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Application of Finite-Time Thermodynamics to

Solar Power Conversion

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ABSTRACT

Thermal conversion of solar power depends on a heat engine operating between the receiver and sink temperatures. The principle of operation of a solar-thermal power plant is presented in terms of finite heat transfer rates and an internally reversible heat engine. The theory of finite-time thermodynamics and maximum power is utilized to determine the upper limit to power output for terrestrial and space-based solar-thermal engines. Parametric equations are derived for determining optimum performance variables. In terrestrial systems an optimum receiver temperature is determined for maximum power output; whereas for space-based systems, maximum power is achievable at any receiver temperature.

1. INTRODUCTION

Classical thermodynamics shows how the transfer of thermal energy may be converted into mechanical work by a heat engine operating between two thermal reservoirs of temperatures T_{ih} and T_{ic} , with

$T_{ih} > T_{ic}$, and how heat transfer at infinitesimal temperature differences between the reservoirs and the heat engine lead to a reversible process and the well-known Carnot efficiency as the upper limit to conversion of heat to mechanical work [e.g., Reynolds and Perkins 1977]:

$$\eta_c \equiv \frac{W}{Q_h} = 1 - \frac{T_{ic}}{T_{ih}} \quad (1.1)$$

The requirement of heat transfer at vanishing temperature differences implies an infinitely slow rate of heat transfer, or an infinite heat transfer area, both of which are unacceptable in practical engines. In essence, we seek not from Nature to produce work, but to produce power, or work in a finite period of time. Thus, all thermal power plants utilize thermodynamics in converting rates of heat transfer at finite temperature differences, \dot{Q}_h , to mechanical power, \dot{W} , but without ever closely approximating the theoretical upper limit, usually written as the rate ratio

$$\eta_c = \dot{W}/\dot{Q}_h$$

(In fact, most power plants are designed for maximum power output, which tends to minimize the all-important capital costs of the plant, rather than for maximum thermal efficiency, which would "only" save on fuel costs; this becomes especially important for solar power plants because in this application the capital costs far outweigh operation and maintenance costs, the "fuel" being "free").

This discrepancy led Curzon and Ahlborn [1975] to consider a heat engine whose only irreversibility was finite-time heat transfers across finite temperature differences between the reservoirs and the heat engine:

$$\dot{Q}_h = K_h (T_h - T_{ih}) , \quad \dot{Q}_c = K_c (T_{ic} - T_c) \quad (1.2,3)$$

where now T_{ih} and T_{ic} are the hot and cold temperatures internal

to the engine (i.e., the working fluid temperatures), and T_h and T_c are the reservoir temperatures "external" to the engine. The remarkable result was that with T_{ih} and T_{ic} optimized for maximum power output, and with the internally reversible conversion cycle (1.1), the overall conversion efficiency was independent of K_h and K_c , and dependent only on the reservoir temperature ratio, as previously:

$$\eta_e \equiv \frac{\dot{W}_{\max}}{\dot{Q}_h} = 1 - \sqrt{T_c/T_h} \quad (1.4)$$

This cycle is here referred to as an endoreversible cycle [Callen 1985] and is utilized subsequently to predict maximum possible solar power conversion. It was indicated that practical power plants can have efficiencies in close agreement with (1.4), but not with (1.1) [Curzon and Ahlborn 1975; Callen 1985]. The derivation of (1.4) required the processes, (1.2) and (1.3) to be isothermal, even though finite. Subsequent analyses have shown these processes to indeed yield maximum power [Salamon & Nitzan, 1981]; have been applied to the sequences of combustion engines [Mozurkewich and Berry 1982; Band, Kafri, and Salamon 1982ab; Andresen, Salamon, and Berry 1984]; to fluctuating external temperatures [Salamon, Band, and Kafri 1982]; and recently to refrigeration cycles [Chen and Yan 1988; Yan and Chen 1989]. The result (1.4) was also derived for a continuous, steady-flow cycle which is more typical of stationary power plants [Bejan 1988]; in that analysis, which also allowed for a leakage heat flow, a further maximization of the power output yielded equal values for the optimum hot-side and cold-side conductances: $K_h = K_c = K$.

In conventional power plants, T_h in (1.4) is usually a metallurgical upper limit for the material transferring heat to the working fluid. As such, it is a constant that is maintained by combustion (or other processes) regardless of the magnitude of \dot{Q}_h . In solar plants, by contrast, T_h represents the temperature of the receiver, which varies according to the magnitude of \dot{Q}_h . Therefore, the highest permissible T_h does not lead to the

maximum conversion rate of available energy to mechanical power as in conventional plants, as has been shown for the Carnot cycle [Reynolds and Perkins 1977, p. 236; Howell and Bannerot 1977; Bejan 1987a], and for the endoreversible cycle [Gordon 1988]. In following sections, these methods are utilized in deriving formulae for the optimum temperature ratio T_h/T_c for greatest power conversion. Conversion cycles with diathermal hot-side and cold-side heat exchange processes have also been considered [Bejan, Kearney, and Kreith 1981; Wu 1988]. Here the objective is to provide upper bounds for power output so that (1.4) applies; this equation is approachable in real systems where there is phase change or only small temperature variations in the heat exchangers.

The low-temperature reservoir usually has T_c as the environment temperature to which any heat rate \dot{Q}_c can be rejected. This is so for both conventional thermal plants and terrestrial solar plants. However, for space-based solar-thermal power plants under current development the low temperature "reservoir" at T_c is the space radiator where \dot{Q}_c depends on T_c , thus requiring special treatment.

In what follows, a typical solar-thermal power conversion system is described and characterized, and investigated for maximum possible endoreversible power conversion, for both terrestrial and space applications. This leads to the best receiver and radiator temperatures, and system efficiencies, for various types of designs.

2. DESCRIPTION OF SOLAR HEAT ENGINE

The principle of operation of a solar-thermal power plant is shown schematically in Fig. 1, where the solar irradiance, \dot{q}_s (W/m^2), is incident on projected or aperture area, A_p , of the reflector, or concentrator. In general, \dot{q}_s varies with time (diurnally and annually as well as due to weather patterns) so that various mechanisms are required for the concentrator to track \dot{q}_s [e.g., Duffie and Beckman 1980; Kreith and Kreider

1978]. This time variation is rather slow compared to engine processes; thus, \dot{q}_s may be taken as a constant for present purposes.

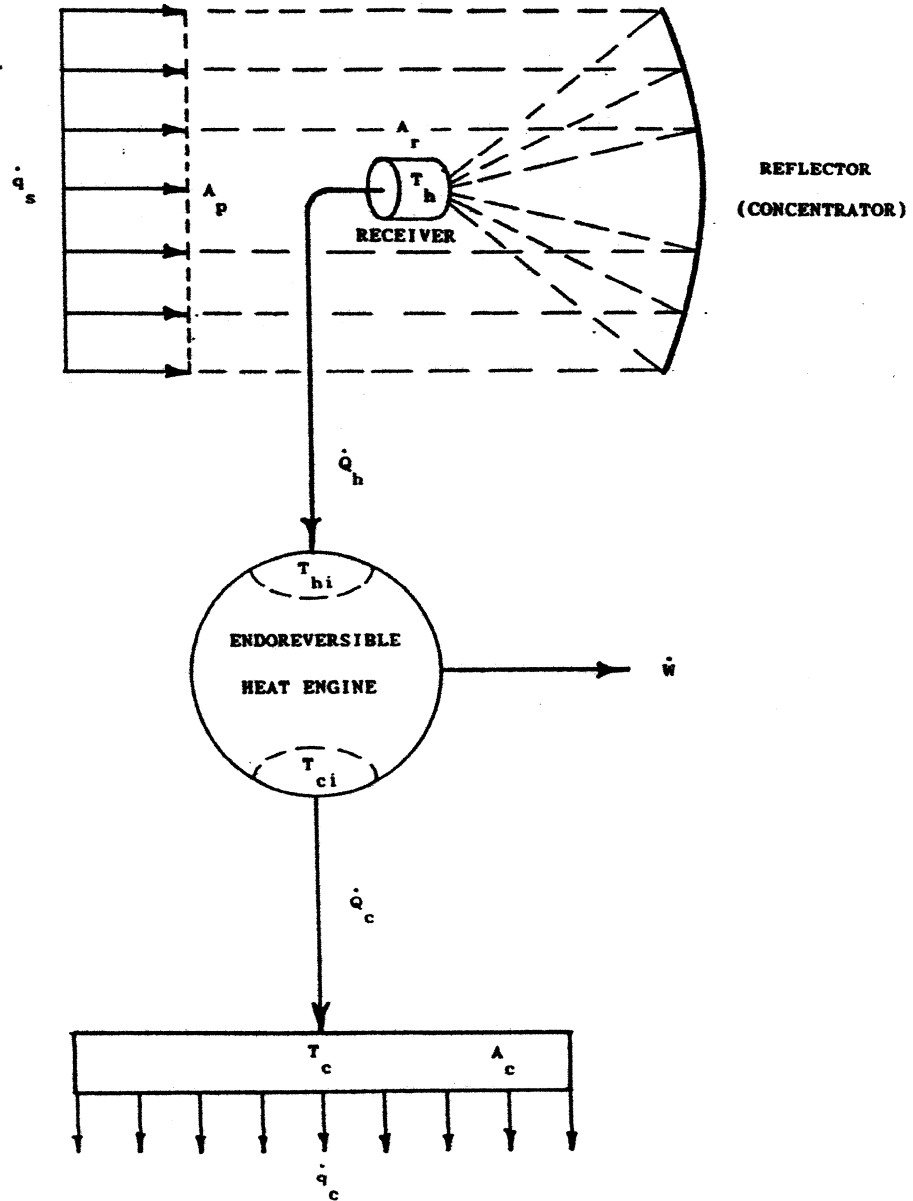


Figure 1. Conceptual Solar-Thermal Power Plant.

Although the source of \dot{q}_s is the sun, and thermodynamic analyses have been applied with T_{sun} as the source temperature [e.g., Bejan 1987b], and conversion of radiant energy has been considered [De Vos 1988], here we take the more pragmatic, or

