FINITE TIME THERMODYNAMIC ANALYSIS AND OPTIMIZATION OF SOLAR-DISH STIRLING HEAT ENGINE WITH REGENERATIVE LOSSES

by

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The present study investigates the performance of the solar-driven Stirling engine system to maximize the power output and thermal efficiency using the non-linearized heat loss model of the solar dish collector and the irreversible cycle model of the Stirling engine. Finite time thermodynamic analysis has been done for combined system to calculate the finite-rate heat transfer, internal heat losses in the regenerator, conductive thermal bridging losses, and finite regeneration process time. The results indicate that exergy efficiency of dish system increases as the effectiveness of regenerator increases but decreases with increase in regenerative time coefficient. It is also found that optimal range of collector temperature and corresponding concentrating ratio are 1000 K~1400 K and 1100~1400, respectively, in order to get maximum value of exergy efficiency. It is reported that the exergy efficiency of this dish system can reach the maximum value when operating temperature and concentrating ratio are 1150 K and 1300, respectively.

Key words: *solar parabolic dish collector, solar-driven Stirling engine, finite-rate heat transfer, exergy efficiency of dish system*

Introduction

Electricity generation by solar power plants has gained importance in recent years. Solar thermal power systems utilize the heat generated by a collector concentrating and absorbing the Sun's energy to drive a heat engine/generator and produce electric power. Currently, three concepts are well known and established: Parabolic trough power plants, solar tower power plants, and dish-Stirling systems. Out of these three the dish-Stirling system has demonstrated the highest efficiency [1, 2]. Over the last 20 years, eight different dish-Stirling systems ranging in size from 2 to 50 kW have been built by companies in the United States, Germany, Japan, and Russia [1]. The Stirling energy system (SES) dish has held the world's efficiency record for converting solar energy into grid-quality electricity, and

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in January 2008, it achieved a new record of 31.25% efficiency rate. Dish-Stirling systems are flexible in terms of size and scale of deployment. Owing to their modular design, they are capable of both small-scale distributed power output, and suitable for large, utility-scale projects with thousands of dishes arranged in a solar park (two plants in the US totaling over 1.4 GW are slated to begin construction in 2010 using the SES technology). In principle, high concentrating and low or non-concentrating solar collectors can all be used to power the Stirling engine. Finite time thermodynamics/finite temperature difference thermodynamics deals with the fact that there must be a finite temperature difference between the working fluid/substance and the source/sink heat reservoirs (with which it is in contact) in order to transfer a finite amount of heat in finite time.

The literature of finite time thermodynamics began with the novel work of Curzon *et al.* [3], who established a theoretical model of a real Carnot heat engine at maximum power output with a different efficiency expression than the well known Carnot efficiency*. In recent years finite-time thermodynamics has been used successfully to study the performance analysis and optimization of low temperature differential Stirling heat engines [5, 6] powered by low concentrating solar collectors [7-9]. In addition, finite-time thermodynamics analysis of heat engines is usually restricted to systems having either linear heat transfer law dependence to the temperature differential both the reservoirs and engine working fluids [10, 11]. However, for higher temperature solar-powered heat engines, radiation and convection modes of heat transfer are often coupled and play a collective role in the processes of engines. Sahin [2] investigated the optimum operating conditions of endoreversible heat engines with radiation and convection heat transfer between the heat source and working fluid as well as convection heat transfer between the heat sink and the working fluid based on simultaneous processes. During the simultaneous processes, used as the steady-state operation in literature, the heat addition and heat rejection processes are assumed to take place simultaneously and are continuous in time as in a thermal power plant. The power output with simultaneous processes is given by $\dot{W} = \dot{Q}_{\text{in}} - \dot{Q}_{\text{out}}$.

In this paper, a general analysis of finite time thermodynamics of a solar dish-Stirling engine has been presented considering both convective and radiative heat transfer phenomena between heat source/sink and working fluid. Optimization has been done by varying the source temperature and concentration ratio. We have obtained the expression of maximum power output and computed the corresponding thermal efficiency. The influence of major parameters like heat leak coefficient, ratio of volume during regeneration processes, regenerator losses etc on the maximum power output and the corresponding overall efficiency is analyzed in detail. The aim of this article is to provide the basis for the design of a solar-powered high temperature differential Stirling engine operated with a high concentrating collector.

System description

As indicated in fig. 1, dish-engine systems use a mirror array to reflect and concentrate incoming direct normal insolation to a receiver, in order to achieve the temperatures required to efficiently convert heat to work. This requires that the dish track the sun in two axes. The concentrated solar radiation is absorbed by the receiver (absorber) and transferred to an engine.

* In 1977, Howel *et al.* analysed the Carnot cycle to determine the oprimum value of outlet temperature of solar collector with respect to maximum cycle work output [4].

An endoreversible Stirling heat engine coupled with a heat source and a heat sink, and with a regenerator and conductive thermal bridging losses from absorber to heat sink is depicted in fig. 2 along with its *T-S* diagram in fig. 3. The cycle approximates the compression stroke of a real Stirling heat engine as an isothermal heat rejection process (1-2) to the low temperature sink. The heat addition to the working fluid from the regenerator is modeled as the constant volume process (2-3). The expansion stroke producing work is modeled as isothermal heat addition process (3-4) from a high temperature heat source. Finally the heat rejection to the regenerator is modeled as the constant volume process (4-1). If the regenerator is ideal, the heat absorbed during process 4-1 should be equal to the heat rejected during process 2-3, however, the ideal regenerator requires an infinite area or infinite regeneration time to transfer finite heat amount, and this is impractical. Therefore, it is desirable to consider a real regenerator with heat losses ΔQ_R. In addition, we also consider conductive thermal bridging losses Q_b from the absorber to the heat sink.

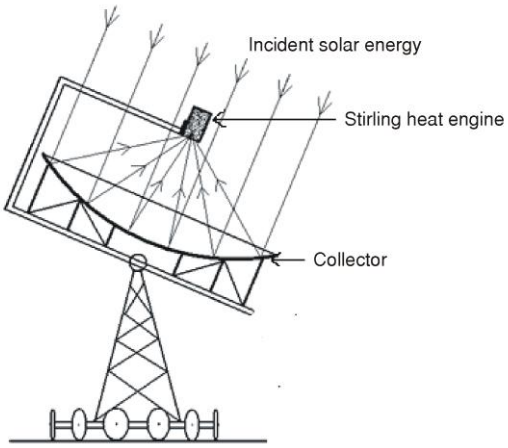

Figure 1. Schematic diagram of the dish system

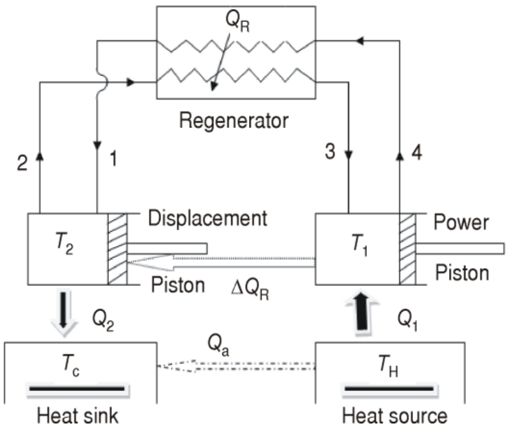

Figure 2. Schematic diagram of the Stirling heat engine cycle

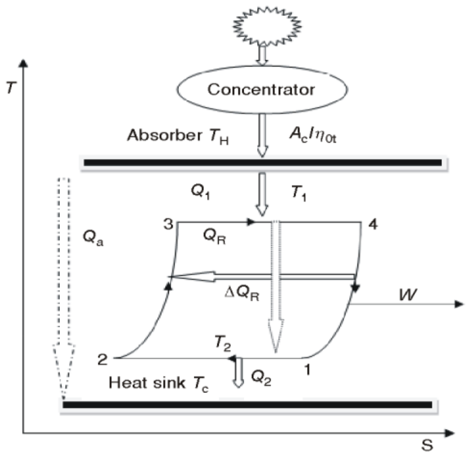

Figure 3. *T-S* diagram of solar dish-Stirling heat engine cycle

Finite time thermodynamics analysis

The analysis of the article includes the mathematical models for the dish solar collector, the Stirling engine as well as the combination of the dish solar collector and the Stirling engine.

Thermal efficiency of the dish solar collector

Actual useful heat gain q_u of the dish collector, considering conduction, convection, and radiation losses is given by [12]:

$$q_u = IA_c\eta_0 - A_H[h(T_H - T_a) + \varepsilon\sigma(T_H^4 - T_a^4)] \tag{1}$$

where I is the direct solar flux intensity, A_c – the collector aperture area, η_0 – the collector optical efficiency, A_H – the absorber area, h – the conduction/convection coefficient, T_H – the absorber temperature, T_a – the ambient temperature, ε – emissivity factor of the collector, and σ – the Stefan's constant.

Thermal efficiency of the dish collector is defined by η_d:

$$\eta_d = \frac{Q_u}{IA_c} = \eta_0 - \frac{h(T_H - T_a) + \varepsilon\sigma(T_H^4 - T_a^4)}{IC} \tag{2}$$

Finite-time thermodynamics analysis of the Stirling heat engine

Regenerative heat loss to the regenerator

It should be pointed out that also the regenerative branches are affected by internal thermal resistances to and from the thermal regenerator. Thus, regenerative losses are inevitable. One may quantify these regenerative losses by [4]:

$$\Delta Q_R = nC_v\,(1 - \varepsilon_R)\,(T_1 - T_2) \tag{3}$$

where, C_v is the heat capacity of the working substance, n – the number of mole partaking in the regenerative branches, ε_R – the effectiveness of the regeneration and T_1 and T_2 are temperatures of the working fluid in the high temperature isothermal process 3-4 and in the low temperature isothermal process 1-2, respectively. When $\varepsilon_R = 1$ the Stirling cycle operators with ideal (complete) regeneration.

To achieve a more realistic case, the time of the regenerative heat transfer processes should also be considered in the thermodynamic analysis of a dish-Stirling heat engine [2]. For this purpose, it is assumed that the temperature of the engine working fluid/substance is varying with time in the regenerative processes as given by [13]:

$$\frac{dT}{dt} = \pm\alpha \tag{4}$$

where α is the proportionality constant which is independent of the temperature difference and dependent only on the property of the regenerative material, called regenerative time constant and the ± sign belong to the heating and cooling processes, respectively. The time (t_r) of two constant volume regenerative processes is given by:

$$t_R = t_3 + t_4 = 2\alpha(T_1 - T_2) \tag{5}$$

Heat transfer across the Stirling cycle

Heat supplied to the working fluid at temperature T_1 (Q_1) and the heat released by the working fluid at temperature T_2 (Q_2), during the two isothermal processes are:

$$Q_1 = Q_h + \Delta Q_R = nRT_1 \ln\frac{V_1}{V_2} + nC_v(1-\varepsilon_R)(T_1 - T_2) \tag{6}$$

and

$$Q_2 = Q_c + \Delta Q_R = nRT_2 \ln\frac{V_1}{V_2} + nC_v(1-\varepsilon_R)(T_1 - T_2) \tag{7}$$

respectively, where n is the mole number of the working substance, R – the universal gas constant, T_1 and T_2 are the temperatures of the working substance during the high and low temperature isothermal branches, and V_1 and V_2 – the volumes of the working substance along the constant-volume heating and cooling branches, as shown in fig. 3.

Invariably, there are thermal resistances between the working substance and the external heat reservoirs in the dish-Stirling engine. In order to obtain a certain power output, the temperatures of the working substance must therefore be different from those of the heat reservoirs. When convective and radiative heat transfer mode is consider between absorber (source/sink) and working fluid**,** heat transfer can be written as:

$$Q_1 = [h_H(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)]t_1 \tag{8}$$

and

$$Q_2 = h_C(T_2 - T_C)t_2 \tag{9}$$

At sink side only convective mode is predominant. Here h_H and h_C are the thermal conductances between the working substance and the heat reservoirs at temperatures T_H and T_C, h_{HR} is the high temperature side radiative heat transfer coefficient, and t_1 and t_2 are the times spent on the two isothermal branches at temperatures T_1 and T_2, respectively.

The conductive thermal bridging losses from the absorber at temperature T_H to the heat sink at temperature T_C is assumed to be proportional to the cycle time and given by [13, 14]:

$$Q_a = k_a(T_H - T_C)t \tag{10}$$

where k_a is the heat leak coefficient between the absorber and the heat sink and t – the cyclic period.

Taking in account the major irreversibility mentioned above, the net heats released from the absorber Q_H and absorbed by the heat sink Q_C are given as:

$$Q_H = Q_1 + Q_a \tag{11}$$

$$Q_C = Q_2 + Q_a \tag{12}$$

Thus the total cycle time is given by:

$$t = t_1 + t_2 + t_3 + t_4 = \frac{nRT_2 \ln\lambda + nC_v(1-\varepsilon_R)(T_1 - T_2)}{h_H(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)} + \\ + \frac{nRT_2 \ln\lambda + nC_v(1-\varepsilon_R)(T_1 - T_2)}{h_C(T_2 - T_L)} + \frac{T_1 - T_2}{2\alpha} \tag{13}$$

For the thermodynamic cycle 1-2-3-4-1, work, power output, and thermal efficiency is given by [15, 16]:

$$W = Q_H - Q_C \tag{14}$$

$$P = \frac{W}{t} = \frac{Q_H - Q_C}{T} \tag{15}$$

Using eqs. (8)-(15), we have:

$$P = \frac{T_1 - T_2}{\dfrac{T_1 + Y_1(T_1 - T_2)}{h_H(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)} + \dfrac{T_2 + Y_1(T_1 - T_2)}{h_C(T_2 - T_C)} + Y_2(T_1 - T_2)} \tag{16}$$

$$\eta_s = \frac{T_1 - T_2}{T_1 + Y_1(T_1 - T_2) + k_a(T_H - T_C)\left[\dfrac{T_1 + Y_1(T_1 - T_2)}{h_H(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)} + \dfrac{T_2 + Y_1(T_1 - T_2)}{h_C(T_2 - T_C)} + Y_2(T_1 - T_2)\right]} \tag{17}$$

where

$$Y_1 = \frac{C_v(1 - \varepsilon_R)}{\mathrm{R}\ln\lambda} \quad \text{and}$$

$$Y_2 = \frac{2}{\alpha n \mathrm{R}\ln\alpha}$$

where $\lambda = V_1/V_2$.

For the sake of convenience, a new parameter $x = T_2/T_1$ is introduced into eqs. (16) and (17), then we have:

$$P = \frac{T_1 - xT_1}{\dfrac{T_1 + Y_1(T_1 - xT_1)}{h_H(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)} + \dfrac{xT_1 + Y_1(T_1 - xT_1)}{h_C(xT_1 - T_C)} + Y_2(T_1 - xT_1)} \tag{18}$$

To maximize the power output, take the derivative of eq. (18) with respect to the temperature T_1 and x and equate it to zero, namely $\partial P/\partial T_1 = 0$ and $\partial P/\partial x = 0$, the optimal working fluid temperature T_{1opt} and x_{opt} for this condition can be obtained from eq. (19) and (20), respectively:

$$\beta_1 T_{1opt}^8 + \beta_2 T_{1opt}^5 + \beta_3 T_{1opt}^4 + \beta_4 T_{1opt}^3 + \beta_5 T_{1opt}^2 + \beta_6 T_{1opt} + \beta_7 = 0 \tag{19}$$

$$E_1 x^2 + E_2 x + E_3 = 0 \tag{20}$$

where $\beta_1 = h_{HR}^2 B_1 x$, $\beta_2 = h_{HR}x(2B_1h_{HC} - 3B_2h_{LC}x)$, $\beta_3 = 2h_{HR}x(3B_2h_{LC}T_L - B_1B_3)$, $\beta_4 = -3B_2h_{HR}h_{LC}T_L^2$, $\beta_5 = h_{HC}x(B_1h_{HC} - B_2h_{LC}x)$, $\beta_6 = 2h_{HC}x(B_2h_{LC}T_L - B_3B_1)$, $\beta_7 = B_1 x B_3^2 - B_2 h_{HC} h_{LC} T_L^2$, $B_1 = x + A_1(1 - x)$, $B_2 = 1 + A_1(1 - x)$, $B_3 = h_{HC}T_H + h_{HR}T_H^4$, $E_1 = B(T_1^2 - Y_1T_1T_Ch_C)$, $E_2 = B(Y_1h_CT_1^2 - Y_1h_CT_1T_C) + T_1^2h_C$, $E_3 = -[Bh_CT_1(T_L + Y_1T_1)] + T_1^2h_C$, $B = h_H(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)$

Therefore, the maximum power output and the corresponding optimal thermal efficiency of the Stirling engine are:

$$P_{\max.} = \frac{1-x}{\dfrac{1+Y_1(1-x)}{h_\mathrm{H}(T_\mathrm{H}-T_\mathrm{1opt})+h_\mathrm{HR}(T_\mathrm{H}^4-T_\mathrm{1opt}^4)}+\dfrac{x+Y_1(1-x)}{h_\mathrm{C}(xT_\mathrm{1opt}-T_\mathrm{C})}+Y_2(1-x)} \tag{21}$$

$$\eta_\mathrm{sopt} = \frac{1-x}{1+Y_1(1-x)+k_\mathrm{a}(T_\mathrm{H}-T_\mathrm{C})\left[\begin{array}{l}\dfrac{1+Y_1(1-x)}{h_\mathrm{H}(T_\mathrm{H}-T_\mathrm{1opt})+h_\mathrm{HR}(T_\mathrm{H}^4-T_\mathrm{1opt}^4)}+\\ +\dfrac{x+Y_1(1-x)}{h_\mathrm{C}(xT_\mathrm{1opt}-T_\mathrm{C})}+Y_2(1-x)\end{array}\right]} \tag{22}$$

Special cases:

(1) When $h_\mathrm{HR} = 0$, *i. e.* only convection heat transfer is considered and radiation heat transfer is neglected between the absorber and the working fluid, eq. (19) is simplified to:

$$\beta_5 T_\mathrm{1opt}T^2 + \beta_6 T_\mathrm{1opt} + D_7 = 0$$

By solving eq. (19) we get:

$$T_\mathrm{1opt} = \frac{T_\mathrm{C}+F_1T_\mathrm{H}}{x+F_1}$$

where

$$F_1 = \frac{x^2+Y_1x(1-x)}{[1+Y_1(1-x)]^{0.5}}$$

(2) When $\varepsilon_\mathrm{R} = 1$, the Stirling engine achieves the condition of perfect/ideal regeneration, although the time of regeneration process is still considered. Then maximum power output and thermal efficiency is given by:

$$P_{\max.} = \frac{1-x}{\dfrac{1}{h_\mathrm{H}(T_\mathrm{H}-T_\mathrm{1opt})+h_\mathrm{HR}(T_\mathrm{H}^4-T_\mathrm{1opt}^4)}+\dfrac{x}{h_\mathrm{C}(xT_\mathrm{1opt}-T_\mathrm{C})}+Y_2(1-x)}$$

$$\eta_\mathrm{sopt} = \frac{1-x}{1+k_a(T_H-T_C)\left[\dfrac{1}{h_\mathrm{H}(T_\mathrm{H}-T_\mathrm{1opt})+h_\mathrm{HR}(T_\mathrm{H}^4-T_\mathrm{1opt}^4)}+\dfrac{x}{h_\mathrm{C}(xT_\mathrm{1opt}-T_\mathrm{C})}+Y_2(1-x)\right]}$$

(3) When $h_\mathrm{HR} = 0$, $\varepsilon_\mathrm{R} = 1$, and considering $x = (T_\mathrm{H}/T_\mathrm{C})^{1/2}$, then maximum power and thermal efficiency can be given as:

$$P_{\max.} = \frac{F_2(\sqrt{T_\mathrm{H}}-\sqrt{T_\mathrm{C}})^2}{1+F_2Y_2(\sqrt{T_\mathrm{H}}-\sqrt{T_\mathrm{C}})^2}$$

$$\eta_{\text{sopt}} = 1 - \sqrt{\frac{T_{\text{H}}}{T_{\text{C}}}}$$

However, physically for finite time regenerative time ε_{R} should be less than unity. This shows that in the investigation of the Stirling heat engine, it would be impossible to obtain new conclusions if the regenerative losses were not considered.

(4) When the time of regenerative processes is directly proportional to the mean time of two isothermal processes, *i. e.*

$$t_{\text{r}} = \gamma(t_1 + t_2)$$

where γ is the proportionality constant then

$$P_{\text{max.}} = \frac{1-x}{(1+\gamma)\left[\dfrac{1+Y_1(1-x)}{h_{\text{H}}(T_{\text{H}} - T_{\text{1opt}}) + h_{\text{HR}}(T_{\text{H}}^4 - T_{\text{1opt}}^4)} + \dfrac{x+Y_1(1-x)}{h_{\text{C}}(T_2 - T_{\text{C}})}\right]}$$

$$\eta_{\text{sopt}} = \frac{1-x}{1+Y_1(1-x)+k_{\text{a}}(T_{\text{H}} - T_{\text{C}})\left[\dfrac{1+Y_1(1-x)}{h_{\text{H}}(T_{\text{H}} - T_1) + h_{\text{HR}}(T_{\text{H}}^4 - T_{\text{1opt}}^4)} + \dfrac{x+Y_1(1-x)}{h_{\text{C}}(T_2 - T_{\text{C}})}\right]}$$

(5) When the regenerative time is zero, *i. e.* $t_{\text{r}} = 0$, the maximum power output is given by:

$$P_{\text{max.}} = \frac{1-x}{\dfrac{1+Y_1(1-x)}{h_{\text{H}}(T_{\text{H}} - T_{\text{1opt}}) + h_{\text{HR}}(T_{\text{H}}^4 - T_{\text{1opt}}^4)} + \dfrac{x+Y_1(1-x)}{h_{\text{C}}(T_2 - T_{\text{C}})}}$$

The maximum power and the corresponding thermal efficiency of the system

The maximum power and the corresponding thermal efficiency of the system is product of the thermal efficiency of the collector and the optimal thermal efficiency of the Stirling engine [13]. Namely:

$$\eta_{\text{Ov}} = \eta_{\text{d}}\eta_{\text{sopt}} \tag{23}$$

Using eqs. (2) and(22), we get:

$$\eta_{\text{Ov}} = \left[\eta_0 - \frac{h(T_H - T_a) + \varepsilon\sigma(T_{\text{H}}^4 - T_{\text{a}}^4)}{IC}\right]\cdot$$

$$\cdot\frac{1-x}{1+Y_1(1-x)+k_{\text{a}}(T_{\text{H}} - T_{\text{C}})\left[\begin{array}{l}\dfrac{1+Y_1(1-x)}{h_{\text{H}}(T_{\text{H}} - T_{\text{1opt}}) + h_{\text{HR}}(T_{\text{H}}^4 - T_{\text{1opt}}^4)} + \\ + \dfrac{x+Y_1(1-x)}{h_C(xT_{\text{1opt}} - T_{\text{C}})} + Y_2(1-x)\end{array}\right]} \tag{24}$$

Numerical results and discussions

In order to evaluate the effect of the absorber temperature (T_H), the concentrating ratio (C), the effectiveness of the regenerator (ε_R), the heat leak coefficient (k_0), heat transfer coefficients and volume ratio (λ) on the solar-powered dish-Stirling heat engine system, all the other parameters will be kept constant as n = 10 mol, R = 4.3 J/molK, C_v = 15 J/molK, ε = = 0.92, T_0 = 300 K, h = 20 W/m^{2}K, σ = 5.67·10^{-8} W/m^{2}K^4, α = 1000 K/s, and I = 1000 W/m^2. The results obtained are as follows.

Effect on thermal efficiency of solar dish

The effect of the absorber temperature T_H and the concentrating ratio C on thermal efficiency of the collector is shown in fig. 4.

From fig. 4, one can observe that the thermal efficiency of the collector decreases rapidly with increasing of the absorber temperature T_H, increases with the increasing of concentration ratio C. This is predominantly due to increase in convective and radiative heat losses at higher absorber temperature. The maximum thermal efficiency is limited by the optical efficiency of the concentrator.

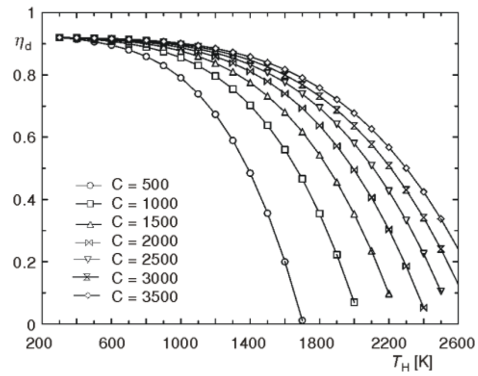

Figure 4. Variation of the thermal efficiency of the collector for different receiver temperature and the concentrating ratio

Effect on Stirling engine

Effect of absorber temperature (T_H)

Figures 5-7 shows the variation on work output (W), maximum power output (P_{max}) and thermal efficiency of Sterling engine (η_s) with respect to absorber temperature (T_H). Work output and maximum power output both increase with increase in absorber temperature. Thus it is desirable to have high temperature heat source to obtain higher power and work output.

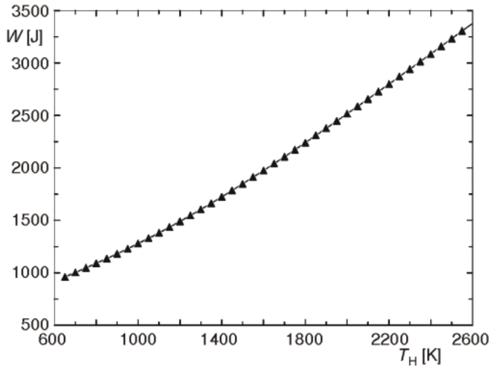

Figure 5. Variation on work output of the Stirling engine with respect to absorber temperature

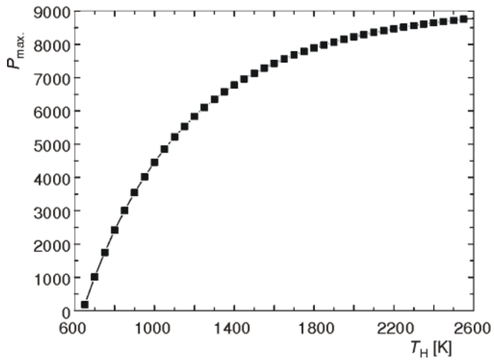

Figure 6. Variation on maximum power output of the Stirling engine with respect to absorber temperature

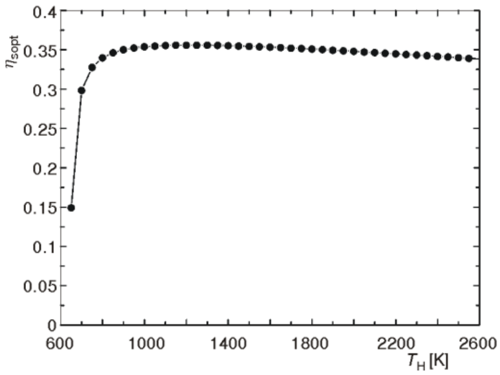

Figure 7. Variation on thermal efficiency of the Stirling engine with respect to absorber temperature

Optical thermal efficiency of Stirling engine increases rapidly at the beginning and decrease slowly afterwards with the increase of absorber temperature. The optimum range of absorber temperature is 1150-1300 K where efficiency reaches at its maximum value. The reason for the decrease is conductive thermal bridging losses from the absorber to the heat sink whose effects are more pronounced at higher absorber temperature.

Effect of heat sink temperature (T_C)

It is seen from figs. 8-10 that as the heat sink temperature increases, work output decreases whereas maximum power output and thermal efficiency of engine increases. The decrease in work output is due to increase in heat transfer at lower temperature of cycle. Power output increase due to decrease in thermal bridging losses (Q_a). The effect of T_C is more pronounced for maximum power output and less pronounced for heat input (Q_1) to the heat engine.

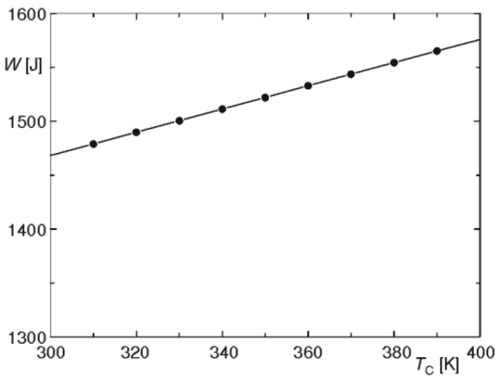

Figure 8. Variation on work output of the Stirling engine with respect to heat sink temperature

Effect of effectiveness of regenerator (ε_R)

As the regenerative effectiveness increases, the heat transfers (Q_1 and Q_2) decreases but the regenerative heat transfer

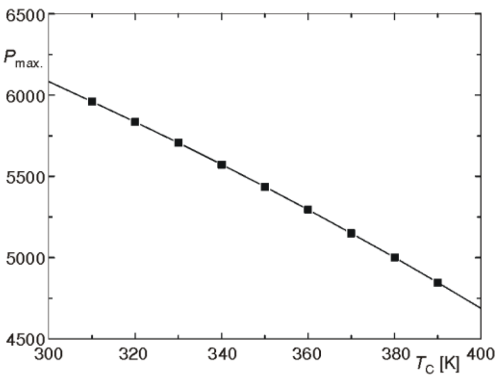

Figure 9. Variation on maximum power output of the Stirling engine with respect to heat sink temperature

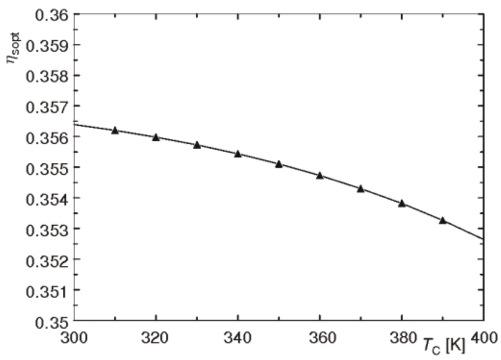

Figure 10. Variation on thermal efficiency of the Stirling engine with respect to heat sink temperature

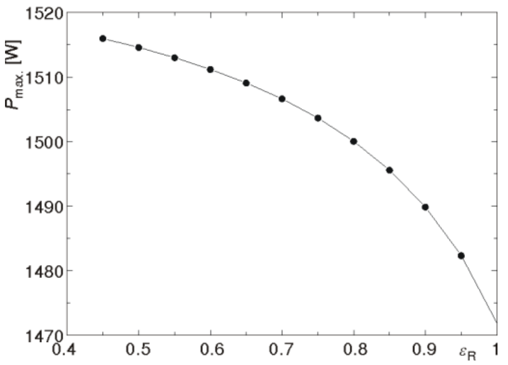

Figure 11. Variation on work output of Stirling engine with respect to regenerator effectiveness

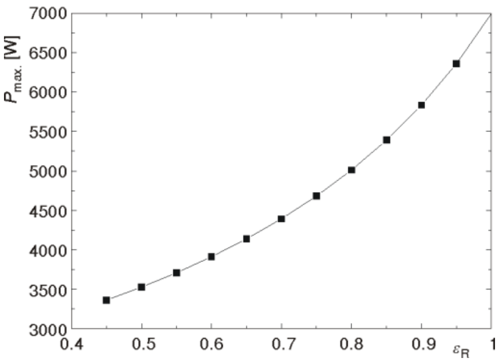

Figure 12. Variation on maximum power output of Stirling engine with respect to regenerator effectiveness

(Q_R) increases. It can be seen from figs. 11-13 that work output decreases with effectiveness while maximum power output increases. The decrease in work output is due to decrease in optimum temperature at heat addition (T_{1opt}) with increase in effectiveness of regenerator. It is also found that the optimal thermal efficiency of the Stirling engine increases with the increasing of the effectiveness of the regenerator and is influenced greatly by it.

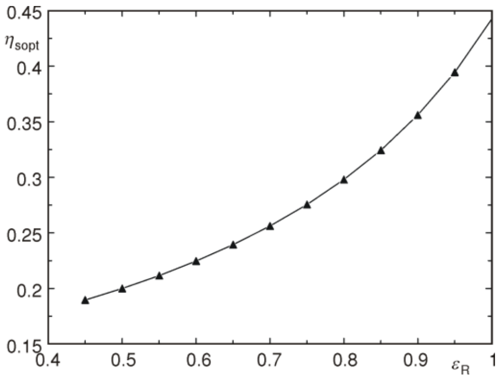

Figure 13. Variation on maximum power output of Stirling engine with respect to regenerator effectiveness

Effect on solar dish-Stirling system

Effect of T_H and C

From fig. 14 it can be seen that for a given concentrating ratio, the maximum power thermal efficiency of the system increases with the increasing of the absorber temperature until the maximum thermal efficiency is reached and then decreases with the increasing of the absorber temperature; for a given absorber temperature, the maximum power thermal efficiency increases with the increasing of the concentrating ratio. The values of the optimum absorber temperature and the concentrating ratio are about 1100 K and 1300, respectively, which makes the thermal efficiency get up to its maximum value about 33.16% which is close to Carnot efficiency at about 50%, approximately. It is also found that for a given concentrating ratio, when the absorber tem-

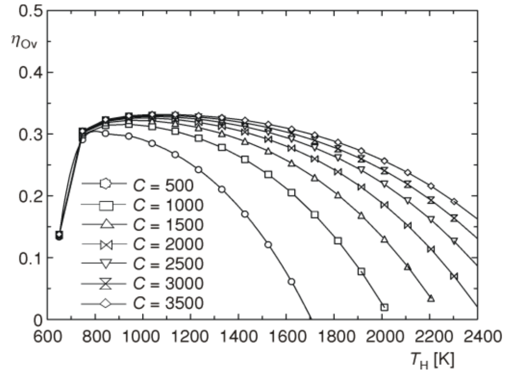

Figure 14. Variation on overall efficiency of system with respect to absorber temperature and concentration ratio

perature exceeds its optimum value, if keep increasing the absorber temperature, the maximum power thermal efficiency of the system decreases rapidly. This shows that the range of the operation absorber temperature cannot exceed its optimum temperature, which is very important for the solar dish collector because the absorber temperature varies with direct solar flux intensity and changes with time.

Effect of heat capacitance rate (h_H)

Figure 15 shows the effect on heat capacitance rate of heat source side on overall efficiency of the system at various absorber temperatures (T_H)**.** Efficiency increases with increase in T_H but its effect is not much pronounced. The effect of h_H has more influence on maximum power output as shown in fig. 16.

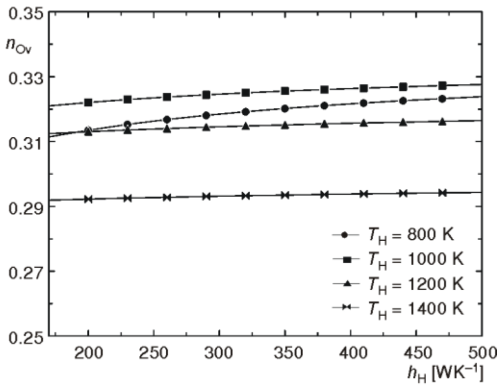

Figure 15. Variation on overall efficiency of the system with respect to heat capacitance at sink side at various absorber temperatures

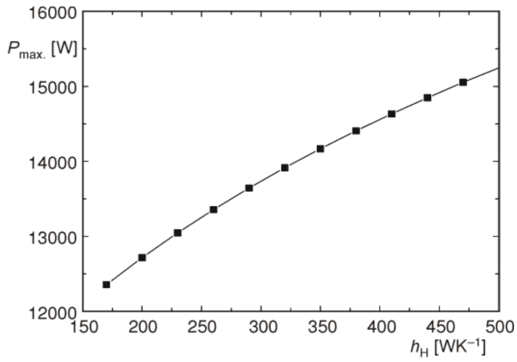

Figure 16. Variation on maximum power output of the system with respect to heat capacitance at heat source side

Effect of heat capacitance rate at sink side

Effect of h_C (fig. 17) is same as h_H (fig. 18) on efficiency and maximum power output of the system.

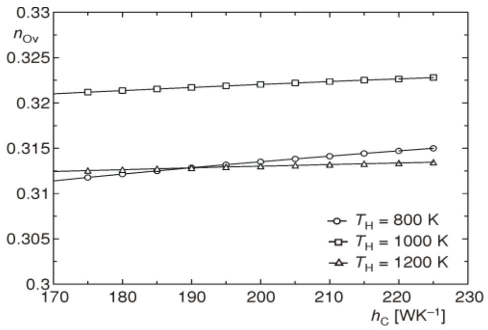

Figure 17. Variation on overall thermal efficiency of the system with respect to heat sink coefficient at various absorber temperatures

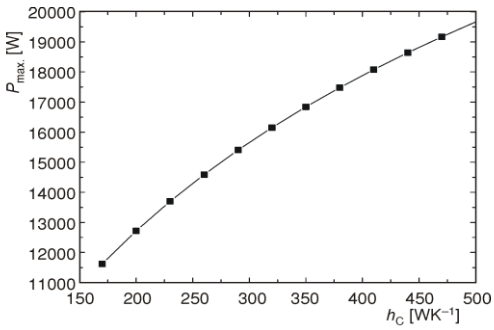

Figure 18. Variation on maximum power output of the system with respect to heat sink side capacitance at various absorber temperatures

Effect of cycle temperature ratio (x)

The variation of the power output and thermal efficiency with respect to the cycle temperature ratio ($x = T_1/T_2$) for a typical set of operating parameters is shown in fig. 19. It is seen from fig.19 that the power output first increase and then decrease while the efficiency monotonically decreases as the cycle temperature ratio (x) decreases. These properties can be directly expounded by eqs. (21) and (24), because the power output is not monotonic functions of x while the efficiency is a monotonically increasing function of x. The optimum value of x ranges from 0.56-0.58.

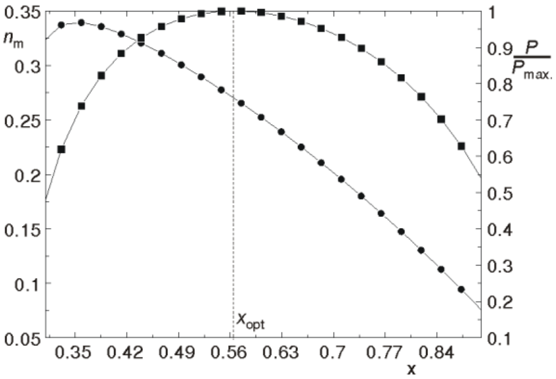

Figure 19. Effect of temperature ratio on overall efficiency and power output of the system

Effect of heat leak coefficient (k_0)

The effect of the heat leak coefficient on the maximum power thermal efficiency of the system is shown in fig. 20. It is seen from the figure that the heat leak coefficient reduces the efficiency of system with increase and rate of decrease is more at lower absorber temperature.

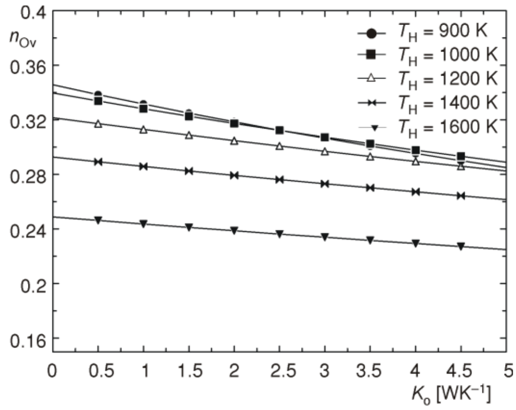

Figure 20. Variation of the maximum power thermal efficiency of the dish system at different heat leak coefficient and the absorber temperature

Effect of effectiveness (ε_R)

The effect of the effectiveness of the regenerator on the maximum power thermal efficiency of the system is shown in fig. 21. It is seen that the maximum power efficiency increases with the increasing of the effectiveness of the regenerator. Therefore, the most efficient and cost effective regenerator should be used for the Stirling engine.

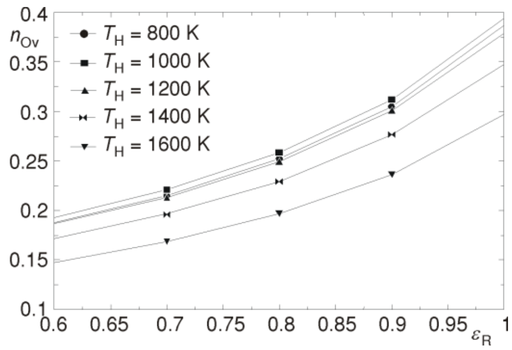

Figure 21. Variation of the maximum power efficiency of the dish system for different effectiveness of the regenerator and the absorber temperature

Conclusions

Finite-time thermodynamics has been applied to optimize the maximum power output and the corresponding thermal efficiency of the solar-powered dish-Stirling heat engine with regenerative losses. It is found that regenerative effectiveness and heat source/sink temperatures effects the optimum thermal efficiency and maximum power output of the system. It is also desirable to have a high temperature heat source and low temperature heat sink from the point of view of higher power output and the corresponding thermal efficiency. Other factors like conductive thermal bridging losses, heat transfer coefficients at source/sink side, temperature ratio of cycle and finite regenerative processes time are also included in the analysis. The values of optimum absorber temperature (T_H), collector concentrating ratio (C) and temperature ratio (x) are about 1100 K, 1300, and 0.57, respectively. Thus the present analysis provides a new theoretical basis for the design, performance evaluation and improvement of solar dish-Stirling heat engine.

Nomenclature

A – heat transfer area, [m^2]
C – collector concentration ratio
C_v – specific heat capacity, [Jmol^{-1}K^{-1}]
h – heat transfer coefficient, [WK^{-1}] or [WK^{-4}] or [Wm^{-2}K^{-1}]
I – direct solar flux intensity, [Wm^{-2}]
k_a – heat leak coefficient between the absorber and the heat sing, [WK^{-1}]
n – the mole number of working fluid, [mol]
P – power, [W]
Q – heat transfer, [J]
R – the gas constant, [Jmol^{-1}K^{-1}]
T – temperature, [K]
t – total cycle time, [s]
W – work, [J]

Greeks *symbols*

ε – emissivity factor
ε_R – effectiveness of regenerator
λ – ratio of volume during regenerative processes
η – thermal efficiency
σ – Stefan's constant, [Wm^{-2}K^{-4}]

Subscripts

a – ambient or optics
C – heat sink
c – collector
H – absorber
max. – maximum optimum condition
R – regenerator
1, 2 – initial, final
1, 2, 3, 4 – state points

References

[1] Mancini, T., Heller, P., Dish-Stirling Systems: An Overview of Developmentand Status [J], *J. Solar Energy Eng., 125* (2003), 2, pp. 135-151
[2] Sahin, A. Z., Finite-Time Thermodynamic Analysis of a Solar Driven Heat Engine, *Exergy Int., 1,* (2001), 2, pp. 122-126
[3] Curzon, F. L., Ahlborn, B., Efficiency of Carnot Heat Engine at Maximum Power Output, *Am. J. Phys, 43* (1975), 2, pp. 22-24
[4] Howell, J. R., Bannerot, R. B., Optimum Solar Collector Operation for Maximizing Cycle Work Output, *Solar Energy, 19* (1977), 2, pp. 149-153
[5] Wu, F, *et al*., Optimum Performance of Irreversible Stirling Engine with Imperfect Regeneration, *Energy Conversion and Management, 39* (1998), 8, pp. 727-732
[6] Wu, F., *et al.,* Performance and Optimization Criteria of Forward and Reverse Quantum Stirling Cycles, *Energy Conversion and Management, 39* (1998), 8, pp. 733-739
[7] Chen, J., *et. al*., Efficiency Bound of a Solar-Driven Stirling Heat Engine System, *Int. J. Energy Res., 22* (1998), 9, pp. 805-812
[8] Costea, M., Petrescu, S., Harman, C., The Effect of Irreversibilities on Solar Stirlingengine Cycle Performance, *Energy Convers Manage, 40* (1999), 15-16, pp. 1723-1731
[9] Tlili, I., Timoumi, Y., Ben Nasrallah, S., Analysis and Design Consideration of Mean Temperature Differential Stirling Engine for Solar Application, *Renew, Energy, 33* (2008) 3, pp. 1911-1921

[10] Chen, L., Li, J., Sun, F., Optimal Temperatures and Maximum Power Output of a Complex System with Linear Phenomenological Heat Transfer Law, *Thermal Science, 13* (2009) 4, pp. 33-40
[11] Li, J., Chen, L., Sun, F., Maximum Work Output of Multistage Continuous Carnot Heat Engine System with Finite Reservoirs of Thermal Capacity and Radiation between Heat Source and Working Fluid, *Thermal Science, 14* (2010), 1, pp. 1-9
[12] Thombare, D. G., Verma, S. K., Technological Development in the Stirling Cycle Engines, *Renew Sustain Energy Rev., 12* (2008). 1, pp. 1-38
[13] Durmayaz, A., *et al*., Optimization of Thermal Systems Based on Finite-Time Thermodynamics and Thermoeconomics, *Prog Energy Combust Sci., 30* (2004), 2, pp. 175-271
[14] Kaushik, S. C., Kumar, S., Finite Time Thermodynamic Evaluation of Irreversible Ericsson and Stirling Heat Engines, *Energy Convers Manage, 42* (2001), 3, pp. 295-312
[15] Wu, F., *et al*., Performance Optimization of Stirling Engine and Cooler Based on Finite-Time Thermodynamic, Chemical Industry Press Beijing, 2008, pp. 59
[16] Yaqi, L., He, Y., Wang, W., Optimization of Solar-Powered Stirling Heat Engine with Finite-Time Thermodynamics, *Renew, Energy, 36* (2010), 1, pp. 421-427

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