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Exploring the Pathway to High Efficiency IC Engines through Exergy Analysis of Heat Transfer Reduction

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ABSTRACT

Heat transfer is one of the largest causes of exergy destruction in modern engines. In this paper, exergy distribution modeling was used to determine the potential of reduced engine heat transfer to provide significant gains in engine efficiency. As known from prior work, of itself, reducing heat transfer creates only a small increase in efficiency-most of the exergy is redirected into the exhaust stream-requiring both mechanical and thermal recovery of the exhaust exergy. Mechanical regeneration, through turbocharging and over-expansion, can lead to efficiencies exceeding 50%. Adding thermal regeneration, through high enthalpy steam injection or a bottoming cycle, can increase the efficiency potential to approximately 60%. With implementation of both mechanical and thermal regeneration, the only remaining cause of substantial exergy destruction is the combustion process. Thus, efficiency gains significantly beyond 60% are only possible by reducing the entropy generated in the fuel conversion process.

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INTRODUCTION

Internal combustion (IC) engines are one of the most common chemical energy-to-work conversion devices in the world. This is especially true in transportation, where piston/cylinder IC engines power most personal and commercial ground vehicles. As a result, there have been many efforts to improve IC engines by increasing power, improving efficiency, and reducing emissions. This has been especially true in the last few decades, where stringent emissions and fuel economy constraints have mandated significant changes in overall system design and control.

Reducing heat transfer losses to the coolant jacket holds promise for making additional, substantive gains in IC engine efficiency. Currently, engines lose a large fraction (a quarter to a third) of the incoming fuel energy to heat transfer, which significantly limits their efficiency. In order to quantify the potential opportunities, we developed a zero-dimensional model to explore the effects of heat transfer reduction. The model parameters were chosen to agree with measured engine data wherever available. A refined heat transfer model was implemented to account for convection from both mean and turbulent flow in the cylinder, such that total heat transfer compares well with the Hohenberg correlation while the crank-angle-resolved values agree well with measured heat flux data. The details of the model, including verification by comparison with measured data, are included in an Appendix.

In this paper we use the model to analyze how engine energy and exergy flows change with reduced heat transfer, and explore engine configurations-some novel and some already under consideration-that exploit these changes to enable significant gains in efficiency. We will begin by examining the energy and exergy distributions of three modern, canonical engines. Next we discuss possible methods to reduce engine heat transfer, and show the effect heat transfer reduction has on the exergy distribution. This leads to possible configurations that exploit low heat rejection (LHR) to improve efficiency-including turbomachinery, charge dilution, and thermal regeneration. We conclude by acknowledging the implementation challenges associated with such configurations.

ENERGY AND EXERGY DISTRIBUTIONS

The efficiency of an engine is usually measured by its work output per unit energy input, which makes it important to understand the possible outflows for the input fuel energy. An energy balance performed on an IC engine shows that the inflow of fuel energy is balanced by three significant outflows: work, exhaust enthalpy, and heat transfer. In traditional SI engines the incoming energy is divided roughly equally between the three outflows, yielding a brake, LHV-based efficiency that peaks in the mid-to-high 30's (%).

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Traditional CI engines are more efficient-due primarily to their higher compression ratio and lean, unthrottled operation-but the energy breakdown remains similar, with the loss divided roughly equally between the exhaust and heat transfer. From this the motivation for designing a LHR engine is clear; the reduction of energy outflow as heat might, in part, be redirected into work, increasing the efficiency of the engine.

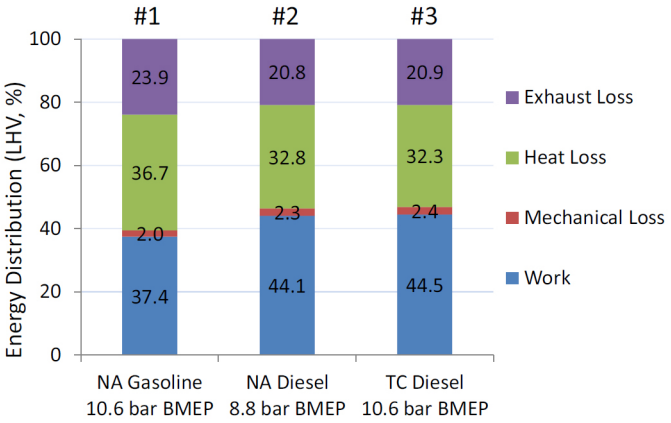


Figure 1. Energy breakdowns of three modern engine configurations (see Table 1 for details): 10:1 naturally aspirated SI, 17:1 naturally aspirated CI, and 17:1 turbocharged and intercooled CI.[1]

Figure 1 shows the energy distributions derived from modeling three conventional engine configurations: a naturally-aspirated SI engine with a 10:1 compression ratio (#1), a naturally-aspirated CI engine with a 17:1 compression ratio (#2), and an intercooled, turbocharged CI engine with a 17:1 compression ratio (#3). The engines were assumed to have aluminum heads and pistons, but steel/iron valves and liners. The model parameters for these cases, and all the other cases examined, are given in Table 1 in the appendix. The results show that over half of the chemical energy inducted into the engine is transferred to the environment through heat loss and exhaust enthalpy, limiting the brake efficiency to around 40% in all three cases. It is important to note that the results in Fig. 1 are the peak efficiencies for engines operating at high load. Throughout this paper we will only consider peak engine efficiencies and operation without intake throttling.[2]

While first-law analysis is useful for examining energy flows in an engine, incorporating the second law is essential for understanding efficiency limits. This necessitates examining the engine on an exergy, as opposed to energy, basis. Exergy analysis tracks the work that can be obtained from a given fuel resource, and helps to identify both where work capacity is destroyed (by entropy generation) and where unused exergy is discharged to the environment.

The ideal engine would convert 100% of its incoming exergy to work. However, any real engine will have lower exergy efficiency due to architectural constraints (basic engine design) and imperfections in the components used to implement the design. When using exergy analysis, it is important to realize the distinction between exergy transfer and exergy destruction. Exergy destruction occurs whenever there is entropy generation (through irreversibility). This includes entropy generation due to unrestrained chemical reaction (i.e., the combustion process), heat transfer across a finite temperature difference, expansion or compression in imperfect devices, and the mixing of gas streams of differing thermodynamic state or composition. Exergy destruction always lowers efficiency.

In contrast, exergy transfers do not necessarily reduce the maximum possible efficiency of an engine. Exergy transferred from the cylinder to the exhaust port (during blowdown and the exhaust stroke) does not necessarily constitute exergy destruction; for example, the exergy could still be utilized by an exhaust turbine.[3] This is also true for heat transfer, which carries exergy to the cooling system. However, it should be noted that exergy transferred to the environment is destroyed as it irreversibly equilibrates with its surroundings. This includes exergy emitted in the exhaust stream and heat transferred from the cooling system through the radiator.

Exergy analysis can be used to locate possible areas for improvement, as maximum efficiency is achieved when exergy destruction is minimized. This includes not only minimizing exergy destruction within the system, but also minimizing exergy transfer to the environment. The perfect engine would generate no entropy and would reject no exergy to the environment. While this ideal cannot be realized, it is still possible that architecture changes (or device improvements) can make significant progress towards this limit. Exploring the pathway toward that objective, specifically through heat transfer reduction, is the central topic of this study.

[1] Mechanical loss is determined from the ratio of brake work to indicated work; it does not include pumping work. Although it ultimately leaves the engine as heat transfer through the radiator or by passive loss, we choose to separate it from gas-to-surface convective heat transfer from the working fluid and therefore report it separately.

[2] Providing dynamic range without reducing efficiency is another important research area, but here we choose to focus on improving peak efficiency only.

[3] The blowdown process (and any throttling process) is irreversible and does destroy exergy, but the exergy remaining in the stream afterward can still be extracted

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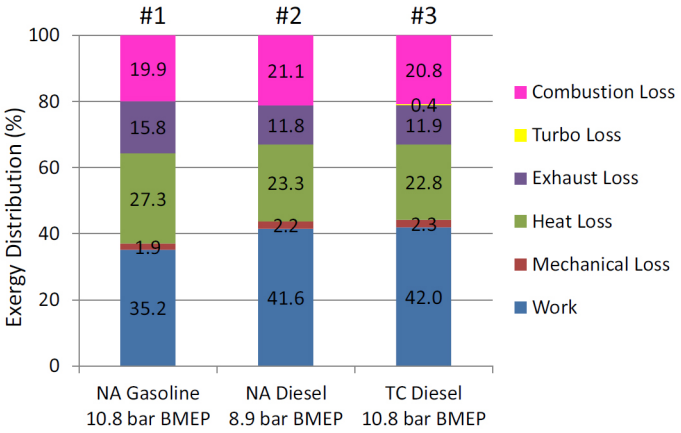


Figure 2. Exergy breakdowns of three modern engine configurations: 10:1 naturally aspirated SI, 17:1 naturally aspirated CI, and 17:1 turbocharged and intercooled CI.

The exergy breakdowns for the same engine configurations as Fig. 1 are shown in Fig. 2. Although the work remains unchanged for each engine, the percentages change due to the two different bases (LHV vs. exergy). However, the exergy balance also indicates where the remainder of the work potential is destroyed. In each case approximately 20% of the incoming exergy is destroyed by the entropy generation of the combustion reaction used to convert the chemical energy in the fuel to sensible energy in the combustion products.[4] Stated another way, the upper limit of exergy efficiency has been reduced to around 80% by choosing combustion to transform the fuel's chemical energy. Although combustion entropy generation can be decreased by performing combustion at high internal energy states [1], we will accept combustion exergy destruction as a result of the engine architecture and focus our analysis on other means of improvement.[5]

Comparable to combustion, heat transfer is the other large loss of exergy in these engines. It is worth noting that heat transfer exergy destruction occurs across three temperature differences: first from the gas to the wall (boundary layer convection and radiation), then from the wall to the coolant (conduction, convection, and boiling), and lastly from the coolant to the environment (through the radiator and passive loss).[6] Figure 3 shows the distribution of the exergy destruction due to heat transfer that occurs at each location in our model of a typical SI engine with variable surface temperatures, a fixed 373K coolant temperature, and a fixed 300 K environment temperature. The data show that almost half of the exergy destruction occurs in the heat transfer between the gas and the wall. This demonstrates the necessity of eliminating heat transfer to the walls; even a perfect thermal device operating between the wall and environment could recover barely over half of the exergy lost through heat transfer.

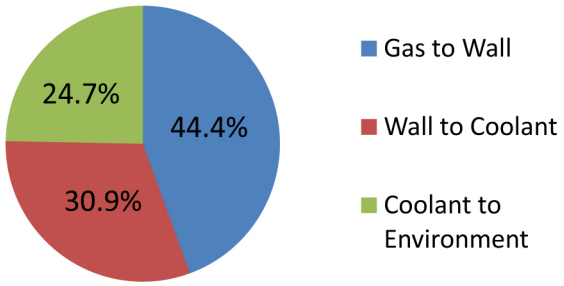


Figure 3. Distribution of total heat transfer exergy destruction by location in a conventionally cooled SI engine.

Accepting that exergy destruction due to combustion is inevitable leaves two options for improving engine efficiency. Exergy transferred to the environment-either as matter in the exhaust stream or heat through the coolant jacket-needs to be retained within the system and converted to work in order to achieve significant efficiency improvements. Although it is ultimately necessary to reduce both transfers, we will focus on the change in exergy distribution due to reduced heat transfer, and explore how that can be used to make substantial improvements in efficiency.

Heat Transfer Reduction Strategies

There are two ways to reduce heat transfer between the cylinder gas and surface. The first is to reduce the heat transfer coefficient, which is a function of the fluid properties and motion. This is difficult to do in traditional engines since the turbulence necessary to promote flame propagation or fuel-air mixing (for SI and CI engines respectively) also contributes to high heat transfer. This problem is compounded by the high combustion temperatures and surface-to-volume ratio near TDC. However, high gas viscosity (due to the burned gas temperature) during the expansion stroke helps to laminarize the cylinder gas, which in turn reduces the heat transfer coefficient after combustion.

Heat transfer can also be minimized by reducing the temperature difference between the gas and the walls, which can be accomplished by decreasing the gas temperature or increasing the surface temperatures. Decreased peak temperature is often obtained by diluting the charge with excess air or exhaust gas. This approach is used extensively

[4] A small portion of the "combustion" column actually corresponds to the mixing of the fuel and air streams, not chemical reaction. This mixing is essential for the combustion process, so we choose to combine it with the exergy destruction resulting from the combustion reaction.

[5] Many of the engine configurations used in this paper will decrease combustion irreversibility, although that is not the direct motivation for proposing them, and the resulting gains are relatively small compared to other improvements.

[6] Radiative heat transfer is usually only significant in CI combustion, due to soot formation.

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in homogeneous charge compression ignition (HCCI) engines, high efficiency dilute gas engines (HEDGE) [2], lean-burn engines, and other configurations. A large fraction of the efficiency gains in these designs is due to reduced heat transfer losses resulting from the reduced peak temperature. However, in all of these examples, the work output suffers as the fuel input to the system is reduced with dilution.

We choose to examine the alternative to decreased gas temperature via dilution-increased surface temperatures due to increased thermal resistance in the wall. This could be accomplished by using ceramic materials on the combustion chamber surface. The low thermal conductivity of the ceramic leads to high temperatures at the gas/wall interface, which reduces heat transfer. There have been many attempts, both through modeling and experimentation, to determine the benefits of an LHR engine design with ceramic parts or coatings. Some of these attempts illustrate the promise of LHR strategies, while others demonstrate the significant challenges that accompany LHR engine design.

Previous LHR Research Efforts

Starting in the 1980's, there have been numerous research efforts that demonstrated increased efficiency through elevated surface temperatures in compression ignition engines. Some efforts, such as those by Thring [3] or Moore [4], indicated that an uncooled cylinder liner was critical to reduce heat transfer loses. However others, such as Wade [5] and Miyairi [6], were able to achieve significant efficiency gains with ceramic coatings on the head, piston, and valves, but with a cooled cylinder liner. It seems reasonable that liner temperature would only have a small effect on heat transfer (compared to head and piston temperature) since the heat flux is highest near TDC where relatively little cylinder wall is exposed. Furthermore, by the point in the expansion stroke where the walls form a significant portion of the surface area, the expanding gas will have cooled, and the flow will have laminarized, reducing heat transfer losses. These engines, and many other LHR research engines, demonstrated approximately a 50% reduction in the in-cylinder heat rejection, with a corresponding increase in fuel economy of about 10%.

Even in successful LHR efforts, there were still several problems that had to be addressed. In all cases, volumetric efficiency decreased as the intake charge was heated by hot surfaces during induction. As a result, researchers like Taymaz [7] argue that boosting to increase load is necessary to allow LHR configurations to be viable alternatives to conventionally cooled engines. This also illustrates the need to maintain a cool intake valve and port, even as the other surface temperatures increase. Parlak [8], in addition to Taymaz, observed higher exhaust enthalpy in the LHR configuration. In both cases a large fraction of the heat transfer reduction was redirected into exhaust enthalpy rather than increased efficiency. This demonstrates that exhaust enthalpy utilization is critical to make large efficiency gains in LHR engines.

LHR engine research also demonstrated the possibility for improved engine emissions. Thring and Assanis [9] observed reduced hydrocarbon, carbon monoxide, and soot emissions. This was attributed to the increased gas temperature resulting from the thermal barrier coatings, aiding in the oxidation of incomplete combustion products. However, it should be noted that NOx emissions increased in all the LHR engine configurations due to the positive correlation between combustion temperature and NOx formation.

Despite the promise shown by some LHR engine studies, other efforts saw no improvement in efficiency despite achieving lower heat transfer losses. In nearly all of these cases this was the result of poor combustion. Cheng and Wong [10] observed poor performance at all engine loads which they attributed to non-optimized combustion timing. Dickey [11] observed degraded combustion due to less premixed burning. Both of these cases show that the increased gas temperature of LHR engines has a significant effect on ignition delay that cannot be ignored. Kimuru [12] attributed poor combustion to “low flame angular velocity”- effectively a measure of swirl. This shows that high temperatures may also damp fluid motion (through increased viscosity), making fuel-air mixing a greater challenge. In all of these cases smoke emissions increased. These efforts show that careful optimization of combustion is necessary to achieve efficiency improvement in LHR engines, and also that traditional CI strategies and timings may by ineffective in LHR configurations. While combustion optimization presents a design challenge, it is not a fundamental flaw in the LHR engine strategy. In recent years improved fuel injection systems have allowed for more precise control of CI combustion, and it seems likely that combustion can be efficient and optimized even in a LHR engine.[7]

While the prospects of controlling CI combustion in a high-temperature environment have improved, there are other issues that remain unresolved. Perhaps foremost among these is the challenge of lubricating the hot surfaces present in LHR engines. Many conventional engine lubricants are not feasible for LHR engines; they would decompose at the elevated surface temperatures. Jaichandar [13] argues that even new synthetic lubricants would be unable to cope with the hot surfaces. This problem is compounded by the fact that many high-temperature lubricants cannot be pumped by conventional means at ambient temperatures, making lubrication during startup an issue. Research in self-lubricating composites by Sutor [14], Gaydos [15], and others demonstrate some promise in solid lubrication, but lubrication of hot surfaces remains a significant issue. It is important to note that engines with conventionally cooled

[7] This includes piezoelectric injectors capable of multiple injections, and better fuel/air mixing and atomization due to higher injection pressure.

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liners avoid this problem, and can still use typical engine lubricants, rings, etc.

Reduced Heat Transfer Results

Due to the challenge of lubricating a hot cylinder liner, we will consider only increased valve, port, head, and piston temperatures; the liner (except the region above the travel of the upper ring land) will remain conventionally cooled, allowing the use of traditional lubricants and rings. As stated in the previous section, this approach has already been implemented in research, and still may significantly reduce the heat transfer. Figure 4 helps to confirm this, as it shows that only 15% of the heat loss occurs through the liner in a typical SI engine with conventionally cooled surfaces. Insulating the exhaust port and valves, along with the head and piston, is much more essential to reduce heat transfer loses.

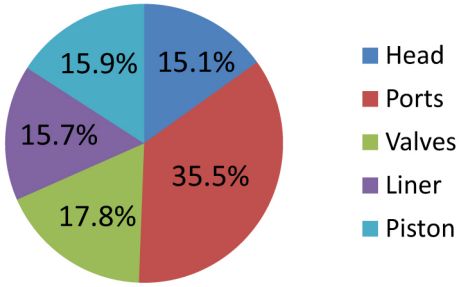


Figure 4. Distribution of heat outflow by location in a conventionally cooled SI engine.

In this paper we explore engine configurations with increased surface temperatures, and investigate various methods of exploiting the resulting heat transfer reduction. We will examine increased surface temperatures by applying a 2.5 mm thick thermal barrier coating with properties similar to 8YSZ in a conduction model. Additionally, all underlying metal was changed to iron/steel to take advantage of the lower thermal conductivity.[8] This is similar to the wall configuration used in Fujimoto's [16] LHR engine model, and will be used in all of the LHR configurations in this paper. Although this particular coating is somewhat thicker than examples seen in practice, it does not seem excessively so. Furthermore, this coating would not lead to an unrealistic reduction in heat transfer-modeling indicates that the reduction would be less than 50%. This heat transfer reduction was observed in several LHR research projects, and as such the coating model is not unreasonable for estimating the exergy flows of hypothetical LHR engines.

It should be noted that there is a difference between an *adiabatic* engine and an engine with *no net heat transfer*. An adiabatic engine would have *no* heat transfer, which is impossible to achieve. An engine with no *net* heat transfer would allow heat transfer both to and from the fluid, but these transfers would be balanced. While this yields zero net heat flow in the energy balance, there is still exergy destruction due to the heat transfer across a finite temperature difference both in-to and out-of the combustion chamber. This heat transfer is detrimental because the exergy flows out of the hot combustion products where it could have been utilized as expansion work, and flows into the charge during the induction process, heating the gas and lowering volumetric efficiency. These distinctions indicate that in modeling low heat rejection engines it is important to account for the heat transfer in both directions.

While our focus will be to determine the effects of heat transfer reduction, it is also important to address the effects of compression ratio. All of the configurations have a 17:1 geometric compression ratio. Although this choice is somewhat arbitrary, it is similar to the compression ratio of many boosted Diesel engines, which typically range from 16:1 to 18:1. We would like to choose a high compression ratio to improve efficiency, but it is likely that further increases in compression ratio would require high TDC surface-to-volume ratios, increasing heat transfer losses and *decreasing* efficiency. As a result, it seems that 17:1 will provide a reasonable starting point; increased compression ratio can no longer improve efficiency, so other improvements, such as heat transfer reduction, must be implemented to realize efficiency gains.

Since poor combustion was an issue in previous LHR experiments, it is also important to note that we will use a premixed charge in all the configurations, even though that is often not compatible with high surface temperatures, high compression ratios, and boosting. Recent research efforts, such as premixed charge compression ignition (PCCI), are focused on controlling combustion at high temperature and compression ratio. As a result we will take for granted that combustion can be properly phased and controlled at the conditions studied, even with stoichiometric mixtures.[9]

The ceramic wall coating model was applied to a 17:1 CR cast iron engine (#5) to define a baseline LHR configuration (#6). Figure 5 shows that the 17:1 engine is significantly more efficient than its 10:1 counterpart (#4). The reduced heat transfer configuration leads to a further increase in efficiency, but also with increased exergy transfer out in the exhaust. This is in accordance with the second law, which requires that a steady-state system reject the entropy it takes in and generates. Reducing entropy transferred out of the system as heat requires that this entropy be rejected elsewhere. In these engine architectures more entropy *must be* transferred out in the exhaust, typically leading to higher exhaust temperatures (and thus more exergy carried out in the exhaust).[10] The increase in efficiency (coupled with the

[8] The properties of steel and cast iron are alloy dependent, and are assumed to be identical to each other in this work. In practice, the liner, head, and ports in an LHR engine would probably be cast iron, while the valves, seats, and piston would be steel.

[9] It is likely that to control combustion some or all of the fuel will have to be directly injected. However, that should not have a significant effect on the modeling results.

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increase in exhaust enthalpy) calculated in our model is comparable to the results of the LHR engine experiments previously discussed. The combustion destruction has decreased somewhat-reduced heat transfer leads to combustion occurring at a higher internal energy state-which demonstrates an additional advantage of an LHR configuration. It is important to note that when using lower thermal conductivity metals like cast iron, coatings may be necessary not only to provide additional insulation, but also to protect the metal from experiencing excessive surface temperatures.

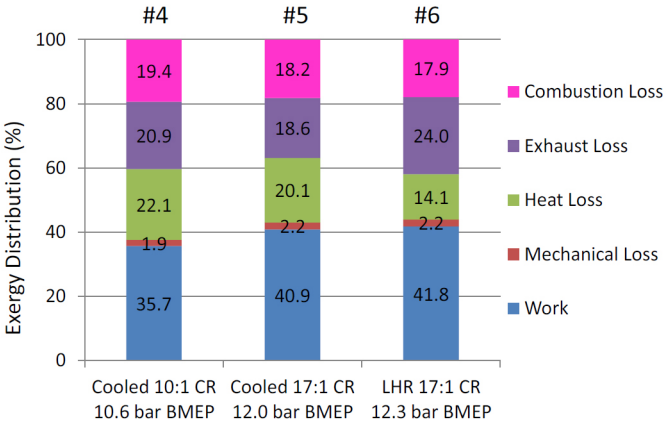


Figure 5. Exergy breakdown comparison for conventionally cooled and LHR engines, with the typical-CR, cast iron engine for reference. All cases are naturally aspirated.

The model also shows the negative effects of LHR engines observed in practice. There is little increase in load due to decreased volumetric efficiency offsetting reduced heat transfer loses, as confirmed by LHR experiments. It is noteworthy that the TBC insulation in the intake port and the conventionally-cooled liner is helping to reduce the volumetric efficiency penalty by reducing heat transfer *to* the incoming gas. It is significant that in this configuration peak surface temperatures (located on the piston and exhaust valve) exceeded 1100 K. This is a higher surface temperature than is observed in LHR research engines, but is well within the range of surface temperatures achieved in gas turbines [17]. As a result, either ceramics must be made to withstand these temperatures in reciprocating engines, or the engine configuration must be altered to decrease peak temperatures. While these temperatures present a significant obstacle, we will first examine means to offset the decrease in volumetric efficiency, specifically through intake charge compression, before addressing the need to lower peak cycle temperatures.

Turbomachinery

The results from the LHR engine configuration demonstrate the same requirement as previous LHR engine efforts-the increased exhaust exergy must be utilized or minimized if truly significant efficiency gains are to be realized through reduced heat transfer. The most obvious way to accomplish this is by implementing turbomachinery. The turbine in a turbocharger can use the enthalpy in the exhaust to compress the incoming charge, increasing the load and the efficiency (due to the higher effective compression ratio).[11]

Three potential LHR engines were modeled with a compressor (C) before the intake port, each with a different post-exhaust-port configuration. The first was a turbocharged engine; the backpressure in the exhaust port was set such that the turbine (T or T1) extracted enough work from the exhaust to compress the incoming charge (including losses). The second was turbo-compounded; the backpressure was held fixed while the exhaust was expanded in a second turbine (T2) to atmospheric pressure. Any excess work (beyond what was required by the compressor) was compounded to the shaft output (with a mechanical efficiency penalty). In this configuration, the exhaust port backpressure was set to maximize the total work.[12] The schematics of the engine configurations are shown in Fig. 6.

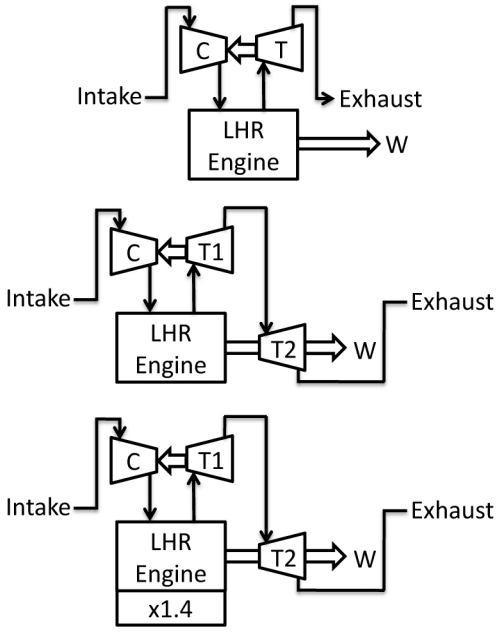


Figure 6. Model schematics for LHR engine configurations with turbo-machinery: turbocharged (top), turbo-compounded (middle), and over-expanded/ turbo-compounded (bottom).

[10] This assumes the product composition is relatively unchanged, and thus higher temperatures are required to reject additional entropy at ambient pressure.

[11] As used here, effective compression ratio is the ratio of TDC density to ambient density, as opposed to geometric compression ratio, which is the ratio of BDC volume to TDC volume.

[12] We assume that multiple turbomachinery stages could be implemented, where necessary, to prevent choking.

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