T out, K	920	915	925	925	924	924	934	934
P out, MPa	0.516	0.515	0.514	0.514	0.514	0.514	1.545	1.545
Mass flow, kg/s	1.22	1.27	1.58	1.67	1.79	1.85	2.83	2.59
Power, kW	145	150	178	189	203	211	310	282
Efficiency	90.0%	90.0%	86.3%	86.7%	87.1%	87.3%	83.8%	83.4%
Alternator								
Power, kW	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0
Speed, rpm	45,000	45,000	45,000	45,000	45,000	45,000	45,000	45,000
Alt efficiency	100%	100%	100%	100%	100%	95.0%	93.8%	93.8%
EM loss, kW	0.0	0.0	0.0	0.0	0.0	2.63	3.33	3.33
Brg loss, kW	0.0	0.0	0.0	3.92	4.13	4.22	12.99	5.28
Wind loss, kW	0.0	0.0	0.0	0.0	4.78	4.70	9.82	8.37
Losses tot., kW	0.0	0.0	0.0	3.92	8.91	11.6	26.1	17.0
Recuperator								
T in hot, K	920	912	922	922	923	923	933	933
T in cold, K	550	550	556	556	555	555	559	560
T out hot, K	569	575	581	581	581	581	585	586
T out cold, K	902	894	903	904	904	905	915	914
DP/Phot	0.015	0.015	0.015	0.015	0.015	0.015	0.015	0.015
DP/Pcold	0.0516	0.00500	0.00547	0.00563	0.00585	0.00597	0.00486	0.00480
Heat load, kW	222	223	280	297	318	329	513	466
Effectiveness	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95
Heat Source								
T in, K	902	894	903	904	904	905	915	914
P in, MPa	0.991	0.992	0.991	0.991	0.991	0.991	2.97	2.97
T out, K	1150	1150	1150	1150	1150	1150	1150	1150
P out, MPa	0.964	0.963	0.962	0.962	0.962	0.962	2.89	2.89
DP/P	0.027	0.027	0.027	0.027	0.027	0.027	0.027	0.027
Mass flow, kg/s	1.22	1.25	1.55	1.64	1.75	1.81	2.77	2.53
Heat load, kW	157	166	199	210	224	231	339	311
React. power, kW	320	339	406	428	456	471	692	634
Gas Cooler								
T in gas, K	569	575	581	581	581	581	585	586
P in gas, MPa	0.506	0.506	0.506	0.506	0.506	0.506	1.52	1.52
DP/P gas	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010
Gas flow, kg/s	1.22	1.27	1.58	1.67	1.79	1.85	2.83	2.59
NaK flow, kg/s	0.823	0.975	1.20	1.25	1.34	1.38	1.97	1.81
Heat load, kW	107	116	149	157	168	173	272	250
Effectiveness	0.97	0.97	0.97	0.97	0.97	0.97	0.97	0.97
Radiator								
Rad. T in, K	530	518	524	528	529	531	547	544
Rad. P in, MPa	0.276	0.276	0.276	0.276	0.276	0.276	0.275	0.275
Pump T out, K	395	395	394	394	394	394	394	394
Pump P out, MPa	0.276	0.276	0.276	0.276	0.276	0.276	0.276	0.276
Mass flow, kg/s	0.867	1.03	1.26	1.31	1.41	1.45	2.08	1.90
Pump DP, MPa	0.138	0.138	0.138	0.138	0.138	0.138	0.139	0.139
Pump power, kW	1.11	1.18	1.28	1.31	1.38	1.39	1.55	1.52
Pump efficiency	12.7%	14.1%	15.9%	16.2%	16.6%	16.9%	21.8%	20.4%
Bleed flow	5.0%	5.0%	5.0%	5.0%	5.0%	5.0%	5.0%	5.0%
Alternator T out, K	395	395	394	436	483	507	574	521
T sink, K	200	200	200	200	200	200	200	200
Total area, m^2	111	125	157	167	180	186	281	256
Heat load, kW	216	235	301	323	350	365	582	525
Cycle								
Efficiency	31.9%	30.1%	25.1%	23.8%	22.4%	21.7%	14.7%	16.1%

IV. Conclusions

Requisite modeling capabilities are dictated by the specific design question at hand. However, as a general recommendation, conceptual designs of closed-Brayton-cycle space power conversion systems must include realistic representations of turbomachinery efficiencies, mechanical losses, and electromechanical losses. Cycle efficiency errors of 30% and mass estimate errors of 20% are possible using even moderately unrealistic representations.

Mechanical losses in bearings and alternators can be significant factors in closed-Brayton-cycle energy balances. A two-engine, 100-kWe case with 3-MPa peak pressure had mechanical loss predictions as high as 23% of the total power output.

Existing loss models yield significantly different loss predictions. Using two available models, a 40% difference in mechanical losses was predicted for a 100-kWe closed-Brayton-cycle system operating at 3-MPa peak pressure. More research is needed to reduce the uncertainty in journal and thrust bearing loss predictions over a range of relevant operating conditions.

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