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.

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1.0 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 [2, 8]. 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[2]. The Stirling Energy System (SES) dish has held the world’s efficiency record for converting solar energy into grid-quality electricity, and 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 1.4GW are slated to begin construction in 2010 using the Stirling Energy Systems (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 and Ahlborn [1], 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 [3-4] powered by low concentrating solar collectors [5, 6]. 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 [14, 15]. 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. Ahmet Z. Sahin [8] 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 \( W = (Q_{\text{in}} - Q_{\text{out}}) \).

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.

2.0 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.

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 infiniteregeneration time to transfer finite heat amount, and this is impractical. Therefore, it is desirable to consider a real regenerator with heat losses $\Delta Q_R$. In addition, we also consider conductive thermal bridging losses $Q_b$ from the absorber to the heat sink.
3.0 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. These models are described as follows.

3.1 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 [10]:

$$q_u = I A_c \eta_0 - A_H [h(T_H - T_a) + \varepsilon \sigma (T_H^4 - T_a^4)]$$  \hspace{1cm} (1)

where $I$ is the direct solar flux intensity, $A_c$ is the collector aperture area, $\eta_0$ is the collector optical efficiency, $A_H$ is the absorber area, $h$ is conduction/convection coefficient, $T_H$ is the absorber temperature, $T_a$ is the ambient temperature, $\varepsilon$ is emissivity factor of the collector, $\sigma$ is the Stefan’s constant.

Thermal efficiency of the dish collector is defined by $\eta_d$:

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

(2)

3.2 Finite-time thermodynamics analysis of the Stirling heat engine

3.2.1 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 (Howell and Bannerot, 1977)

$$\Delta Q_R = n C_v (1 - \varepsilon_R) (T_1 - T_2)$$  \hspace{1cm} (3)

where $C_v$ is the heat capacity of the working substance, $n$ is the number of mole partaking in the regenerative branches, $\varepsilon_R$ is 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 [8]. 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 [9]

$$\frac{dT}{dt} = \alpha$$  \hspace{1cm} (4)

where $\alpha$ 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

$$t_R = t_3 + t_4 = 2 \alpha (T_1 - T_2)$$

3.2.2 Heat transfer across the Stirling cycle

$Q_1$ is the amount of heat supplied to the working fluid at temperature $T_1$ and $Q_2$ is the amount of heat released by the working fluid at temperature $T_2$, then, during the two isothermal processes

$$Q_1 = Q_h + \Delta Q_R$$

$$= nRT_1 \ln(V_1/V_2) + nC_v (1-\varepsilon_R) (T_1-T_2)$$

and

$$Q_2 = Q_c + \Delta Q_R$$

$$= nRT_2 \ln(V_1/V_2) + nC_v (1-\varepsilon_R) (T_1-T_2)$$

respectively, where $n$ is the mole number of the working substance, $R$ is 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 $V_2$ are the volumes of the working substance along the constant-volume heating and cooling branches, as shown in Figure 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 considered between absorber (source/sink) and working fluid, heat transfer can be written as

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

and

$$Q_2 = h_C(T_2 - T_C)t_2$$

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 [7,9]:

$$Q_a = k_a(T_H-T_C)t$$

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

Taking into 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:

$$Q_H = Q_1 + Q_a$$

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

Thus the total cycle time is given by:
\[
t = t_1 + t_2 + t_3 + t_4 \tag{13}
\]

For the thermodynamic cycle 1-2-3-4-1, work, power output and thermal efficiency is given by[12, 13]:
\[
W = Q_H - Q_C \tag{14}
\]
\[
P = \frac{W}{t} = \frac{Q_H - Q_C}{t} \tag{15}
\]

Using Eqs.(8)-(15), we have:
\[
P = \frac{T_1 - T_2}{T_1 + Y_1(T_1 - T_2) + K_a(T_H - T_C)} \left[ \frac{T_1 + Y_1(T_1 - T_2)}{h_H(T_H - T_1) + h_{HR}(T_H^2 - T_1^2)} + \frac{T_2 + Y_1(T_1 - T_2)}{h_C(T_2 - T_C)} + Y_2(T_1 - T_2) \right]
\]

\[
\eta_s = \frac{T_1 - T_2}{T_1 + Y_1(T_1 - T_2) + K_a(T_H - T_C)} \left[ \frac{T_1 + Y_1(T_1 - T_2)}{h_H(T_H - T_1) + h_{HR}(T_H^2 - T_1^2)} + \frac{T_2 + Y_1(T_1 - T_2)}{h_C(T_2 - T_C)} + Y_2(T_1 - T_2) \right] \tag{17}
\]

where
\[
Y_1 = \frac{C_v(1 - \varepsilon_R)}{\ln \lambda} \quad \text{and}
\]
\[
Y_2 = \frac{V_1}{V_2}
\]

\[
\text{Where } \lambda = \frac{V_1}{V_2}
\]
For the sake of convenience, a new parameter \( x = T_2/T_1 \) is introduced into Eqs. (16) and (17), then we have:

\[
P = \frac{T_L - xT_1}{h_H(T_H - T_L) + h_{HR}(T_H^{\Delta} - T_L^{\Delta}) + \frac{XT_1}{h_C(xT_1 - T_c)} + Y_2(T_1 - xT_1)}
\]

(18)

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

\[
\beta_1 T_{1\text{opt}}^8 + \beta_2 T_{1\text{opt}}^5 + \beta_3 T_{1\text{opt}}^4 + \beta_4 T_{1\text{opt}}^3 + \beta_5 T_{1\text{opt}}^2 + \beta_6 T_{1\text{opt}} + \beta_7 = 0
\]

(19)

\[
E_1 x^2 + E_2 x + E_3 = 0
\]

(20)

where \( \beta_1 = h_{HR}^2 B_1 x \), \( \beta_2 = h_{HR} x (2B_1 h_{HC} - 3B_2 h_{LC} x) \), \( \beta_3 = 2h_{HR} x (3B_2 h_{LC} T_L - B_1 B_3) \), \( \beta_4 = -3B_2 h_{HR} h_{LC} T_L^2 \), \( \beta_5 = h_{HC} x (B_1 h_{HC} - B_2 h_{LC} x) \), \( \beta_6 = 2h_{HC} x (B_2 h_{LC} T_L - B_3 B_1) \), \( \beta_7 = B_1 x B_3^2 - B_2 h_{HC} h_{LC} T_L^2 \), \( B_1 = x + A_1 (1 - x) \), \( B_2 = 1 + A_1 (1 - x) \), \( B_3 = h_{HC} T_H + h_{HR} T_H^4 \).

\( E_1 = B (T_1^2 - Y_1 T_1 T_{chc}) \), \( E_2 = B (Y_1 h_{HC} T_1^2 - Y_1 h_{C} T_1 T_c) + T_1^2 h_{C} \), \( E_3 = -[B h_{C} T_1 (T_L + Y_1 T_1)] + T_1^2 h_{C} \), \( B = h_{HR} (T_H - T_1) + h_{HR} (T_H^4 - T_1^4) \).

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

\[
P_{\text{max}} = \frac{1 - x}{h_H (T_H - T_{1\text{opt}}) + h_{HR} (T_H^{\Delta} - T_{1\text{opt}}^{\Delta}) + \frac{x + Y_1(1 - x)}{h_C(xT_{1\text{opt}} - T_c)} + Y_2(1 - x)}
\]

(21)

\[
\eta_{\text{opt}} = \frac{1 - x}{1 + Y_1(1 - x) + K_a (T_H - T_c)} \left[ \frac{1 + Y_1(1 - x)}{h_H (T_H - T_{1\text{opt}}) + h_{HR} (T_H^{\Delta} - T_{1\text{opt}}^{\Delta}) + \frac{x + Y_1(1 - x)}{h_C(xT_{1\text{opt}} - T_c)} + Y_2(1 - x)} \right]
\]

(22)

Special cases:

1. when \( h_{HR} = 0 \), 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:

\[
\beta_5 T_{1\text{opt}} T^2 + \beta_6 T_{1\text{opt}} + D_7 = 0
\]
Solve the Eqn. (24) we get:

\[ T_{1\text{opt}} = \]

where

2. When \( \varepsilon_R = 1 \), 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:

\[
P_{\text{max}} = \frac{1 - x}{\frac{1}{h_H(T_H - T_{1\text{opt}})} + \frac{x}{h_C(xT_{1\text{opt}} - T_C)} + Y_2(1 - x)}
\]

\[
\eta_{\text{opt}} = \frac{1 - x}{1 + h_a(T_H - T_C)\left[\frac{1}{h_H(T_H - T_{1\text{opt}})} + \frac{x}{h_C(xT_{1\text{opt}} - T_C)} + Y_2(1 - x)\right]}
\]

3. when \( h_{HR} = 0 \), \( \varepsilon_R = 1 \) and considering \( x = \sqrt{\frac{T_H}{T_C}} \), then maximum power and thermal efficiency can be given as:

\[
P_{\text{max}} = \frac{F_2\left(\sqrt{\frac{T_H}{T_C}} - \sqrt{\frac{T_H}{T_C}}\right)^2}{1 + F_2Y_2\left(\sqrt{\frac{T_H}{T_C}} - \sqrt{\frac{T_H}{T_C}}\right)^2}
\]

\[
\eta_{\text{opt}} = 1 - \sqrt{\frac{T_H}{T_C}}
\]

However, physically for finite time regenerative time \( \varepsilon_R \) 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.

\( t_c = \gamma(t_1 + t_2) \)

where \( \gamma \) is the proportionality constant then
\[ P_{\text{max}} = \frac{1 - x}{1 + Y_1(1 - x) + k_0(T_H - T_C)\left[\frac{1 + Y_1(1 - x)}{h_H(T_H - T_2) + h_HR(T_H^4 - T_1^4)} + \frac{x + Y_1(1 - x)}{h_C(T_2 - T_C)}\right]} \]

\[ \eta_{\text{sopt}} = \frac{1 - x}{1 + Y_1(1 - x) + k_0(T_H - T_C)\left[\frac{1 + Y_1(1 - x)}{h_H(T_H - T_2) + h_HR(T_H^4 - T_1^4)} + \frac{x + Y_1(1 - x)}{h_C(T_2 - T_C)}\right]} \]

5. When the regenerative time is zero, i.e. \( t_r = 0 \), the maximum power output is given by:

\[ P_{\text{max}} = \]

3.3 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:

\[ \eta_{\text{Ov}} = \eta_d \eta_{\text{sopt}} \] (23)

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

\[ \eta_{\text{Ov}} = \]

\[ \frac{k_0[h(T_H - T_2) + \varepsilon_r(T_H^4 - T_1^4)]/IC \times [(1 - x)/(1 + Y_1(1 - x) + k_0(T_H - T_C)(1 + Y_1(1 - x) + k_0(T_H - T_C))]} \]

\[ \text{ (24) } \]

4.0 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 (\( \varepsilon_R \)), the heat leak coefficient (\( k_0 \)), heat transfer coefficients and volume ratio (\( \lambda \)) on the solar-powered dish-Stirling heat engine system, all the
other parameters will be kept constant as $n = 10 \text{ mol}$, $R = 4.3 \text{ Jmol}^{-1}\text{K}^{-1}$, $C_v = 15 \text{ Jmol}^{-1}\text{K}^{-1}$, $\varepsilon = 0.92$, $T_0 = 300 \text{ K}$, $h = 20 \text{ Wm}^{-2}\text{K}^{-1}$, $\sigma = 5.671 \text{ Wm}^{-2}$, $\alpha = 1000 \text{ Ks}^{-1}$, $I = 1000 \text{ Wm}^{-2}$. The results obtained are as follows.

4.1 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 Figs. 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 opticaleefficiency of the concentrator.

![Fig.4 Variation of the thermal efficiency of the collector for different receiver temperature and the concentrating ratio.](image)

4.2 Effect on Stirling engine

4.2.1 Effect of $T_H$
Fig. 5-7 shows the variation on work output ($W$), maximum power output ($P_{\text{max}}$) and thermal efficiency of Sterling engine ($\eta_s$) 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.

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.
4.2.2 Effect of heat sink temperature ($T_C$)

It is seen from Fig. 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.

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

Fig. 8 Variation on Work output of Stirling engine with respect to heat sink temperature
Fig. 9 Variation on maximum power output of Stirling engine with respect to heat sink temperature

Fig. 10 Variation on thermal efficiency of Stirling engine with respect to heat sink temperature
4.2.3 Effect of effectiveness of regenerator ($\varepsilon_R$)

As the regenerative effectiveness ($\varepsilon_R$) increases, the heat transfers ($Q_1$ and $Q_2$) decreases but the regenerative heat transfer ($Q_R$) increases. It can be seen from Fig. 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.

![Graph 1](image1)

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

![Graph 2](image2)

**Fig. 12 Variation on maximum power output of Stirling engine with respect to regenerator effectiveness**
4.3 Effect on solar dish-Stirling system

Effect of various parameters on overall efficiency of system is given as:

4.3.1 Effect of $T_H$ and $C$

From Figs. 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 temperature 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.
4.3.2 Effect of heat capacitance rate

Fig. 15 shows the effect on heat capacitance rate \((h_H)\) 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.
4.3.3 Effect of heat capacitance rate at sink side

Effect of $h_C$ is same as $h_H$ on efficiency and maximum power output of the system.
4.3.4 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 equations (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.

Fig. 19 Effect of temperature ratio on overall efficiency and power output of the system
4.3.5 *Effect of heat leak coefficient (k₀)*

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 the system with increase and the rate of decrease is more at lower absorber temperature.

![Graph showing the variation of maximum power thermal efficiency with heat leak coefficient](image)

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

4.3.6 *Effect of εᵣ*

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

5.0 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
- $h$ heat transfer coefficient, WK$^{-1}$ or WK$^{-4}$ or Wm$^{-2}$K$^{-1}$
- $I$ direct solar flux intensity Wm$^{-2}$
- $C_v$ specific heat capacity, Jmol$^{-1}$K$^{-1}$
- $n$ the mole number of working fluid, mol
- $\tau$ regenerative time constant, Ks$^{-1}$
- $P$ power, W
- $Q$ heat transfer, J
- $R$ the gas constant, Jmol$^{-1}$K$^{-1}$
- $t$ total cycle time, s
- $T$ temperature, K
W  work, J
λ ratio of volume during regenerative processes

Greeks

η thermal efficiency
σ Stefan’s constant, Wm\(^{-2}\)K\(^{-4}\)
ε emissivity factor
ε\(_R\) effectiveness of regenerator
k\(_0\) heat leak coefficient W K\(^{-1}\)

Subscripts
cc collector
H absorber
C heat sink
max maximum/optimum condition
R regenerator
a ambient or optics
1, 2 initial, final
1, 2, 3, 4 state points
References


