Development of Low-Exergy-Loss, High-Efficiency Chemical Engines

Investigators
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Abstract
Development of an apparatus capable of hosting combustion processes at volumetric compression ratios in excess of 100:1 and mean piston speeds of order 100 m/s has been successfully completed. Non-reacting gas compression tests indicate that the current piston/ring design is capable of achieving pressures as high as 90% of their isentropic values (based on volume) with a repeatability of about 1%. These values are more than adequate for a systematic investigation of combustion at extreme states.

Two preliminary, light-load combustion tests have just been conducted using methanol and diesel as fuels. While these initial results are more symbolic than systematic, they indicate that our strategies for fuel injection, injection synchronization, and energy (work) extraction are also functioning correctly, and that we are well positioned to investigate the requirements for combustion under very high-speed, extreme compression conditions.

In a parallel effort, we continue to investigate what is possible in the design space of ultra-high efficiency engines. While previous efforts focused on batch engine geometries (i.e., piston-cylinder), efforts during the past year have focused on understanding efficiency optimization for steady-flow (i.e., turbine) engines. Although similar in many respects, a key difference is the inability of these engines to provide reversible work interactions. Although still leading to a form of the same extreme-states principle that led to the experimental work cited above, this research shows that there exist intrinsic limitations on the state to which the working fluid may be compressed due to the inherent irreversibilities of the turbomachinery. Equally important, is that our preliminary assessment indicates that these states are well outside of the current engine operating envelope such that the possibility exists for significant improvement in the efficiency of steady flow combustion engines.

Introduction
Efficiency remains one of the most important tools available to combat carbon dioxide accumulation in the environment. While often identified only with the end uses of energy, efficiency is equally important in the transformations that occur long before end use. In particular, efficiency in the transformation of chemical bond energy to work—whether to turn an electrical generator or to drive a vehicle down the road—is a key resource that must be tapped in order to stabilize our environment. A device that performs the transformation from chemical energy to work is called an engine.

In this project we seek to develop ways to provide significantly more efficient engines than are available today. We do this by taking a very fundamental approach; we
consider the theoretical limits based solely on thermodynamics and work backwards from there to see where and why efficiency is lost, and therefore what might be done to recoup it. Unlike essentially all other engine research of which we are aware, we take the approach that unless proven impossible, potential approaches to ultra-high efficiency engines must be considered seriously, and the key issues that prevent their realization must be systematically understood and investigated.

The result of having taken this approach in a previous GCEP project was identification of an extreme-states principle for combustion engines: The only way to reduce the efficiency loss due to unrestrained reaction in an internal combustion engine is to conduct the energy conversion at the highest-possible state of energy density. Quantitative assessment of that idea in the context of using a piston-cylinder configuration for work extraction led to identification of a promising approach as being extreme compression (>100:1) at rates approaching an order of magnitude faster than current engines. While combustion at such states would require a very different engine design than what we currently have (most likely free-piston with electromagnetic extraction), the key research issues to be understood were identified to be whether combustion could be successfully conducted (initiated, phased, and completed) under such conditions, and whether the indicated efficiency after real-world-implementation losses would be sufficient to warrant development of such a class of engines.

This report gives details of progress achieved in the past year towards those goals. Results from both our efforts to identify the key issues associated with piston-cylinder combustion at extreme states, and our continuing efforts to identify possible engine architectures for achieving ultra-high efficiency are reported.

Progress and Results

Task 1: Demonstration of Low-Irreversibility Combustion by Extreme Compression

The past year has been focused on finishing the hardware design and implementation for the extreme compression device and testing the device’s capabilities to ensure that it can achieve the states necessary to demonstrate significant increases in efficiency. The bulk of the implementation work is now completed, and the device is well-positioned for exploring extreme-state combustion. Figure 1 shows a picture of the device and its supporting structure.

The single-shot, free-piston device is designed for rapid charge compression to compression ratios greater than 100:1 using a direct-injection, PCCI-like combustion strategy followed by rapid expansion. Using a free piston would normally require a new method for work extraction, such as the linear-alternator setup being investigated by researchers at Sandia National Labs [1]. Rather than implement this part of the device, efficiency will be calculated using indicated work, measured over a single cycle. Our objective is to show feasibility of the extreme-state principle: that entropy generation during combustion is significantly reduced when carried out at high energy states [2].
Figure 1: Photograph of the experiment with its supporting structure. The high pressure reservoir is located behind the working cylinder.

A cross-section of the device is shown in Fig. 2. The basic design includes a large poppet-valve assembly that allows air to move from the high-pressure reservoir into the cylinder, actuating the piston. The piston moves down the bore at high speed (with mean piston speed an order of magnitude faster than standard piston engines), compressing the air in the high-pressure base section. These high piston speeds help to reduce heat transfer by allowing for a very long bore (~2.5 m) such that our TDC surface to volume ratio is low.

The base section at the end of the cylinder is interchangeable which allowed for early prototyping to lower pressures (compression ratios up to 50:1) with a smaller base section. The new combustor base can operate at pressures over 1500 bar, allowing for stoichiometric combustion at compression ratios over 100:1, and lean combustion to much higher compression ratios.

During the past year, significant progress was made in a number of areas critical to realization of the overall apparatus. Details of this progress in several key areas is given below.
Subsystem Advancements

Valve Design: Last year’s report focused on the analysis and characterization of a prototype valve mechanism that was used to determine valve flow rate and opening time requirements. During the past year the actual valve was built and tested. A cross-section of the design is shown in Fig. 3.
The current valve, actuated by helium venting through several solenoids, opens in less than 15 ms and has high repeatability. The 40-mm-diameter poppet valve also allows for very high mass flow rates from the reservoir to the cylinder, enabling the desired compression ratios (as demonstrated below).

**Piston Design:** In the last report, we also outlined our strategy for using a high-tolerance, non-pressurized piston ring to reduce mass blowby [3]. The device’s long bore (~2.5 m) makes achieving a uniform oil film for hydrodynamic lubrication complicated, and its high piston speeds and pressures make a conventional, positive-sealing strategy difficult due to the likelihood of melting the rings. Figure 4 shows a diagram of the piston design that has resulted from iteration and testing. Three graphite bearings, energized by o-rings, travel along the length of the piston to keep the piston centered in the bore. A graphite ring at the front of the piston reduces the mass blowby with a diameter 0.0001-0.0002 inches less than the smallest diameter in the cylinder. A back ring with a slightly smaller diameter reduces the chance of metal-to-metal contact between the piston and the cylinder wall if one of the bearings were to fail. The back ring has three small notches to allow gas flow such that the pressure never builds up and pushes the graphite bearings off the wall. Finally, there is an aluminum disk at the back containing neodymium magnets to aid in position sensing (as discussed in the next section).

![Figure 4: Prototype piston design.](image)

During air-compression tests, recorded pressures are ~90% of the isentropic pressure based on the volumetric compression ratio (as determined from Hall sensor data). The losses are comprised of a combination of heat transfer and mass blowby. A modeling effort is now underway to determine the relative contributions of these two losses as well as a second effort focused on improving the sealing strategy. However, we note that while achieving low heat loss and mass leakage is important for efficient compression in an actual engine, the present design is more than adequate for the purposes of demonstrating combustion under extreme-state conditions.
**Sensing and Data Acquisition:** To measure indicated work, measurements of cylinder pressure and volume are required. One of the challenges for this device has been sensing piston position (for volume measurement) without penetrating the cylinder wall in such a way that might compromise sealing, high speed operation, or repeatability. Along the length of the bore, eleven variable-reluctance (VR) sensors produce a pulse when the magnets pass the sensor. These sensors respond to changes in the magnetic field and are therefore velocity dependent. While they work well for positions along the bore where the piston is moving, they do not work well at TDC when the piston’s velocity is zero. For TDC measurements, a Hall sensor works well since it senses magnetic field directly. From the combination of these sensor signals, the piston position-time history can be readily reconstructed.

The position sensors are read by a 16-bit, 16-channel, multiplexed Iotech data acquisition board. The board can sample at 1 MHz which allows for sampling all 11 VR sensors, a Hall sensor, and various static pressure sensors at a reasonable rate. The board also has 1 MSample of on-board memory, approximately the length of one experimental data set for our device, which ensures that the data are not overwritten should there be a problem with the computer operating system accessing the data immediately.

A PCB 102A03 piezoelectric pressure transducer is used for cylinder pressure. This sensor measures pressures to ~1000 bar and is capable of following signals with frequencies up to ~125 kHz (1/4 of its resonant frequency). The resolution is 0.014 bar and the non-linearity is less than 1% full scale. This sensor is read by a Gagescope high-speed, A/D board with 12-bit resolution. The Gagescope has two channels, enabling high speed sampling without cross-talk across channels. The second channel on this board reads a PCB 101A04, 0-70 bar piezoelectric pressure transducer positioned on the driver side of the device. Figure 5 shows the layout of our sensing and data acquisition system.

**Figure 5:** Layout of sensors and data acquisition.
**Combustion Base Piece:** The combustion base piece is made from a 4340 VAR alloy steel, heat-treated forging. VAR stands for vacuum arc remelt, a process that reduces the possibility and size of inclusions in the stock metal. This forging was then machined with a bore that matches our current cylinder to within 0.0002 inches. Five injector ports, an exhaust valve port, and a pressure transducer port are also included. Optical access is provided via a window assembly that threads in from the bottom and provides seating and sealing for a conical sapphire window. A steel plug can also be used in place of the sapphire window for experiments that do not require optical access. A cross-section of the combustor is shown in Fig. 6.

![Cross-section of combustor](image)

**Figure 6:** Cross-section of combustor.

The combustor’s large size (400 mm diameter) required that the billet come from a forging whose defects and inclusions are not necessarily consistent or predictable. There were also uncertainties, due to the vessel’s size, about the material properties after the heat-treatment near the center of the device (where combustion takes place). To determine these material properties, a core from the center of the device was obtained and tested for yield strength. These data were then used in the finite-element model we had developed to analyze the device. To test the combustor for material defects, the finite element model was used in combination with strain gauges and hydrotesting to verify the combustor’s performance at high pressures. The device was tested to 2000 bar (30,000 psi) with the model showing similar strains (to within ~10% of the actual values). It is therefore likely that the device does not have any irregularities, discontinuities or large inclusions that will alter its performance through the desired range of working pressures (to 1000 bar).

**Fuel Injection System:** The fuel injection system is built around Bosch CRIP2 injectors from a common-rail diesel system used in passenger cars. The injectors are of the valve-
covered orifice (VCO) type and operate at a rail pressure of 1500 bar. The stock injectors are being used with two modifications: The injector clamping mechanism was strengthened to withstand our much higher cylinder pressures, and several custom-drilled nozzles have been obtained that allow us to use a variety of spray patterns. The injectors are connected to the stock Bosch common rail. At 1500 bar the compressibility of liquid fuels is significant, and the rail acts as an accumulator volume to maintain a roughly constant pressure during operation.

When running with diesel fuel, the rail pressure is supplied by a Bosch CP3 pump powered by an electric motor. A pulse-width modulated solenoid bypass valve controls rail pressure via feedback from a rail mounted pressure transducer. When running with alcohol fuels, the poor lubricating quality of the fuel could rapidly damage the Bosch pump. In this case, a Haskel air-powered intensifier pump provides rail pressure. This has the advantage of not using the fuel as a lubricant, however it does have the disadvantage of some fluctuation in rail pressure during pump cycling.

Unlike a conventional slider-crank engine, timing of the free piston is only set by initial conditions and is not directly controlled. Hence the timing of the actual piston stroke must be acquired prior to injection in order to properly phase combustion. To accomplish this, the output of one of the variable-reluctance position sensors triggers a counter on the data-acquisition hardware which then sends the trigger to the injector driver at the appropriate time. The injector driver is a commercial system from Genotec which, upon receipt of the trigger, provides a suitable current profile to the injector.

**Experimental Data for Air Compression**

Initial testing consisted of compressing air without fuel to verify that the device could achieve the desired compression ratios and to gauge repeatability. The first tests used a lower-pressure base piece which only allowed us to reach compression ratios of 50:1. An example of the pressure and volume profiles for these first tests is shown in Fig. 7.

A reservoir pressure of ~10 bar was required to achieve the compression ratio of 47. This pressure is out of a 70 bar range, verifying that much higher compression ratios are possible by increasing the driving pressure. Similarly, the mean piston speed (MPS) for this run was ~60 m/s with peak speeds near 80 m/s. These early experiments used a relatively heavy, steel piston (~0.83 kg); a lighter piston (shown later) can be used to achieve higher piston speeds.

The peak pressure achieved experimentally is ~90% of the isentropic pressure based on volume. The losses are a combination of mass blowby past the piston rings and heat transfer. Efforts are now underway to determine the relative proportions of these two loss mechanisms using a finite element package to model the system.

Figure 8 shows the pressure-volume profile for the same experimental data set. The black line shows the isentropic pressure-volume profile for the same volumetric compression ratio.
Figure 7: Pressure vs. time and volume vs. time for an air compression ratio of 47. The points on the volume vs. time plot indicate the points at which sensor data was taken from the VR sensors or the Hall sensor.
Figure 8: Full pressure-volume trace and a zoomed in version of the pressure volume profile. The isentropic profile starts to deviate from the actual profile at a volume ratio of 0.07. The star markers in (b) show data points from position sensors.
Another feature that was important to verify during initial testing was repeatability. The device showed less than a 1% change in compression ratio across multiple tests with the same operating conditions. This repeatability is largely due to the piston ring design where friction is low and essentially independent of pressure.

Air compression to higher pressures was also validated with the new combustor. Figure 9 shows a data set with a CR of 74:1. A reservoir pressure of 21.4 bar was used to achieve this compression ratio and the mean piston speed was 105 m/s.

![Figure 9: Pressure and volume profiles for air compression to a compression ratio of 74:1.](image)

**Initial Experimental Data with Combustion**

As described in the fuel injection system section of this report, the Bosch CRIP2 injectors have been modified with custom-drilled nozzles. To begin combusting, a nozzle with a single hole was used in just one of the five injectors. This was done so that the overall stoichiometry (and hence pressure rise) was kept very low—a prudent way to begin combustion testing.

Initial combustion runs were obtained just prior to the writing of this report. Both neat methanol and diesel fuel have been successfully burned. Figure 10 shows the location of two of the initial combustion tests in the operating domain of piston speed and effective compression ratio (red circles). For comparison, some specific data points and operating regions from current engine designs, and the operating range of the Sandia high-pressure combustion bomb, are also shown. Systematic exploration of this operating space is our key objective over the next several months.
Figure 10: Mean piston speed vs. effective compression ratio for a range of current engines. DI = direct injection, 4S = 4 stroke, NA = naturally aspirated, TCAC = turbocharged and after-cooled. [4]
Task 2: Theoretical Investigation of Low-Irreversibility Engines

A comprehensive approach towards understanding the thermodynamics of high-efficiency piston engines has been developed using a dynamical-systems approach, as was described in our previous report [3]. This involved minimization of entropy generation (thereby, exergy destruction) in a generalized piston engine with active energy transfers in the form of work, heat, and matter, and passive energy losses due to heat transfer and blowby. While details of the results and in-depth analyses are available in [2], some fundamental principles and conclusions from that work are as follows:

1. Entropy generation due to unrestrained chemical reaction can be minimized only by performing combustion at high internal energy states.

2. Entropy generated due to mixing of reactants is small in comparison to that generated by combustion.

3. To minimize net entropy generation during combustion, work extraction in the piston engine cycle must be performed only after complete combustion.\(^1\)

4. The optimal cycle minimizing net entropy generation involves bang-bang control of heat, matter, and work as energy transfers.

Though these principles were developed using a piston-engine system, they can be extended to simple-cycle, steady-flow engines (gas turbines). However, gas-turbine engines have an additional source of entropy generation due to kinetic energy dissipation in turbomachinery. This additional mode of irreversibility accounts for about 22% of the exergy destroyed in simple-cycle, gas-turbine engines and, as such, cannot be neglected.

Extending the theoretical studies of low-irreversibility cycles to simple-cycle, gas-turbine engines, and incorporating irreversibilities associated with turbomachinery has been the central task undertaken during the last year.

Gas-Turbine Engine Model and Optimization

The quasi-one dimensional model considered for steady-flow engine optimization is shown in Fig. 11. No specific engine architecture or thermodynamic cycle is assumed. The flow axis of the engine is the longitudinal dimension. The reactants are unmixed at the inlet, and complete combustion products emerge at the outlet. No other matter transfers are permitted. Passive heat loss is negligible in practical gas-turbine engines, hence this mode of energy transfer is not considered. Energy transfers as work are permitted at any location in the engine.

\(^1\) For hydrocarbon fuels, the reaction is almost complete at the switching point. Any difference is negligible.
The thermodynamic state of the gas as it passes through the engine is defined using the local thermodynamic properties $h(x), P(x), Y(x)$ for differential control volumes at every location $x$, where

$h(x) = \text{mass-specific enthalpy of the gas at the location } x$

$P(x) = \text{pressure of the gas at the location } x$

$Y(x) = \text{mass-fraction vector of chemical species in the gas at the location } x$

Noting that the $x$ coordinate is a time-like coordinate, it can be replaced by $t$, and all rates of change in thermodynamic properties may be discussed in terms of temporal derivatives of the working fluid.

Since we are interested in the minimization of entropy generation, we need to look at the change in entropy of the gas as we move ahead in time $t$. We can divide the time-evolution of the thermodynamic state into three periods, based on the mixing of fuel and air, i.e., unmixed, mixing, and mixed. Using Gibbs equation we can write the following equations for the change in entropy of the composite fuel-air system:

$$ds = \frac{\dot{m}_{\text{fuel}}}{\dot{m}} \left( \frac{dh_{\text{fuel}}}{T_{\text{fuel}}} - \frac{\nu_{\text{fuel}} dP}{T_{\text{fuel}}} \right) + \frac{\dot{m}_{\text{air}}}{\dot{m}} \left( \frac{dh_{\text{air}}}{T_{\text{air}}} - \frac{\nu_{\text{air}} dP}{T_{\text{air}}} \right) \quad (t < t_{\text{mix}}) \quad (1)$$

$$ds = \delta s_{\text{mix}} \quad (t = t_{\text{mix}}) \quad \text{Instantaneous mixing} \quad (2)$$

$$ds = \frac{dh}{T} - \frac{\nu dP}{T} - \sum_j \left( \frac{\mu_j dY_j}{M_j T} \right) \quad (t > t_{\text{mix}}) \quad \text{Homogeneous reaction} \quad (3)$$
We can relate the change in state to energy transfers using the first law of thermodynamics

\[ \delta w = dh + d(k.e.) \] (4)

Energy transferred to the system as work manifests in two modes, namely, enthalpy \((h)\) and kinetic energy \((k.e.)\). Kinetic energy is entropy-free by itself, but entropy is generated when energy is exchanged between these modes. When \(k.e.\) is converted to \(h\) in a diffuser or at the compressor blades, or vice-versa in a nozzle or at the turbine blades, a part of it is dissipated due to viscosity and entropy is generated. The fraction of \(k.e.\) dissipated depends on the local flow field and can be related to the gradients in the flow. However, in a general thermodynamic formulation the dissipated energy can be accounted for by introducing an irreversibility factor \(\sigma\)

\[ \delta w = (vdP + d(k.e.))\sigma \] (5)

Using equations (4), (5), and observing that all changes in the entropy of the system in an adiabatic engine are due to entropy generation, the entropy-change equations can be modified to the following:

\[
\delta S_{gen} = (\sigma - 1) \left\{ \frac{\dot{m}_{fuel}}{\dot{m}} \left( \frac{d(k.e.)_{fuel}}{T_{fuel}} + \frac{V_{fuel}dP}{T_{fuel}} \right) + \frac{\dot{m}_{air}}{\dot{m}} \left( \frac{d(k.e.)_{air}}{T_{air}} + \frac{V_{air}dP}{T_{air}} \right) \right\} \quad (t < t_{mix}) \] (6)

\[ \delta S_{gen} = \delta S_{mix} \quad (t = t_{mix}) \quad \text{Instantaneous mixing} \] (7)

\[
\delta S_{gen} = \frac{vdP(\sigma - 1)}{T} + \frac{d(k.e.)(\sigma - 1)}{T} - \sum \left( \frac{\bar{\mu}_j dY_j}{T} \right) \quad (t > t_{mix}) \] (8)

where,

\[ \bar{\mu}_j = \frac{\mu_j}{M_j} \quad \text{(chemical potential expressed on a mass basis)} \] (9)

The \(dP\) and \(d(k.e.)\) terms represent the entropy generation due to irreversible compression/expansion and the \(dY\) term represents the entropy generation due to combustion.

The irreversibility factor for work transfer can be related to the polytropic efficiency of the energy transformation by
Having modeled the local entropy generation, an optimal control problem for minimizing the total entropy generation in the gas-turbine engine can be formulated as follows:

$$\min s_{gen} = \int ds(h, P, Y, \dot{P}) dt$$

State Dynamics:

$$\dot{h} = v(h, P, Y) \dot{P} \sigma + \frac{d(k.e.)}{dt}(\sigma - 1)$$  (I Law of thermodynamics)

$$\dot{Y}_i = f_i(h, P, Y)$$  (Reaction kinetics)

Constraints:

$$|\dot{P}| \leq \dot{P}_{\text{max}}$$  (Bounded work transfer rate)

$$T \leq T_{\text{max}}$$  (Bounded temperature)

$$P \leq P_{\text{max}}$$  (Bounded pressure)

$$P_f - P_0 < \delta$$  (Exit criterion)

$$\xi(t_f) = 1 - \frac{s_{eq}(h, P, Y_{eq}) - s(h, P, Y)}{s_{eq}(h, P, Y_{eq}) - s(h, P, Y)} \geq \xi_0$$  (Reaction completion variable)

**Equivalence with Piston Engine Optimization**

The optimal control problem formulated above is similar to that developed for the adiabatic piston engine by K.Y. Teh [2]. The exception involved, is the inclusion of kinetic energy as an additional mode of energy storage, and the consequent irreversibility associated with its dissipation.

If the compression and expansion processes are considered to be reversible, then combustion becomes the only source of entropy generation. The problem then becomes identical to the adiabatic piston engine problem (with expansion to atmospheric pressure), albeit in $H - P$ thermodynamic coordinates instead of $U - V$ coordinates.
Results and Conclusions

Having extended the thermodynamic model to include simple-cycle gas-turbine engines with non-ideal work interactions, the proposed optimal control problem can be reduced to simpler problems that sequentially include individual entropy generation sources and factors (i.e., combustion, mixing, polytropic compression and expansion, pressure and temperature limits). The thermodynamic solutions to these reduced-model problems can then be used to build a complete understanding for steady-flow engines. The reduced-model problems are as follows:

1. Premixed cycle with reversible work: Entropy generation in this model is only due to combustion.

2. Non-premixed cycle with reversible work: Entropy generation due to mixing is also present in this model, in addition to that due to combustion.

3. Non-premixed cycle with polytropic work: Entropy generation due to irreversible compression and expansion is included, along with the previously considered sources of entropy generation.

The results and conclusions from the analysis of these models are discussed below. The roles of peak pressure and/or peak temperature limits are also discussed.

Premixed cycle with isentropic work and no limits on pressure or temperature

The cycle with minimal irreversibility can be described by the following bang-bang control strategy and switching criterion:

\[ \dot{P} = \begin{cases} +\dot{P}_{\text{max}} v(h, P, Y) < v_{eq}(h, P, Y_{eq}) & \text{(Compression)} \\ -\dot{P}_{\text{max}} v(h, P, Y) \geq v_{eq}(h, P, Y_{eq}) & \text{(Expansion)} \end{cases} \]  

(0.1)

Thermodynamically, this implies that the optimal process consists of:

1. isentropic compression of the premixed fuel-air mixture at the maximum possible compression rate, even as the mixture auto-ignites and combusts, and
2. isentropic expansion of the reacting mixture after it reaches the switching point.

This strategy minimizes entropy generation due to combustion and constitutes the extreme-state principle for this idealization of a gas-turbine engine. Auto-ignition effectively limits the maximum extent of compression in the cycle.

Non-premixed cycle with isentropic work and no limit on temperature

It can be shown that entropy generated due to mixing of fuel and air is much smaller than that generated by combustion at all temperatures and pressures of interest. Hence, we treat this contribution as a constant in the optimization problem, thereby making no change to the optimal control strategy (i.e., optimal cycle). However, allowing non-premixed cycles has other important implications. Non-premixed cycles allow us to
control the state at which combustion begins, hence we can take the mixture to higher pressures (by avoiding auto-ignition). The optimal cycle then consists of:

1. isentropic compression of fuel and air separately to a specified pressure ratio, with subsequent mixing of the fuel and air,

2. further compression of the reacting mixture until the switching point is criterion is reached, and

3. isentropic expansion after the switching point is exceeded.

The switching point criterion for this cycle is same as that of the premixed cycle. The control is also the same after the fuel and air have been mixed, which can be shown on the line diagram in Figure 12.

![Figure 12: Control strategy for an optimal, non-premixed cycle along the specific volume coordinate.](image)

This cycle is pressure limited only by the choice of the pressure ratio at which mixing of fuel and air is initiated. This pressure ratio can theoretically be chosen as high as materials permit. Increasing this pressure ratio causes combustion to occur at more extreme states, thereby leading to lower entropy generation, and greater work extraction. This can be seen by comparing two cycles as shown in Fig. 13.

**Conclusions for cycles with reversible work**

1. Entropy generation due to combustion can be reduced by combusting at high enthalpy states (extreme states).

2. More extreme states can be accessed by mixing fuel and air and initiating combustion at higher pressure ratios (which are only limited by material pressure limits). The only other limitation is due to the peak temperature in the cycle, which will be discussed after considering polytropic-work cycles.

3. If operating below material pressure limits, the optimal cycle involves compression until the switching point is reached. This entails compression during
combustion, a strategy which is qualitatively different from the constant H-P combustion in the conventional Brayton cycle.

**Figure 13**: Extreme state principle for gas turbine cycles with reversible work as the only allowable interaction.

**Non-premixed Cycles with irreversible work**

As discussed before, practical gas-turbine engines face significant exergy destruction due to irreversible compression and expansion. Finding a mathematical solution to this optimal control problem requires knowledge of the rate of change of kinetic energy at all locations in the engine, which is dependent on individual engine details. Hence, the following two-step thermodynamic approach has been taken to obtain a solution applicable to generic gas-turbine engines:

1. The entropy generated pre-equilibration (i.e., before the fuel and air have undergone complete combustion) is minimized by minimizing the entropy of the equilibrium attractor.

2. The entropy generated due to work extraction from the equilibrated products—dependent on the pressure of the post-combustion state—is added in, and the overall entropy generation is minimized. This provides a maximum pressure ratio for the cycle (corresponding to minimum entropy generation), beyond which net exergy destruction increases again.

While the first step can be performed for a general engine, the second step requires the knowledge of the engine’s design. However, entropy generation due to work extraction is a monotonic function of post-combustion pressure, so the cycle obtained
from the first step has the qualitative characteristics of the optimal cycle. This cycle will be termed the optimal cycle for the discussion henceforth, while a specific architecture will be assumed for quantitative plots shown below.

The optimal cycle thus obtained, has two significant differences from the previous optimal cycles:

1. Efficiency-based pressure limit: Unlike the pressure limit imposed by material strength constraints, this pressure limit is chosen to maximize efficiency. This limit is a consequence of conflicting requirements associated with minimization of exergy destruction due to combustion vs. viscous dissipation in turbomachinery. Exceeding this pressure ratio decreases exergy destroyed due to combustion but causes greater exergy destruction during the compression and expansion processes. The quantitative value of this limit depends on the exact engine architecture.

2. Multiple switching criteria: Compression or expansion can be performed at any location in the engine using different devices, each having its own polytropic efficiency; e.g., compressors, diffusers, etc. Furthermore, variation in polytropic efficiency for different blade stages in a compressor can be treated as series of compressors with different polytropic efficiencies. This leads to multiple choices in performing compression and expansion at any location in the engine. The optimal cycle determines the options we need to exercise to minimize entropy generation, thereby leading to multiple, device-based, switching criteria in the problem.

The control strategy for the optimal cycle before equilibration can be represented using the following line diagram:

Figure 14: Optimal cycle (pre-equilibriumation strategy)
The post-equilibration strategy for the optimal cycle is expansion to atmospheric pressure and purely axial, steady-flow velocity. The line diagram shown above assumes one each of the compressor, diffuser, nozzle, and turbine stages. It also assumes the polytropic efficiencies to be: (1) $\eta_D < \eta_C$ (2) $\eta_N < \eta_T$. However, there could be more than one of these devices with different efficiencies, thereby leading to many more switching points.

The pressure limit enforced by efficiency maximization depends on the engine design. For an engine with state-of-the-art turbomachinery with polytropic efficiency greater than 90%, this limit could be greater than 1000 bar. For engines having 70% efficient devices this limit could be 30-40 bar. The existence of this limit can be seen in Fig. 15 for an engine with an assumed polytropic efficiency of 75% for the compressor and the turbine and neglecting kinetic energy of the flow. The plot is based on thermodynamic calculations using natural gas as the fuel.

![Figure 15: Pressure limit enforced due to efficiency considerations in the optimal cycle](image)

The plot shows a decrease in net work output from the engine if operated at 150:1 instead of 25:1, although combustion exergy destruction is reduced.

**Conclusions for cycles with irreversible work**

1. Extreme-state principles still apply, however, due to entropy generation during work investment there is a maximum pressure beyond which efficiency cannot be increased. This limit is different from the limit imposed by material strength considerations.
2. If the cycle is designed to operate below the above-mentioned pressure limit, the optimal switching strategy must be employed to maximize efficiency. If the cycle is operated at the pressure limit, i.e., the fuel and air are mixed at this pressure ratio, no further compression is needed since combustion will occur at the pressure corresponding to minimum exergy destruction.

3. The switching criteria for the optimal cycle are device-based and hence determine the appropriate location of different devices along the engine flow path.

Peak Temperature Limit
Temperature limits in gas-turbine engine cycles are chosen to ensure durability of the blade material. Although this constraint proves to be very restrictive, appropriate blade cooling strategies and core-gas cooling strategies must be investigated in conjunction with regenerative reinvestment of the coolant. Efforts to address these issues are currently underway.

Future Plans
The extreme-compression device has established its ability to achieve the thermodynamic states required to demonstrate a significant reduction in combustion irreversibility. The remainder of this year will be spent exploring combustion phasing and its effect on efficiency at these states. When it arrives, our sapphire window will also allow for visualization of the combustion event and a more complete understanding of the details of the process.

The thermodynamic framework established thus far, will be extended to regenerative cycles involving reinvestment of heat and matter, and will be used to develop high-efficiency regenerative cycles. A wide variety of wet cycles, recuperative cycles, etc., can be understood using this framework. Management of kinetic energy is crucial in developing efficient aircraft engines. The thermodynamic framework will also be extended to investigate aircraft engines and thereby improve efficiency.

Publications

References

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