Energy 100 (2016) 115–128

Contents lists available at ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

Maximum-efficiency architectures for heat- and work-regenerative gas turbine engines

Sankaran Ramakrishnan*, Christopher F. Edwards

Dept. of Mechanical Engineering, Stanford University, Stanford, CA 94305, USA

ARTICLE INFO

Article history:
Received 17 June 2014
Received in revised form
11 January 2016
Accepted 12 January 2016
Available online xxx

Keywords:
Irreversibility minimization
Maximum efficiency
Regenerative gas turbine engines
Exergy

ABSTRACT

This study establishes maximum-efficiency architectures for heat- and work-regenerative gas turbine engines using a systematic irreversibility minimization approach. It considers engine architectures that employ two kinds of energy transfers: heat and work. It does not assume any cycle a priori (e.g., heat-recuperative reactive Brayton cycle). Instead, the maximum-efficiency architecture is directly deduced from first principles. Not surprisingly, the optimal architecture has some conventional features such as regenerative heat transfer from post-expansion combustion products to post-compression air, and external heat transfer out during compression (intercooling). But in addition it has three non-conventional features. First, unlike conventional heat recuperation heat is withdrawn between expansion turbine stages and transferred to post-compression air. Second, air is further compressed after heating. Third, compression is required to be part intercooled and part non-intercooled.

1. Introduction

Combustion engines have been developed based on a wide variety of simple-, regenerative-, and combined-cycle architectures that have been optimized for higher efficiency. All engine architectures can be understood as being sequences of three kinds of energy transfers—as work, as heat, and with matter—and the resulting mechanical, thermal, diffusive, and chemically-reactive equilibration processes. Each process in the sequence has a well-defined *process length*. For example, a reactive Brayton architecture is a sequence of work input, combustion (equilibration of fuel and air), and work output. The amount of work added in compression, amount of fuel burned, and the amount of work extracted in expansion are the respective lengths of these processes. Thus, optimization of engine architectures involves either varying the process sequence (creating a new engine cycle), or varying the process lengths (changing parameters of a cycle), or both.

Parametric optimization studies perform optimization of only the process lengths for any chosen engine cycle. For the Brayton cycle, a parametric study might involve optimization of the amount of compression work (pressure ratio), or the amount of fuel used (equivalence ratio) for maximum efficiency. However, the optimality of the underlying process sequence itself, i.e., compression, combustion, and expansion is not evaluated. For identifying the overall maximum-efficiency engine architecture, parametric analysis of pre-selected engine cycles is not sufficient. One must also identify the optimal process sequence allowed by physics and engineering constraints. Furthermore, in combustion-engine optimization the combustion process must be modeled accurately as a chemically-reactive process. It must not be treated as a heat-addition process, as is often done in heat-engine analyses. Combustion generates irreversibilities due to chemically-reactive, diffusive, thermal, and mechanical equilibration, whereas, heat addition has only thermal irreversibility [1,2]. This work is aimed at establishing the overall maximum-efficiency engine architecture for heat- and work-regenerative engines and includes identification of the optimal process sequence (the underlying cycle) and the optimal process lengths.

Conventionally, regenerative heat transfer is employed in engine cycles in two ways: internal-regenerative heat transfer from post-expansion gases to post-compression air, and external heat transfer to the environment between air compression stages (intercooling) which have been widely studied in research literature [3–7] and in thermodynamics textbooks [8]. In addition, an alternative approach to regenerative heat transfer was studied by Dellenback [9], Cardu and Biaca [10], and Cai and Jiang [11]. In this

* Corresponding author.
E-mail addresses: sankaranr@alumni.stanford.edu (S. Ramakrishnan), cfe@stanford.edu (C.F. Edwards).

http://dx.doi.org/10.1016/j.energy.2016.01.044
0360-5442/

approach, energy is transferred out as heat between two turbine stages, i.e., prior to complete work extraction, and supplied to post-compression air. Although intermediate removal of energy results in less work output from the downstream turbine, Dellenback shows that an overall higher efficiency can obtained. This is because such a heat transfer strategy raises the post-compression air to higher temperatures. However, in their studies Cardu and Biaca, and Cai and Jiang, show a corresponding decrease in air-specific work. The scope for efficiency increase is also shown to be limited due to practical limitations of heat exchanger effectiveness. In a re-assessed version of this approach Dellenback [12] shows that two stages of heat regeneration, between and after expansion stages, has a greater potential for increasing efficiency.

In this paper we consider all permissible ways to employ internal and external heat transfer using heat exchangers, work input/output using compressors and turbines, and combustion in burners to establish the architecture for heat- and work-regenerative gas turbine engines that has the highest efficiency over any existing or conceivable cycle. Our approach includes, but is not restricted to, traditional heat recuperation and intercooling. The paper employs attractor-trajectory optimization methodology previously developed by the authors [13] and is structured as follows. In Section 2 the thermodynamic model and methodology is presented. In Sections 3–6 the optimization is performed through four model problems. The optimal heat- and work-regenerative engine architecture obtained is summarized in Section 7 that concludes the paper.

2. Thermodynamic model and methodology

2.1. Optimization objective

The objective is to identify the engine architecture that has maximum exergy efficiency. Such an architecture minimizes total entropy generation inside and outside the engine [14].

$$\text{Max exergy efficiency} \Rightarrow \min \dot{S}_{gen}^{Total} = \min \left[\dot{S}_{gen}^{Engine} + \dot{S}_{gen}^{Environment}\right] \quad (1)$$

Irreversibility inside the engine includes compressor and turbine irreversibilities, heat-exchanger irreversibility, and combustion irreversibility. External irreversibility includes the exergy in the exhaust and in heat transferred out from intercoolers that is destroyed in the atmosphere.

2.2. System model

Engine architectures are modeled as sequences of energy transfers and energy transformations (equilibration processes) in devices. A unit mass of fuel and predetermined amount of air are supplied initially unmixed, and chemically equilibrated when mixed in part or as whole downstream. At the engine exit all streams are mixed, chemically equilibrated, and discharged as a single exhaust stream. The desired energy transfer and transformation processes considered in this study are listed in Table 1 along with their respective devices, device symbols, device inefficiencies, and device limitations.

Compressors C and turbines T are modeled as adiabatic but irreversible devices. Their irreversibility is quantified using polytropic efficiencies η_C, η_T and they are constrained by the maximum gas temperature limit $T_{gas,blade}^{max}$. Burners B are modeled as adiabatic and constant-pressure devices. The counter-flow heat exchanger, X, is considered adiabatic to the environment and as having negligible streamwise pressure drop, and is constrained by the maximum gas temperature for heat exchange $T_{gas,X}^{max}$. The unavoidable irreversibility in the heat exchanger is quantified using ΔT_X^{min}, the minimum temperature difference between hot and cold streams that is required to achieve heat transfer within a finite length, area, and for finite conductivity. The heat capacities of the hot and cold streams are often not equal, causing a temperature difference between streams that is greater than the minimum allowed temperature difference, i.e., $\Delta T_X \geq \Delta T_X^{min}$. Intercoolers I are heat exchangers employed for heat transfer to the environment and have similar device limitations and imperfections.

Table 1
List of permissible energy transfers and energy transformations and the respective devices.

Energy transfer or transformation	Device	Symbol	Device metrics
Work	Compressor	C	η_C, $T_{gas,blade}^{max}$
	Turbine	T	η_T, $T_{gas,blade}^{max}$
Heat	Heat exchanger	X	ΔT_X^{min}, $T_{gas,X}^{max}$
	Intercooler	I	ΔT_I^{min}, $T_{gas,I}^{max}$
Combustion	Burner	B	

The two most restrictive device limitations in this study are limits on gas temperature in turbines ($T_{gas,blade}^{max}$) and heat-exchangers ($T_{gas,X}^{max}$). The latter is more stringent since heat-exchanger surfaces, unlike turbine blades, must perform heat transfer, therefore cannot have thermal-barrier coatings. Pressure drop in burners and heat exchangers, and blade-cooling requirements in turbines have not been considered as these issues are highly specific to detailed engine design. We believe, it is more useful to quantify the effect of these losses as corrections to the theoretical maximum efficiency obtained in this study.

2.3. Optimization methodology

To maximize efficiency over all permissible cycles and process lengths, the optimization methodology has two logical steps: i) Systematic evolution of a base process sequence such that all permutations of the allowed heat and work processes are considered, followed by ii) deduction of the optimal process sequence from the base sequence by optimizing process lengths. When the length for any process is optimized one finds that, if non-optimal, the process has an optimal length of zero, i.e., it is eliminated from the base sequence. Thus, by sequential elimination the base sequence is reduced to the optimal sequence while simultaneously establishing the optimal length of the processes that remain in it.

The starting process sequence for this study is the optimal simple-cycle (only work-regenerative) architecture. This is an apt starting sequence because the optimal heat- and work-regenerative architecture to be established must reduce to the optimal work-regenerative configuration if heat transfer is removed from consideration, i.e., in the limit of no (zero-length) heat transfer. In previous work done by the authors the optimal work-regenerative architecture was established to be $CB(TB)_nT$ [15]. This sequence is depicted in Fig. 1. The work-regenerative process sequence is split into three segments: pre-combustion, combustion, and post-combustion as shown below

$$\underbrace{C}_{\text{Pre-combustion}} \quad \underbrace{B(TB)_n}_{\text{Combustion}} \quad \underbrace{T}_{\text{Post-combustion}}$$

Next, one or more stages of heat transfer is introduced into this sequence in the following ways: