ENTROPY GENERATION MINIMIZATION OF IMAGING CONCENTRATING SOLAR COLLECTORS

Soteris A. Kalogirou
Department of Mechanical Engineering, Higher Technical Institute,
P. O. Box 20423, Nicosia 2152, Cyprus.
Tel. +357-22-406466, Fax. +357-22-494953, Email: skalogir@spidernet.com.cy

Abstract - There are three types of imaging concentrating solar collectors, the parabolic trough, the parabolic dish and the central receiver. The second law of thermodynamics is used to analyze the potential for entropy generation minimization of imaging concentrating solar collectors. It is shown that the amount of exergy (useful energy) delivered by solar concentrating collector systems is affected by heat transfer irreversibilities occurring between the sun and the collector, between the collector and ambient air and within the collector receiver (absorber). Analysis is performed and relations are derived in this paper by considering both, an isothermal and a non-isothermal collector which is a more realistic model particularly for the long parabolic trough collectors. Relations for the optimum operating conditions, in terms of the optimum collector outlet temperature for minimum heat transfer irreversibility or entropy generation minimization (or maximum exergy delivery), are derived. The importance of operating at the optimum delivery temperature is analyzed and optimum values of entropy generated are derived for the collector types considered.

1. INTRODUCTION

The second law of thermodynamics provides a scientific basis for analysing how effectively a system utilises available energy.

Entropy generation minimization is synonymous to maximizing exergy or useful energy delivery from a system. The method of entropy generation minimization has emerged during the last two decades as a distinct subfield in heat transfer. The method consists of the simultaneous application of heat transfer and thermodynamic principles in the pursuit of realistic models of heat transfer processes, devices and installations. Realistic models mean the ones that account for the inherent irreversibility of heat, mass and fluid flow processes. In engineering this method is also known as thermodynamic optimization and thermodynamic design.

Additionally, exergy analysis is an important exercise carried out in order to perform a thermodynamic assessment of power generation systems. Exergy analysis finds application in many energy systems. Bejan (1999) has applied an exergy analysis for the optimization of aircraft energy-system design. Exergy analysis establishes the theoretical performance limit and the minimization of exergy destruction brings the design as closely as possible to the theoretical limit. The key problem is the extraction of the maximum exergy from a hot gaseous stream that is gradually cooled and eventually discharged into ambient. The optimal configuration consists of a heat transfer surface with a temperature that decays exponentially in the flow direction. According to the author, the application of these principles in aircraft energy system design also sheds light on the design principle that generated all the systems that use powered flight, engineered and natural.

Basic energy and exergy analysis for solar thermal power system components, i.e., parabolic trough collector and Rankine heat engine is presented by Singh *et al.* (2000). This analysis is carried out in order to evaluate the respective losses as well as the exergetic efficiency for typical solar thermal power systems under given operating conditions. It is shown that the main energy loss takes place at the condenser of the heat engine whereas exergy analysis shows that the collector-receiver assembly is the part where losses are maximum.

Yantoski (2000) also show how exergy could be used for the optimization of the heat transfer through a wall, an electrical conductor and thermal insulation. As shown by the author exergonomics is a mirror image of ordinary economics in which only exergy expenditures are used instead of monetary ones.

Additional to the above applications, Saha and Mahanta (2001) presented an application in which entropy generation minimization was used for the thermodynamic optimization of flat-plate collectors whereas Torres-Reyes *et al.* (2002) showed how the method can be used for designing dryers operated by flat-plate collectors. Finally an exergy analysis of renewable energy sources (solar, wind and geothermal) is presented by Koroneos *et al.* (2003).

The objective of this work is to establish a theoretical framework for analyzing the collection and delivery of solar exergy, based on the second law point of view of entropy generation minimization. The importance of solar energy as an environmental friendly source of energy is well known nowadays, however to use this source effectively one must understand the thermodynamic losses associated with the harnessing of solar energy.

2. COLLECTOR TYPES CONSIDERED

A large number of solar collectors are available in the market. A comprehensive list is shown in Table 1. In this paper only imaging solar collectors are considered.

Table 1. Solar energy collectors

Motion	Collector type	Absorber type	Concentration ratio	Indicative temperature range (°C)
Stationary	Flat plate collector	Flat	1	30-80
	Evacuated tube collector	Flat	1	50-200
	Compound parabolic collector	Tubular	1-5	60-240
Single-axis tracking			5-15	60-300
	Fresnel lens collector	Tubular	10-40	60-250
	Parabolic trough collector	Tubular	15-45	60-300
	Cylindrical trough collector	Tubular	10-50	60-300
Two-axes tracking	Parabolic dish reflector	Point	100-1000	100-500
	Heliostat field collector	Point	200-1500	150-2000

Note: Concentration ratio is defined as the aperture area divided by the receiver/absorber area of the collector.

Imaging collectors are concentrating collectors which focus an image of the sun onto their receiver. These collectors are the parabolic trough, the parabolic dish and the central receiver. These are initially described briefly with emphasis on their use for power generation and subsequently analyzed thermodynamically using the concepts of the second law and in particular the entropy generation minimization. The thermodynamic analysis of these collectors differs somewhat from the analysis of the other collector types as the collector capture area is much bigger than the collector absorber/receiver.

2.1 Parabolic Trough Collector Systems

Parabolic trough collectors, shown schematically in Fig. 1, have a linear parabolic-shaped reflector that focuses the sun's direct beam radiation on a linear receiver located at the focus of the parabola. The collector tracks the sun either from east to west (collector axis in north-south direction) or from north to south (collector axis in an east-west direction) during the day to ensure that the sun is continuously focused on the linear receiver. A heat transfer fluid, or water at high pressure is circulated in the receiver and is heated up to as high as about 400°C.

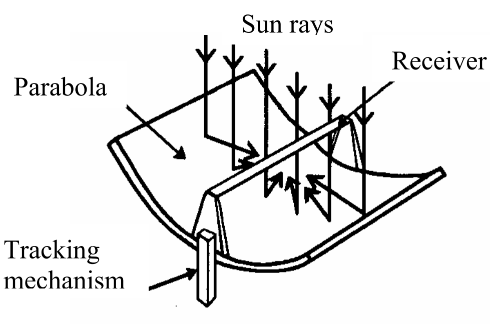

Fig. 1 Schematic of a parabolic trough collector system

Applications of parabolic trough collectors are reported by Bakos *et al.* (1999) and Kalogirou *et al.* (1996). Additionally Lupfert *et al.* (2000) and Geyer *et al.* (2002) present the design of EuroTrough, a new parabolic trough collector, in which an advance lightweight structure is used to achieve cost efficient solar power generation.

The biggest application of this type of system is the Southern California power plants, known as Solar Electric Generating Systems (SEGS), which have a total installed capacity of 354 MWe (Kearney and Price, 1992). There are nine SEGS parabolic trough power plants in total consisting of large fields of parabolic trough collectors that track the sun from east to west, a heat transfer fluid, steam generation system, a Rankine steam turbine and optional thermal storage and/or fossil-fired backup systems (EPRI, 1997; Pilkington Solar International, 1996).

A heat transfer fluid is used in SEGS which circulates through the receiver and returns to a steam generator of a conventional steam cycle power plant. Given sufficient solar input, the plants can operate at full-rated power using solar energy alone. The collector field is made up of a large field of single-axis-tracking parabolic trough solar collectors. The solar field is modular in nature and comprises many parallel rows of solar collectors, normally aligned on a north-south horizontal axis. During the summer months, the plants typically operate for 10–12 hours a day on solar energy at full-rated electric output. To enable parabolic trough collector plants to achieve the rated electric output during overcast or night time periods, the plants have been designed as hybrid solar/fossil plants; i.e., a backup natural gas-fired capability is used to supplement the energy output during periods of low solar radiation. In addition, thermal storage can be integrated into the plant design to allow solar energy to be stored and used when power is required.

2.2 Parabolic Dish Systems

Parabolic dish systems, shown schematically in Fig. 2, collect solar energy coming directly from the sun and concentrate or focus it on a small area. The dish structure must track fully the sun to reflect the beam into the thermal receiver.

The main use of this type of concentrator is for parabolic dish engines. A parabolic dish-engine system is an electric generator that uses sunlight instead of crude oil or coal to produce electricity. The major parts of a system are the solar dish concentrator and the power conversion unit.

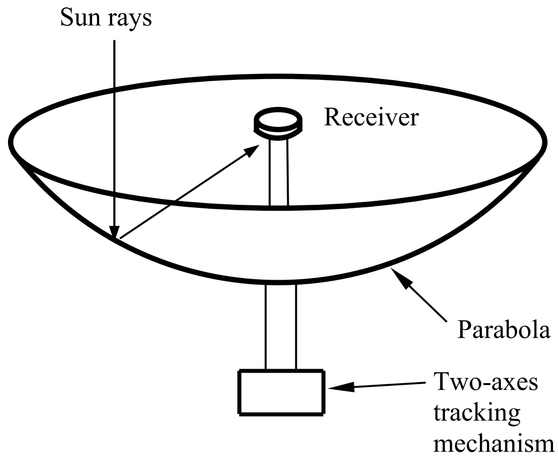

Fig. 2 Schematic of a parabolic dish collector

The power conversion unit includes the thermal receiver and the heat engine. The thermal receiver absorbs the concentrated beam of solar energy, converts it to heat, and transfers the heat to the heat engine. A thermal receiver can be a bank of tubes with a cooling fluid circulating through it. The heat transfer medium usually employed as the working fluid for an engine is hydrogen or helium. Alternate thermal receivers are heat pipes wherein the boiling and condensing of an intermediate fluid is used to transfer the heat to the engine. The heat engine system takes the heat from the thermal receiver and uses it to produce electricity. The Stirling engine is the most common type of heat engine used in dish-engine systems. Other possible power conversion unit technologies that are evaluated for future applications are microturbines and concentrating photovoltaics (Pitz-Paal, 2002).

2.3 Central Receiver Systems

A central receiver system is shown schematically in Fig. 3. In this case incident sunrays are reflected by large tracking mirrored collectors, called heliostats, which concentrate the energy flux towards radiative/convective heat exchangers, called solar receivers, where energy is transferred to a working thermal fluid (Romero *et al.*, 2002). After energy collection by the solar system, the conversion of thermal energy to electricity has many similarities with the conventional fossil-fuelled thermal power plants.

The typical optical concentration factor for this type of system varies from 200 to 1,000 and plant sizes are in the range of 10 to 200 MW. The average solar flux impinging on the receiver has values between 200 and 1,000 kW/m^2. This high flux allows working at relatively high temperatures of more than 1,000°C and to integrate thermal energy in more efficient cycles. Central receiver systems can easily integrate in fossil-fuelled plants for hybrid operation in a wide variety of options and have the potential to operate more than half the hours of each year at nominal power using thermal energy storage.

Central receiver systems are considered to have a large potential for mid-term cost reduction of electricity compared to parabolic trough technology since they allow many intermediate steps between the integration in a conventional Rankine cycle up to the higher exergy cycles using gas turbines at temperatures above 1000 ºC, and this subsequently leads to higher efficiencies and larger throughputs (Schwarzbozl *et al.*, 2000; Chavez *et al.*, 1993). Another alternative is to use Brayton cycle turbines, which require higher temperature than the ones employed in Rankine cycle.

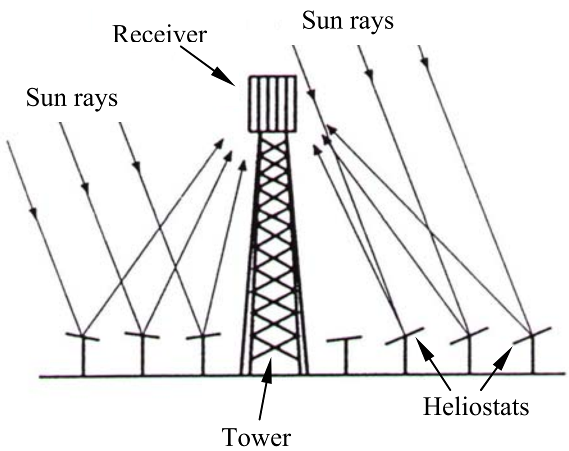

Fig. 3. Schematic of central receiver

3. SECOND LAW ANALYSIS

The analysis presented here is based on Bejan's work (Bejan et al., 1981; Bejan, 1995). The analysis however is adapted to imaging collectors. Consider that the collector has an aperture area (or total heliostat area) A_a and receives solar radiation at the rate Q^* from the sun as shown in Fig. 4. The net solar heat transfer Q^* is proportional to the collector area A_a and the proportionality factor q^* (W/m^2) which varies with geographical position on the earth, the orientation of the collector, meteorological conditions and the time of day. In the present analysis q^* is assumed to be constant and the system is in steady state, i.e.,

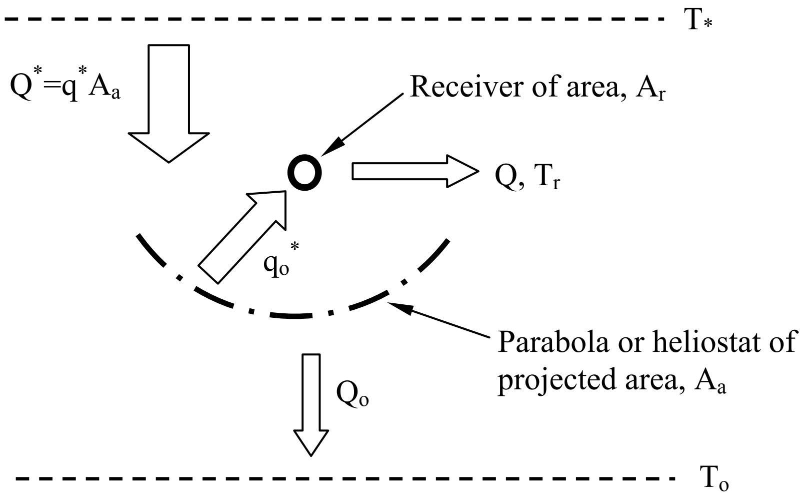

Fig. 4. Imaging concentrating collector model

$$Q^* = q^* A_a \quad (1)$$

For concentrating systems q^* is the solar energy falling on the reflector. In order to obtain the energy falling on the collector receiver the tracking mechanism accuracy, the optical errors of the mirror including its reflectance and the optical properties of the receiver glazing must be considered.

Therefore, the radiation falling on the receiver q_o^* is a function of the optical efficiency, which accounts for all the above errors. For the parabolic trough collector (Kalogirou, 1996):

$$n_o = \rho_m \tau_a \alpha_a \gamma \left[(1 - A_f \tan(\beta)) \cos(\beta) \right] \quad (2)$$

and the radiation falling on the receiver is:

$$q_o^* = n_o q^* = \frac{n_o Q^*}{A_a} \quad (3)$$

The incident solar radiation is partly delivered to a power cycle (or user) as heat transfer Q at the receiver temperature T_r. The remaining fraction Q_o represents the collector-ambient heat loss:

$$Q_o = Q^* - Q \quad (4)$$

For imaging concentrating collectors Q_o is proportional to the receiver-ambient temperature difference and to the receiver area as:

$$Q_o = U_r A_r (T_r - T_o) \quad (5)$$

where U_r is the overall heat transfer coefficient based on A_r. It should be noted that U_r is a characteristic constant of the collector.

Combining equations (4) and (5) it is apparent that the maximum receiver temperature occurs when Q=0, i.e., when the entire solar heat transfer Q^* is lost to the ambient. The maximum collector temperature is given in dimensionless form by:

$$\theta_{\max} = \frac{T_{r,\max}}{T_o} = 1 + \frac{Q^*}{U_r A_r T_o} \quad (6)$$

Combining Eq. (3) and (6):

$$\theta_{\max} = 1 + \frac{q_o^* A_a}{n_o U_r A_r T_o} \quad (7)$$

Considering that $C=A_a/A_r$, then:

$$\theta_{\max} = 1 + \frac{q_o^* C}{n_o U_r T_o} \quad (8)$$

As can be seen from Eq. (8), $\theta_{\max}$ is proportional to C, i.e., the higher the concentration ratio of the collector the higher is $\theta_{\max}$ and $T_{r,\max}$. The term $T_{r,\max}$ in Eq. (6) is also known as the stagnation temperature of the collector, i.e., the temperature that can be obtained at no flow condition. In dimensionless form the collector temperature $\theta=T_r/T_o$ will vary between 1 and $\theta_{\max}$, depending on the heat delivery rate Q. The stagnation temperature $\theta_{\max}$ is the parameter that describes the performance of the collector with regard to collector-ambient heat loss as there is no flow through the collector and all the energy collected is used to raise the temperature of the working fluid to stagnation temperature which is fixed at a value corresponding to the energy collected equal to energy loss to ambient. Thus the collector efficiency is given by:

$$\eta_c = \frac{Q}{Q^*} = 1 - \frac{\theta - 1}{\theta_{\max} - 1} \tag{9}$$

Therefore η_c is a linear function of collector temperature. At stagnation point the heat transfer Q carries zero exergy or zero potential for producing useful work.

3.1 Minimum Entropy Generation Rate

The minimization of the entropy generation rate is the same as the maximization of the power output. The process of solar energy collection is accompanied by the generation of entropy upstream of the collector, downstream of the collector and inside the collector as shown in Fig. 5.

The exergy inflow coming from the solar radiation falling on the collector surface is:

$$E_{x,in} = Q^* \left(1 - \frac{T_o}{T_*} \right) \tag{10}$$

where T_* is the apparent sun temperature as an exergy source. In this analysis the value suggested by Petela (1964) is adopted, i.e., T_* is approximately equal to $\frac{3}{4}T_s$, where T_s is the apparent black body temperature of the sun, which is about 6000K. Therefore T_* considered here is 4500K. It should be noted that in this analysis T_* is also considered constant and as its value is much greater than T_o, $E_{x,in}$ is very near Q^*. The output exergy from the collector is given by:

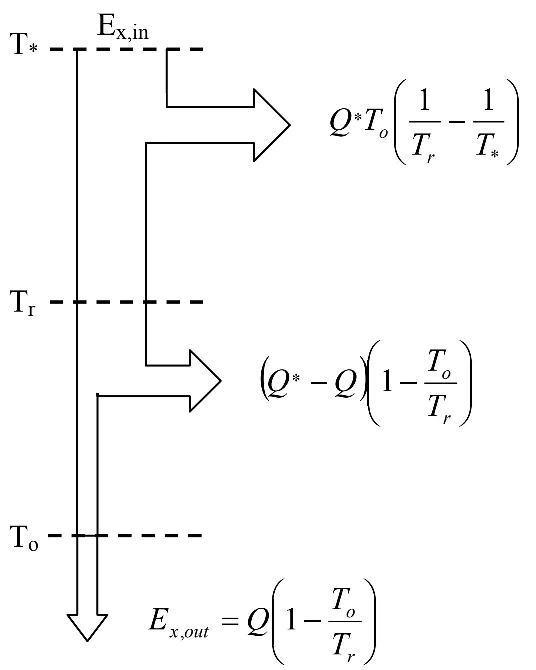

Fig. 5 Exergy flow diagram

$$E_{x,out} = Q \left(1 - \frac{T_o}{T_r} \right) \tag{11}$$

whereas the difference between the $E_{x,in}$-$E_{x,out}$ represents the destroyed exergy. From Fig. 5, the entropy generation rate can be written as:

$$S_{gen} = \frac{Q_o}{T_o} + \frac{Q}{T_r} - \frac{Q^*}{T_*} \tag{12}$$

This equation can be written with the help of equation (4) as:

$$S_{gen} = \frac{1}{T_o} \left[Q^* \left(1 - \frac{T_o}{T_*} \right) - Q \left(1 - \frac{T_o}{T_r} \right) \right] \tag{13}$$

By using Eq. (10) and Eq. (11), Eq. (13) can be written as:

$$S_{gen} = \frac{1}{T_o} \left(E_{x,in} - E_{x,out} \right) \tag{14}$$

or
$$E_{x,out} = E_{x,in} - T_o S_{gen} \tag{15}$$

Therefore, if we consider $E_{x,in}$ constant, the maximisation of the exergy output ($E_{x,out}$) is the same as the minimisation of the exergy generation S_{gen}.

3.2 Optimum Collector temperature

By substituting equations (4) and (5) into equation (13) the rate of entropy generation can be written as:

$$S_{gen} = \frac{U_r A_r (T_r - T_o)}{T_o} - \frac{Q^*}{T_*} + \frac{Q^* - U_r A_r (T_r - T_o)}{T_r} \tag{16}$$

By applying Eq. (8) in (16) and by performing various manipulations:

$$\frac{S_{gen}}{U_r A_r} = \theta - 2 - \frac{q_o^* C}{n_o U_r T_*} + \frac{\theta_{\max}}{\theta} \tag{17}$$

The dimensionless term $S_{gen}/U_r A_r$ accounts for the fact that the entropy generation rate scales with the finite size of the system which is described by $A_r = A_a/C$.

By differentiating Eq. (17) with respect to θ and setting to zero the optimum collector temperature (θ_{opt}) for minimum entropy generation is obtained:

$$\theta_{opt} = \sqrt{\theta_{\max}} = \left(1 + \frac{q_o^* C}{n_o U_r T_o} \right)^{1/2} \tag{18}$$

By substituting $\theta_{\max}$ by $T_{r,max}/T_o$ and θ_{opt} by $T_{r,opt}/T_o$, Eq. (18) can be written as:

$$T_{r,opt} = \sqrt{T_{r,\max} T_o} \tag{19}$$

This equation states that the optimal collector temperature is the geometric average of the maximum collector (stagnation) temperature and the ambient temperature. Typical stagnation temperatures and the resulting optimum operating temperatures for the various types of collectors considered in this work are shown in Table 2. The stagnation temperatures shown in Table 2 are estimated by considering mainly the collector radiation losses.

As can be seen from the data presented in Table 2 for high performance collectors, like the central receiver, it is better to operate the system at high flow rates in order to lower the temperature around the value shown instead of operating at very high temperature, in order to obtain higher thermodynamic efficiency from the collector system.

By applying Eq. (18) to Eq. (17), the corresponding minimum entropy generation rate is:

$$\frac{S_{gen,\min}}{U_r A_r} = 2\left(\sqrt{\theta_{\max}} - 1\right) - \frac{\theta_{\max} - 1}{\theta_*} \tag{20}$$

where $\theta_* = T_*/T_o$. It should be noted that for flat-plate and low concentration ratio collectors, the last term of Eq. (20) is negligible as θ_* is much bigger than θ_{max}-1 but it is not for higher concentration collectors, like the central receiver and the parabolic dish ones, which have stagnation temperatures of several thousands of degrees.

By applying the stagnation temperatures shown in Table 2 to Eq. (20), the dimensionless entropy generated for the various collector types considered here as shown in Fig. 6 are obtained. As can be seen the entropy generated presents the same trend as the stagnation temperature.

3.3 Non Isothermal Collector

So far the analysis was carried out by considering an isothermal collector. For a non isothermal one, which is a more realistic model particularly for the long parabolic trough collectors, and by applying the principle of energy conservation:

Table 2. Optimum collector temperatures for various types of collectors

Collector type	Concentration Ratio	Stagnation temperature (°C)	Optimal temperature (°C)
Parabolic trough	50	565	227
Parabolic dish	500	1285	408
Central receiver	1500	1750	503
Notes: Ambient temperature considered = 25°C			

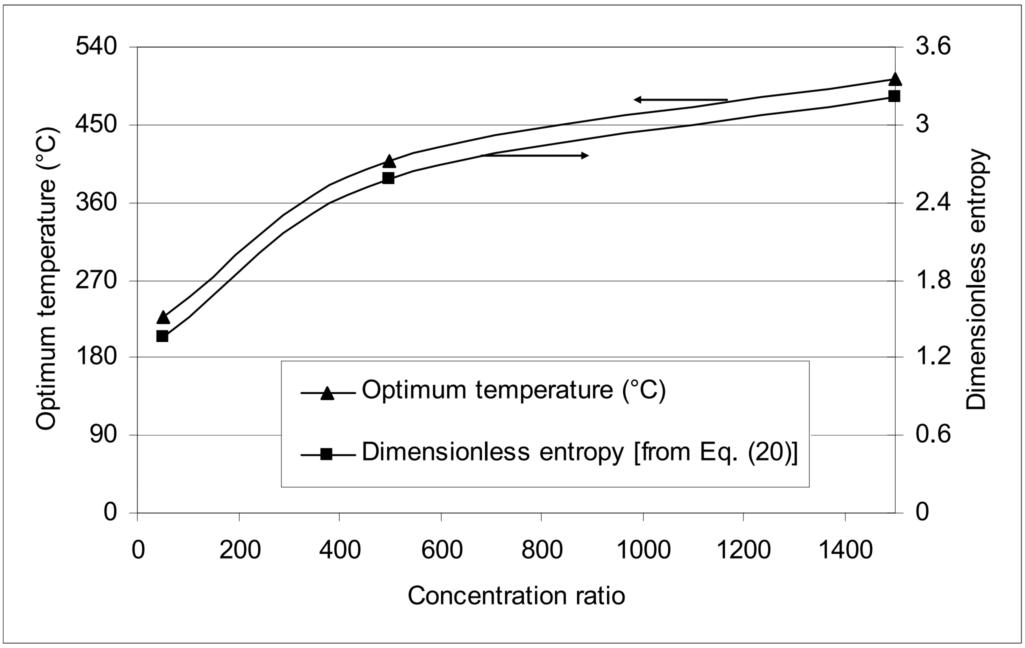

Fig. 6. Entropy generated and optimum temperatures for various types of collectors

$$q^* = U_r(T - T_o) + mc_p \frac{dT}{dx} \qquad (21)$$

where x is from 0 to L (the collector length). The generated entropy can be obtained from:

$$S_{gen} = mc_p \ln\frac{T_{out}}{T_{in}} - \frac{Q^*}{T_*} + \frac{Q_o}{T_o} \qquad (22)$$

From an overall energy balance, the total heat loss is:

$$Q_o = Q^* - mc_p(T_{out} - T_{in}) \qquad (23)$$

Substituting Eq. (23) into Eq. (22) and performing the necessary manipulations the following relation is obtained:

$$N_s = M\left(\ln\frac{\theta_{out}}{\theta_{in}} - \theta_{out} + \theta_{in}\right) - \frac{1}{\theta_*} + 1 \qquad (24)$$

where $\theta_{out}=T_{out}/T_o$, $\theta_{in}=T_{in}/T_o$, N_s is the entropy generation number and M is the mass flow number given by:

$$N_s = \frac{S_{gen}T_o}{Q^*} \quad \text{and} \quad M = \frac{mc_pT_o}{Q^*} \qquad (25)$$

If the inlet temperature is fixed $\theta_{in}=1$, then the entropy generation rate is a function of only M and θ_{out}. These parameters are interdependent because the collector outlet temperature depends on the mass flow rate.

4. CONCLUSIONS

The entropy generation minimisation of imaging concentrating collectors is presented in this paper. As it is shown the irreversibilities of the solar collection process are associated with the heat transfer between sun and the collector and between the collector and the heat reservoir (ambient environment). The optimum collector temperatures that minimise entropy have been derived based on the collector stagnation temperatures. In all the above the irreversibilities associated with the manufacturing of the solar collectors and the materials employed in the construction, as accounted by the collector optical efficiency, have been considered.

NOMENCLATURE

A_a	Absorber area (m^2)
A_r	Receiver area (m^2)
A_f	Collector geometric factor
C	Collector concentration ratio [$=A_a/A_r$]
c_p	Specific heat at constant pressure (J/kgK)
$E_{x,in}$	Exergy in (W)
$E_{x,out}$	Exergy out (W)
m	Mass flow rate (kg/s)
n_c	Collector efficiency
n_o	Collector optical efficiency
N_s	Entropy generation number
q^*	Irradiation per unit of collector area (W/m^2)
q_o^*	Radiation falling on the receiver (W/m^2)
Q	Rate of heat transfer output (W)
Q^*	Solar radiation incident on collector (W)
Q_o	Rate of heat loss to ambient (W)
s	Specific entropy (J/kgK)
S_{gen}	Generated entropy (J/K)
T	Absolute temperature (K)
T_r	Receiver temperature (K)
T_o	Ambient temperature (K)
T_s	Apparent black body temperature of the sun (~6000K)
T_*	Apparent sun temperature as an exergy source (~4500K)
U_r	Receiver-ambient heat transfer coefficient based on A_r (W/m^2K)

Greek

α_α	Absorber absorptance
β	Incidence angle (°)
γ	Collector intercept factor
ρ	Density (kg/m^3)
ρ_m	Mirror reflectance
τ_α	Absorber transmittance
θ	Dimensionless temperature [$=T/T_o$]

REFERENCES

Bejan A., Kearney D.W. and Kreith F. (1981) Second law analysis and synthesis of solar collector systems. *Journal of Solar Energy Engineering*. 103, 23-28.

Bejan A. (1995) *Entropy Generation Minimization*, 2nd Edition, CRC Press, Boca Raton, Chapter 9.

Bejan A. (1999) A role for exergy analysis and optimization in aircraft energy-system design, *ASME Congress and Exposition*, Nashville, Tennessee.

Bakos G.C., Adamopoulos D., Tsagas N.F. and Soursos M. (1999) Design and construction of a line-focus parabolic trough solar concentrator for electricity generation, *Proceedings of ISES Solar World Congress on CD-ROM*, Jerusalem, Israel.

Chavez J.M., Kolb G.J., Meinecke W. (1993) *Second Generation Central Receiver Technologies- A status Report*. Becker M., Klimas P.C. (Eds.), Verlag C.F. Müller GmbH, Karlsruhe, Germany.

Electric Power Research Institute (EPRI), (1997) Renewable Energy Technology Characterizations, *EPRI Topical Report,* No. TR-109496, Palo Alto, CA.

Geyer M., Lupfert E., Osuna R., Esteban A., Schiel W., Schweitzer A., Zarza E., Nava P., Langenkamp J. and Mandelberg E. (2002) Eurotrough - Parabolic trough collector developed for cost efficient solar power generation. *Proceedings of 11th Solar PACES International Symposium on Concentrated Solar Power and Chemical Energy Technologies on CD-ROM*, Zurich, Switzerland.

Kalogirou S. (1996) Parabolic trough collector system for low temperature steam generation: Design and performance characteristics, *Applied Energy,* 55, 1-19.

Kearney D. W. and Price H. W. (1992) Solar thermal plants - LUZ concept (current status of the SEGS plants). *Proceedings of the 2nd Renewable Energy Congress*. Reading UK, Vol. 2, pp. 582-588.

Koroneos C., Spachos T. and Moussiopoulos N. (2003) Exergy analysis of renewable energy sources. *Renewable Energy*, 28, 295-310.

Lupfert E., Geyer M., Schiel W., Zarza E., Gonzalez-Anguilar R.O. and Nava P. (2000) Eurotrough - A new parabolic trough collector with advanced light weight structure. *Proceedings of Solar Thermal 2000 International Conference, on CD-ROM*, Sydney, Australia.

Pilkington Solar International GmbH (1996) *Status Report on Solar Thermal Power Plants*, Köln, Germany.

Pitz-Paal R. (2002) Concentrating solar technologies-The key to renewable electricity and process heat for a wide range of applications. *Proceedings of the World Renewable Energy Congress VII on CD-ROM*, Cologne, Germany.

Petela R. (1964) Exergy of heat radiation, *ASME Journal of Heat transfer*, 68, 187.

Romero M., Buck R. and Pacheco, J.E. (2002) An update on solar central receiver systems projects and technologies, *Journal of Solar Energy Engineering*, Vol. 124, Issue 2.

Saha S.K. and Mahata D.K. (2001) Thermodynamic optimisation of solar flat plate collector. *Renewable Energy*, 23, 181-193.

Schwarzbözl P., Pitz-Paal R., Meinecke W., Buck R. (2000) Cost-optimized solar gas turbine cycles using volumetric air receiver technology, *Proceedings of the Renewable Energy for the New Millennium*, Sydney, Australia, March 8-10, pp. 171-177.

Singh N, Kaushik S.C. and Misra R.D. (2000) Exergetic analysis of a solar thermal power system. *Renewable Energy*, 19, 135-143.

Torres-Reyes E., Navarrete-Gonzalez J.J. and Ibarra-Salazar B.A. (2002) Thermodynamic method for designing dryers operated by flat-plate solar collectors. *Renewable Energy*, Vol. 26, pp. 649-660.

Yantovski E. (2000) Exergonomics in education, *Energy*, 25, 1021-1031.