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Examining Staged Enhancements for Thermodynamic Cycles to Improve Performance using an Intelligent Instruction Software*

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Thermodynamic design and analyses have been carried out on the sequential performance improvement of an air-standard Brayton cycle through staged enhancements from a design perspective employing a relatively new thermodynamics instruction software. A synergistic combination of qualitative physics and artificial intelligence techniques have been used to develop CyclePad to assist in teaching, design and research in applied thermodynamics and advanced energy conversion systems. It provides an articulate virtual laboratory, in terms of visualisation of the schematic combination of a variety of thermodynamic cycles. In this study, several enhancements have been investigated that can be made to the simple air-standard Brayton cycle to obtain improved performance. The addition of a regenerative heat exchanger, followed by multiple stages of reheating and intercooling have been considered. Sensitivity analyses has been carried out to help optimise the design. Such studies demonstrate that analyses of complex cycles can now be carried out, as part of classroom instruction of thermodynamics as well, simultaneously demonstrating the effects of staged enhancement of design on the global performance of the cycle.

INTRODUCTION

THERMODYNAMIC cycle analysis continues to arouse interest among both researchers and educators [1]. Lately computer-aided learning has become popular and a few tools based on software platforms have become available. In addition to softwares for thermodynamic property computation for various substances, of late the trend has been to develop softwares which have some inherent intelligence and which can guide the user in the design of a system without limiting itself to a mere analysis of it. These tools are opening up new vistas of opportunity for the educators since such design-oriented exercises can now be undertaken as part of classroom instruction in the form of student assignments or design projects. Remote Internet-based education is being offered to a greater extent today and such tools could be of immense utility under these configurations. The software platform is expected to augment the present curricula as a new paradigm rather than as a replacement module.

CYCLEPAD: AN INTELLIGENT ENVIRONMENT

A synergistic combination of *qualitative physics* and *artificial intelligence* techniques have been employed to develop CyclePad [2, 3] to assist in teaching, design and research in applied thermodynamics and advanced energy conversion. This quite versatile freeware available over the Internet [4] introduces students to the concept of design as an open-ended process involving synthesis, analysis, and choices among design alternatives. It provides an articulate virtual laboratory [3], in terms of a software environment consisting of a set of parts, corresponding to physical components or important abstractions in the domains of interest, tools for assembling collections of these parts into designs, and facilities for analysing and testing designs. It enables the visualisation of schematic combination of a variety of thermodynamic cycles which enables the students to explore the sensitivity of key parameters on global behaviour of the cycle. Aside from their intrinsic interest, the conceptual design of thermodynamic cycles provides an extremely motivating context for students to learn fundamental principles more deeply than they would otherwise. This is strongly evident in several student and faculty surveys that have been undertaken in several universities [2] from which it transpires that such a tool is welcomed by the students. It does not replace the requirement of solving problems by hand but certainly it supplements that activity quite effectively. For complex configurations, the students are not bogged down by routine calculations and in the mechanics of problem solving. CyclePad automates the numerical analysis of cycles so

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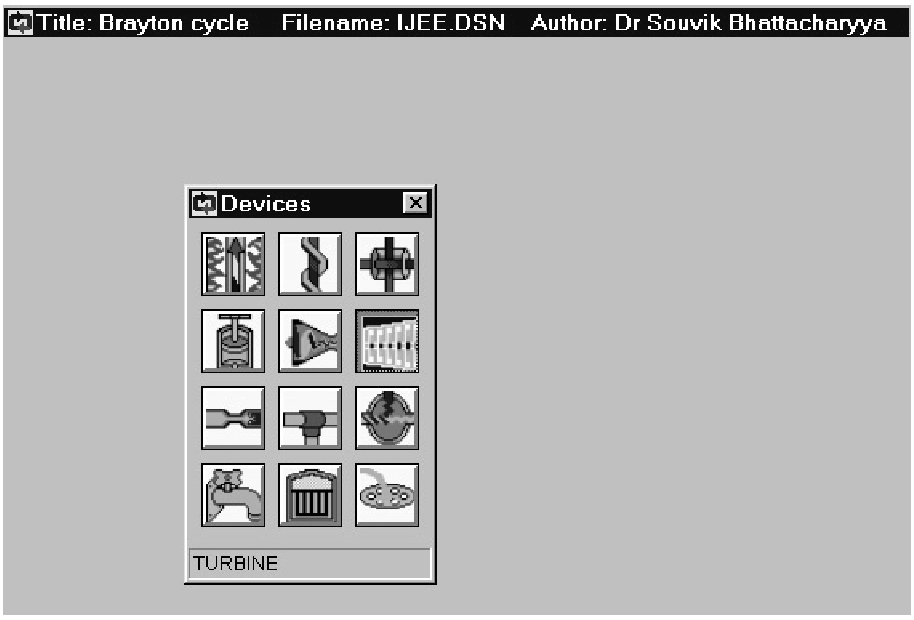

Fig. 1. Component palette of CyclePad.

that one can devote more time and effort in thinking about the implications of a specific design or to explore *what if* options available to the designer. To assist in the investigations, CyclePad provides a hypertext-based query system, offering explanations of any parametric value, substance phase, or modelling assumption, that CyclePad has derived.

CyclePad allows one to specify the structure of the design, through sequential linking of components. The design is then analysed by assuming numerical values for parameters (e.g. operating temperatures and pressures), and making modelling assumptions (e.g. whether or not to consider a device as isentropic) and choosing the working substances. Sensitivity analyses can be performed to understand how different parameters of a design contribute to its performance. For example, CyclePad can determine how the efficiency of a system changes as a function of other parameters, such as condenser pressure in a vapour power cycle. CyclePad performs steady-state analyses of both open and closed cycles while working in two phases, *build mode* and *analyse mode*. In the first phase (build), a graphical editor is employed to place components and to connect them with *stuffs* (component links) from the *Component Palette* (Fig. 1).

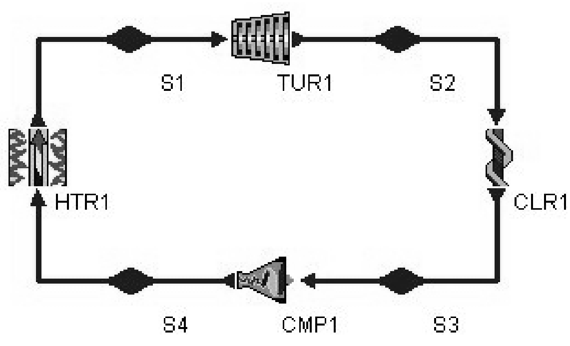

Fig. 2. Basic Brayton cycle as the reference system.

One can only proceed to the next phase (analysis) when every component is connected to another component via links or stuffs, and every stuff has been used as both an input and an output for components in the design. Figure 2 shows such a network of components linked in sequence for a simple Brayton cycle. When there has been an erroneous data furnished or an impossible assumption made, the *contradiction* dialogue box provides the assistance by identifying the item at fault and suggesting possible changes.

In the analysis phase, items specified are: the working fluid, modelling assumptions used to analyse the design, and numerical values for the properties of components and stuffs. As soon as CyclePad receives some information from the user, it draws as many conclusions as it can about the design, based on everything we have told it so far. When we specify a working fluid, for instance, it knows whether to use property tables or an ideal gas approximation. When we specify numerical values, CyclePad sees if it can then calculate other numerical values. It displays the results of its calculations, and we are free to inquire about how values were derived and how one might proceed at any time, using a hypertext query system. As we provide more information, CyclePad deduces more about the physical system. At this juncture, one may want to investigate the relationship between a value that has been assumed and one that CyclePad has calculated in order to determine how to improve the design. The *sensitivity analysis* tool, from the Tools pull-down menu, is employed to do this using a dependent parameter and an independent parameter; this entire process is extremely *interactive* and *user* *defined*. There are many *what if* questions that can be asked for systems analysis in a preliminary design.

REFERENCE SYSTEM

A *simple* air-standard Brayton cycle with the following configuration is chosen as the reference system:

- turbine inlet pressure and temperature of 1 Mpa and 1600 K, respectively
- turbine exhaust pressure of 100 kPa
- compressor inlet temperature of 300 K
- mass flow rate: the entire exercise is performed for a unit flow rate (1 kg/s).

We start off with the basic simple cycle and then proceed to enhance it with modifications, recording the performance improvement over the basic configuration for each enhancement and combinations thereof. This way the students will have a grasp of each of these enhancements and the way they affect the global behaviour of the cycle, relative to the reference system.

CyclePad analysis yields the following statistics for the simple cycle:

- maximum cycle temperature 1600 K
- mean temperature of heat addition 1004 K
- minimum cycle temperature 300 K
- mean temperature of heat rejection 520 K
- Carnot efficiency 81.25%
- thermal efficiency 48.21%
- heat addition 1024 kW
- heat rejection 530.4 kW
- net power output 493.7 kW
- backwork ratio 36.2%.

EFFICIENCY IMPROVEMENT FOR AIR STANDARD CYCLES

Efficiency improvement of thermodynamic cycles almost always involves making it approach a Carnot cycle for the same temperature limits. The heat addition occurs isothermally at the highest temperature and the heat rejection occurs at the lowest temperature in case of the Carnot cycle. We would examine these two characteristics for our chosen basic cycle, the T-s diagram of which is shown in Fig. 3. We notice two things about this cycle: i) during the cooling process, we are throwing away significant amount of heat and ii) not the entire heat addition and rejection is at (or even near) the maximum and minimum cycle temperatures.

When we seek improvement in the efficiency of a cycle, we often consider the mean temperature of heat addition $\overline{T}_{in}$, and the mean temperature of heat rejection, $\overline{T}_{out}$. These represent what the temperature would have been if the same amount of heat had been added or rejected isothermally. They allow us to treat improving cycle efficiencies as we would for a Carnot cycle: by raising $\overline{T}_{in}$ or by lowering $\overline{T}_{out}$.

For reversible heat transfer, the average temperature of heat addition is:

$$\overline{T}_{in} = Q_{\text{in}}/\Delta S$$

and the average temperature of heat rejection is:

$$\overline{T}_{out} = Q_{\text{out}}/\Delta S.$$

Hence, the thermal efficiency is obtained as:

$$\eta_{\text{th}} = (\overline{T}_{in} - \overline{T}_{out})/\overline{T}_{in}$$

which yields the same value as we get from $\eta = \text{W}_{\text{net}}/Q_{\text{in}}$.

To increase η, we would like to add heat at a higher temperature and reject it at a lower temperature.

For the simple Brayton cycle, the average temperature of heat addition is 596 K lower than the maximum cycle temperature. Similarly, the average temperature of heat rejection is 220 K above the minimum cycle temperature. We also

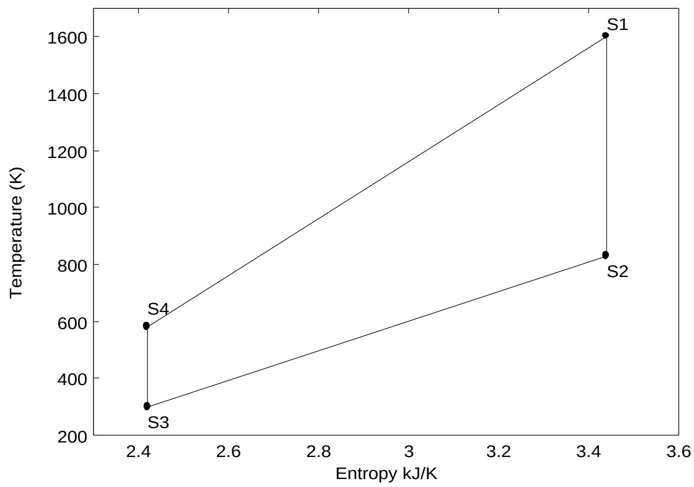

Fig. 3. T-s diagram for the simple Brayton cycle.

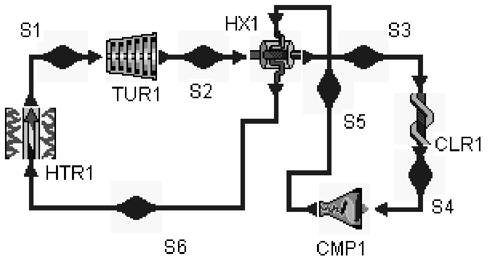

Fig. 4. Regenerative Brayton cycle with a heat exchanger.

see that the amount of heat rejected is too high for the cycle to be efficient and the compressor power input is a significant 36.2% of the turbine power. Let us investigate some enhancements to the basic cycle and examine their effects on global cycle performance.

STEP I: Regenerative heat exchanger

Let us examine the temperatures at the four state points for the simple Brayton cycle as shown in Fig. 3:

$$T_1 = 1600\,K,\ T_2 = 829\,K,$$
$$T_3 = 300\,K,\quad T_4 = 579\,K$$

The heat rejected by the fluid stream during process 2–3 is a potential source for some heat recovery during process 4–1 simply because of the fact that $T_4\,(=829\,K)$ is significantly higher than $T_4\,(=579\,K)$ and there is ample room for useful heat recovery. Consequently, to exploit this potential, we incorporate an ideal, counter-current heat exchanger (Fig. 4).

The ideal heat exchange process is isobaric, and the cold fluid can be heated up all the way to the inlet temperature of the hot fluid. So $T_6 = T_2$ and $T_3 = T_5$. This means that the effectiveness of the heat exchanger is the maximum possible $(=1)$ and anything less, as in case of a real heat exchanger, would produce $T_6 < T_2$ and $T_3 > T_5$.

Table 1. Regenerative cycle statistics relative to simpel Brayton

	Regenerative cycle	*Simple cycle*
Maximum cycle temperature	1600 K	1600 K
Mean temperature of heat addition	1172.6 K	1004 K
Minimum cycle temperature	300 K	300 K
Mean temperature of heat rejection	424.5 K	520 K
Carnot efficiency	81.25%	81.25%
Thermal efficiency	63.80%	48.21%
Heat input	774 kW	1024 kW
Net power output	493.7 kW	493.7 kW
Heat rejected	280.3 kW	530.4 kW

Let us focus on a few crucial and favourable developments in the modified cycle which are evident from the T-s diagram (Fig. 5). A significant amount of heat is being recovered ($Q_{regen} = 250.1\,kW$) from the exhaust stream between S2 and S3 and thereby the compressor outlet state point S5 is now elevated to S6. Consequently heat is now added only between S6 and S1 (as opposed to S5 and S1, in the simple cycle) and hence the average temperature of heat addition has gone up significantly (Table 1). Similarly, heat is now rejected between S3 and S4 (and not between S2 and S4 as in the case of the simple cycle) and hence the average temperature of heat rejection has gone down significantly (Table 1). These two phenomena together have contributed to improvement in cycle efficiency which is now a good 63.8%—much closer to the Carnot limit. A significant drop in heat rejected and as well as heat added is also noted for the regenerative cycle relative to the simple cycle (Table 1).

STEP II: Multistage turbine with reheat

A reheat stage with split turbine expansion can be added to the cycle to bring the average temperature of heat addition closer to the maximum cycle

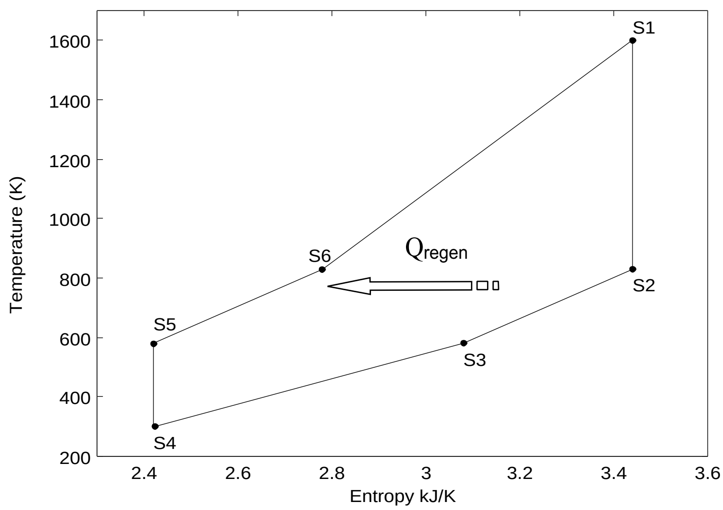

Fig. 5. Brayton cycle with heat exchanger T-s diagram.

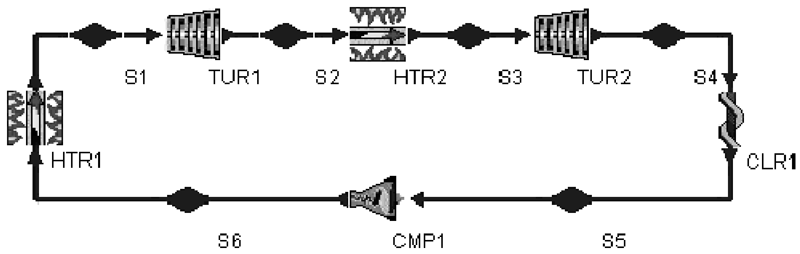

Fig 6. Brayton cycle with a reheat stage between turbines.

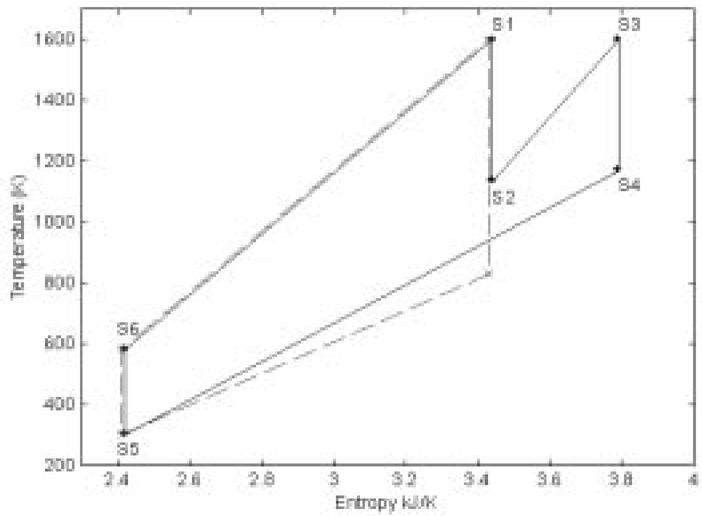

Fig. 7. T-s diagram for Brayton cycle with reheat stage between turbines.

temperature (Fig. 6). The simple cycle is overlaid in broken lines for comparison. The intermediate pressure for the reheat stage is chosen as 300 kPa, giving each turbine approximately equal pressure ratios. We note that the average temperature of heat addition has increased (Table 2 and Fig. 7).

However, the efficiency of the reheat cycle is lower by almost 7%. We note that both the mean temperature of heat addition and heat rejection have increased (Fig. 7 and Table 2). So, while the former works favourably, the latter affects the η adversely. In this case, the net effect is a significant drop in cycle efficiency. In other words, although the heat rejected amount has dropped relative to the simple cycle, the increase in heat input outweighs that to cause a resultant drop in η however, with a rise in net work output.

Table 2. Brayton cycle with reheat statistics.

	Reheat cycle	*Simple cycle*
Maximum cycle temperature	1600 K	1600 K
Mean temperature of heat addition	1089 K	1004 K
Minimum cycle temperature	300 K	300 K
Mean temperature of heat rejection	636 K	520 K
Carnot efficiency	81.25%	81.25%
Thermal efficiency	41.54%	48.21%
Heat added	1491 kW	1024 kW
Heat rejected	871.9 kW	530.4 kW
Net power output	619.6 kW	493.7 kW

STEP III: Multistage compressor with intercooling

Now we revisit the technique of adding an intercooling process between compressor stages to lower the compressor work input for the same pressure ratio and to improve the cycle η by lowering the average temperature of heat rejection for the cycle (Fig. 8). Analogous to the reheat stage, the intermediate pressure for the intercooler is chosen to be 300 kPa, yielding approximately equal pressure ratios for each compressor. From the T-s diagram (Fig. 9), we note that the average temperature of heat rejection has come down to 482 K from 520 K, in case of the simple cycle which is overlaid in broken line.

However, similar to the reheat cycle, the efficiency is once again affected adversely. Although we have met our objective of lowering the mean temperature of heat rejection, it is also clear that the mean temperature of heat addition has reduced as well, causing a net drop in efficiency. Moreover, we note (Table 3) that though the heat input has dropped moderately, the heat rejected has risen since now additional heat rejection occurs in the intercooler. As expected, the compressor work has dropped causing the back-work ratio to improve (lower) by almost 6% over the simple cycle.

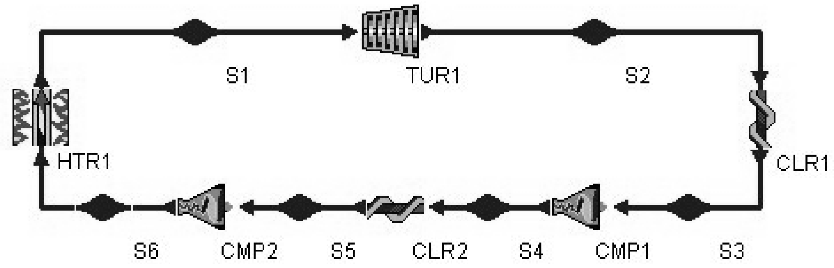

Fig. 8. Brayton cycle with intercooling between staged compression.

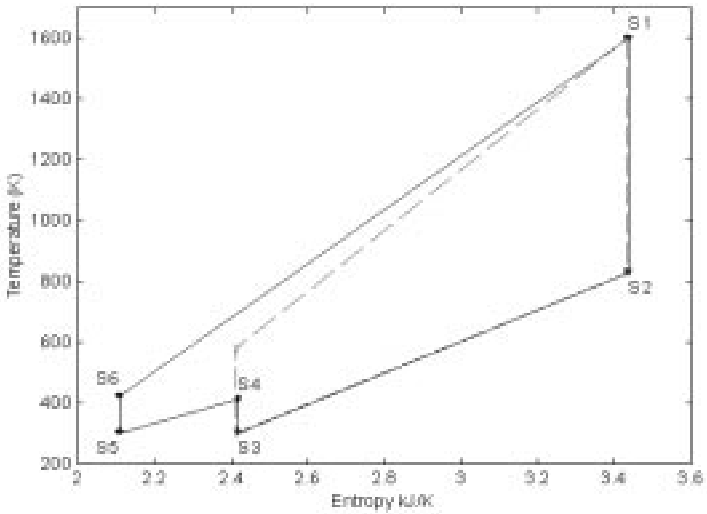

Fig. 9. T-s diagram for Brayton cycle with intercooled compressors.

Table 3. Brayton cycle with intercooling efficiency parameters.

	Intercooled cycle	*Simple cycle*
Maximum cycle temperature	1600 K	1600 K
Mean temperature of heat addition	888 K	1004 K
Minimum cycle temperature	300 K	300 K
Mean temperature of heat rejection	482 K	520 K
Carnot efficiency	81.25%	81.25%
Thermal efficiency	45.67%	48.21%
Heat addition	1181 kW	1024 kW
Heat pejection	641.5 kW	530.4 kW
Net power output	539.3 kW	493.7 kW
Backwork ratio	30.3%	36.2%

STEP IV: Regenerative cycle with intercooling and reheat

In the last three steps, we considered the effect of enhancements on the efficiency of the Brayton cycle. We added a regenerative heat exchanger, a turbine reheat stage, and a compressor intercooling stage. However, we saw that only the regenerative heat exchanger had a positive effect on cycle efficiency, while the other two caused net work output to increase at the cost of efficiency.

Looking back at Step I, we recall that Q_{regen} was limited by the difference (ΔT) between T_2 and T_4. For higher available ΔTs, Q_{regen} would rise resulting in further increase in average temperature of heat addition and hence η. So the objective is to increase the ΔT as far as practicable. When we look at the T-s diagrams in Fig. 7 and Fig. 9, we see that the difference in temperature between the air leaving the compressor and the air leaving the turbine in either cycle is larger than it was for the simple Brayton cycle in both cases. So a combination of Step I and II is in order. Figure 10 shows a cycle with both intercooling and reheating. The T-s diagram is represented in Fig. 11 with the simple cycle shown in broken lines. It is interesting to compare the temperature difference for the simple cycle ΔT $(= T_2 - T_4)$ and the temperature difference of the cycle with both intercooling and reheat stages ΔT $(= T_4 - T_8)$. We note that the latter is about three times greater.

We note that the thermal efficiency of this cycle is still worse than that for the simple cycle. However, we can exploit the larger temperature difference with a heat exchanger. In fact, this cycle was chosen to demonstrate the large potential available for regeneration. The new regenerative cycle is shown in Fig. 12.

The T-s diagram and cycle parameters for this new cycle are shown in Fig. 13 and Table 5. We note that the efficiency is now over 25% greater than that of the basic cycle and within 8% of the Carnot limit. We notice improvements on all fronts. The backwork ratio is trimmed down by over 10% compared to the simple cycle we started with. The net work has gone up correspondingly to a handsome 665.5 kW—a 35% increase over the simple cycle output. The heat rejected is down, as

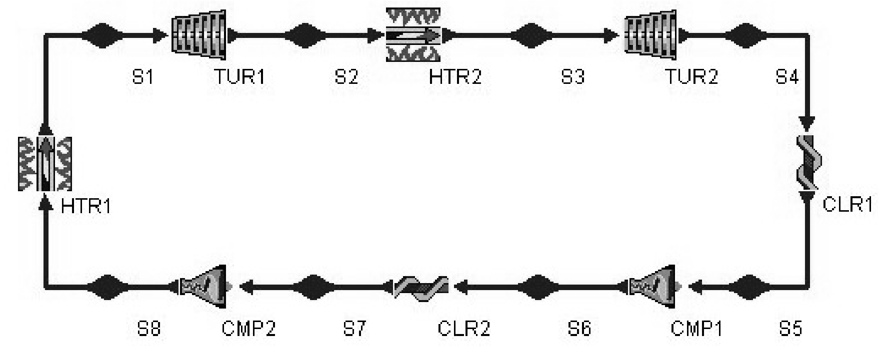

Fig. 10. Brayton cycle with reheat and intercooling.

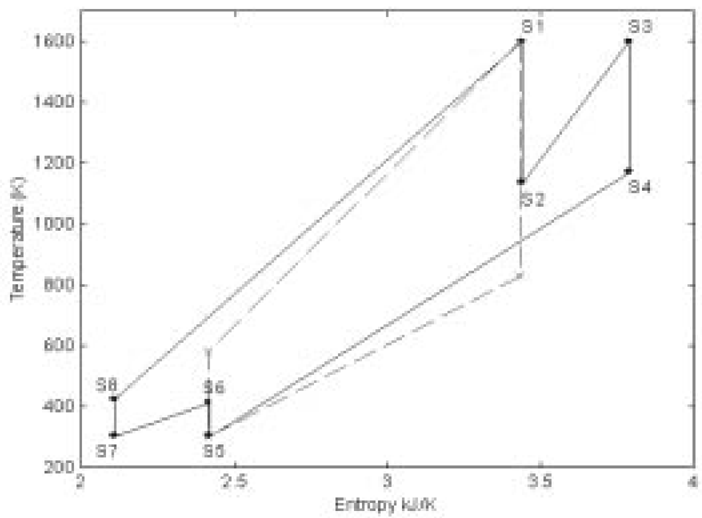

Fig. 11. Brayton cycle with reheat and intercooling T-s diagram.

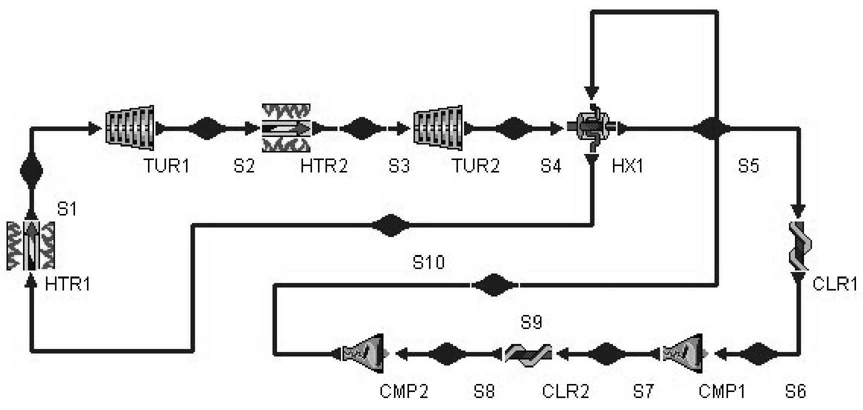

Fig. 12. Brayton cycle with reheat, intercooling and regenerative heat exchange.

Table 4. Brayton cycle with reheat and intercooling.

	Modified cycle	*Simple cycle*
Maximum cycle temperature	1600 K	1600 K
Mean temperature of heat addition	981 K	1004 K
Minimum cycle temperature	300 K	300 K
Mean temperature of heat rejection	585 K	520 K
Carnot efficiency	81.25%	81.25%
Thermal efficiency	40.36%	48.21%

is the heat addition to the cycle. We have achieved a phenomenal increase in mean temperature of heat addition and a substantial drop in mean temperature of heat rejection.

STEP V—Regenerative cycle with 4-stage intercooling and reheat

In Step IV, we examined the addition of regeneration, single-stage reheating, and intercooling to the simple cycle yielding a substantial

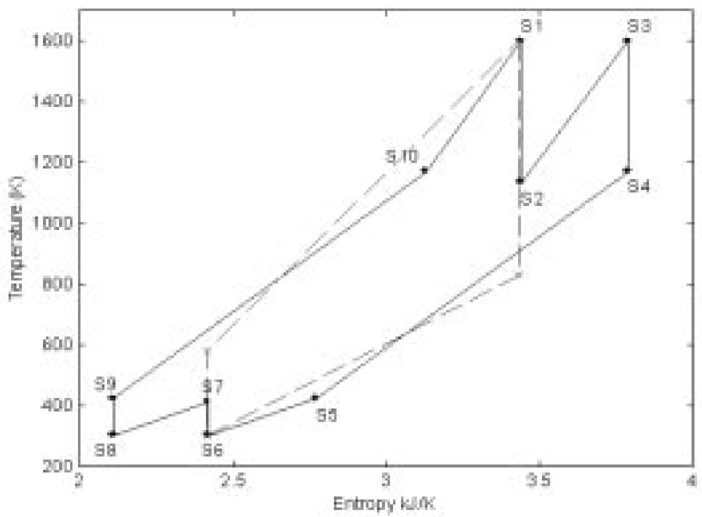

Fig. 13. T-s diagram for Brayton cycle with regeneration, reheat and intercooling.

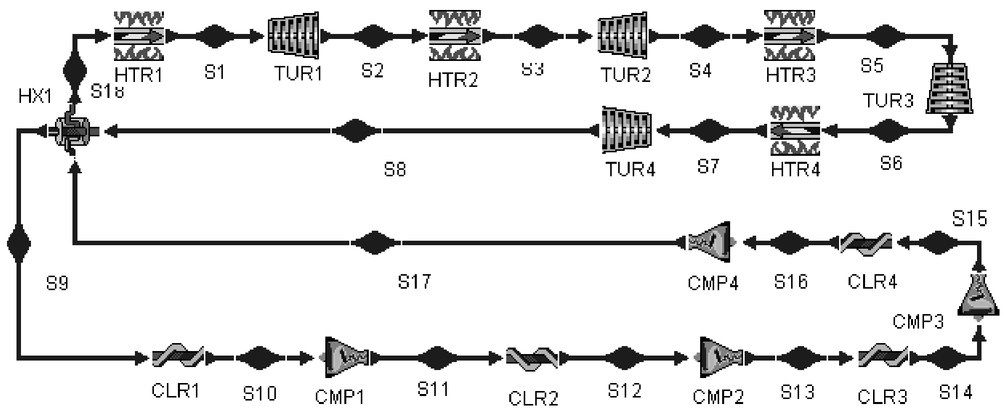

Fig. 14. Brayton cycle with regeneration and 4-stage reheat and intercooling.

Table 5. Brayton cycle with regeneration, reheat, and intercooling.

	Modified cycle	*Simple Brayton*
Maximum cycle temperature	1600 K	1600 K
Mean temperature of heat addition	1364 K	1004 K
Minimum cycle temperature	300 K	300 K
Mean temperature of heat rejection	355 K	520 K
Carnot efficiency	81.25%	81.25%
Thermal efficiency	73.93%	48.21%
Heat addition	900.2 kW	1024 kW
Heat rejection	234.6 kW	530.4 kW
Net power output	665.5 kW	493.7 kW
Backwork ratio	26.1%	36.2%

Table 6. Brayton cycle with regeneration and 4-stage reheat, and intercooling.

	4-stage cycle	*1-stage cycle*
Maximum cycle temperature	1600 K	1600 K
Mean temperature of heat addition	1475 K	1364 K
Minimum cycle temperature	300 K	300 K
Mean temperature of heat rejection	326 K	355 K
Carnot efficiency	81.25%	81.25%
Thermal efficiency	77.89%	73.93%
Heat addition	974 kW	900.2 kW
Heat rejection	215.4 kW	234.6 kW
Net power output	758.6 kW	665.5 kW
Backwork ratio	22.1%	26.1%

improvement in cycle efficiency. At this point, we are curious to find out how the number of stages in a multistage cycle influence the performance relative to the single-stage cycle (Fig. 12). Figure 14 illustrates the schematic representation of such a cycle with four turbines and four compressors, i.e. with four stages each of reheating and intercooling. The individual stage pressures are chosen to yield approximately the same pressure ratio ($\sim$1.78) for each turbine and compressor.

Additionally, we note from the T-s diagram (Fig. 15) that as the number of compression and expansion stages is increased, the ideal cycle will approach the Ericsson cycle, with the entire heat addition occurring near the maximum cycle temperature and the entire heat rejection occurring near the minimum cycle temperature. This is reflected in the cycle performance (Table 6) where we note that the multi-stage cycle is 4% more efficient than the single-stage cycle and is fast

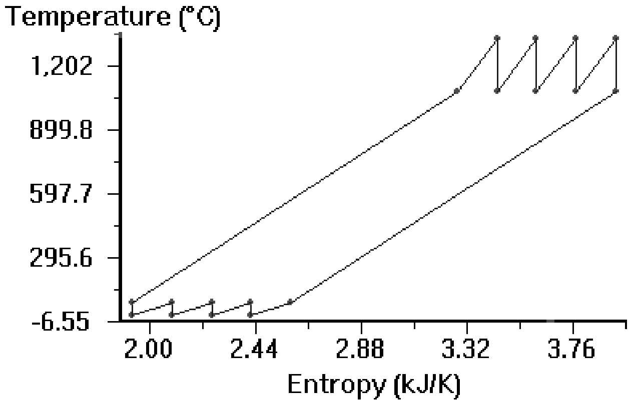

Fig. 15. T-s diagram for the 4-stage regenerative Brayton cycle.

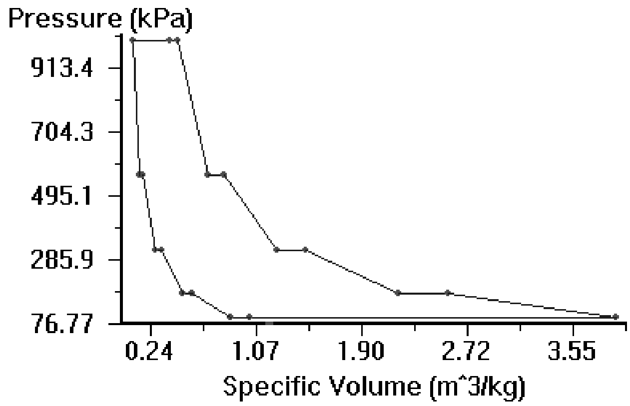

Fig. 16. P-v diagram for the 4-stage regenerative Brayton cycle.

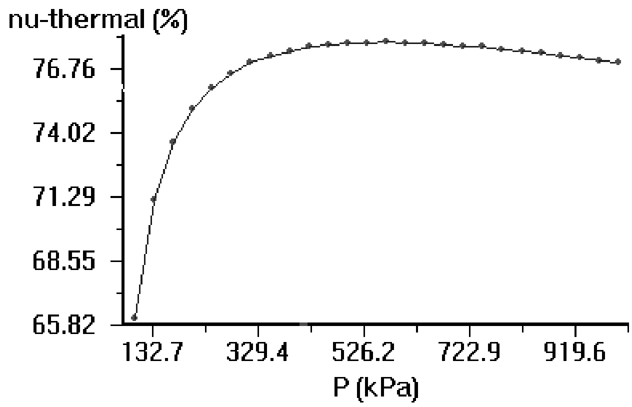

Fig. 17. Thermal efficiency as a function of first reheat stage pressure (P_2 in Fig. 14).

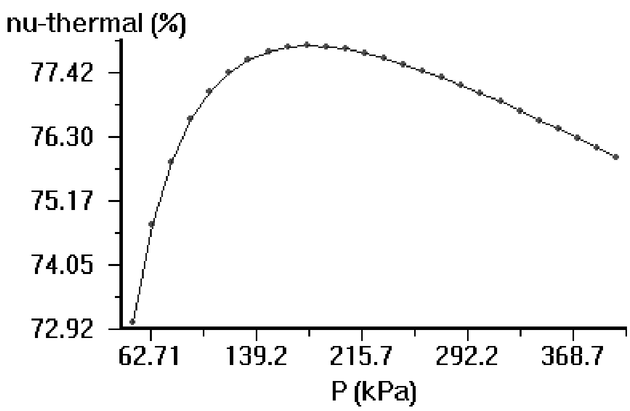

Fig. 18. Effect of first intercooling stage pressure (P_{11} in Fig. 14) on η.

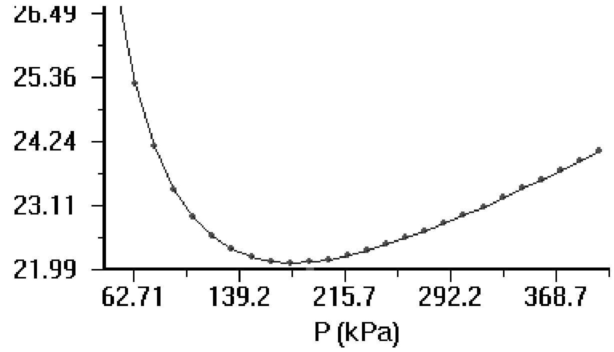

Fig. 19. Effect of first intercooling stage pressure (P_{11} in Fig. 14) on backwork ratio.

approaching the Carnot limit. However, the contribution of each additional stage is typically less and less, and the use of more than two or three stages is seldom justified economically. Backwork ratio has dropped by 4% indicating further reduction in compressor work input yielding higher net work. Mean temperature of heat addition is higher along with a lower mean temperature of heat rejection. Figure 16 illustrates the 4-stage cycle on the P-v plane, as generated by the cycle diagram feature of CyclePad.

As part of our review of the *sensitivity analysis* feature of CyclePad, we present the effect of changing pressure in the first reheat stage on the efficiency (Fig. 17) of the 4-stage cycle where it is observed that there is an optimum pressure at which the efficiency peaks. A similar trend is noticed for the first intercooling stage pressure which has an optimum value for the efficiency to be maximum (Fig. 18). We also record the effect of intercooling stage pressure variation on backwork ratio (Fig. 19) and as a design feature, one can choose the best pressure level where the backwork ratio is the least from this plot.

Thus it is evident that a fairly extensive amount of cycle design and analysis can be performed with this tool quite quickly, which otherwise would be a tedious job for the students and would mostly be left unattempted in absence of such a tool. Recently the software has been introduced in a course on engineering thermodynamics at the University of Canterbury (ENME 335) for the Junior level (2^{nd} professional year) and in an optional course on Energy Engineering (ENME 445) for the final year seniors (3^{rd} professional year). Initial student reaction has been extremely encouraging and a formal survey is being undertaken to identify areas for possible improvements in application. It seems to have generated more interest and motivation to learn thermodynamics. A user's manual [5] has been prepared and made available to the students.

CONCLUSIONS

Thermodynamic design and analyses have been carried out on the sequential performance improvement of an air-standard Brayton cycle through staged enhancements from a design perspective employing a thermodynamics instruction software. A synergistic combination of qualitative physics and AI techniques have been employed to develop CyclePad to assist in teaching, design and research in applied thermodynamics and advanced energy conversion systems. It provides an articulate virtual laboratory, in terms of visualisation of the schematic combination of a variety of thermodynamic cycles. Such studies demonstrate that analyses of complex cycles can now be carried out as part of classroom instruction as well employing intelligent software environments. Sample sensitivity analyses have also been presented in the study to demonstrate the features of the software. CyclePad serves as an effective educational tool towards optimisation of energy conversion systems where a fair amount of complexity is inherent in the design, especially when multiple possible variations of the related design parameters need to be studied.

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