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# Performance Study of Solar Organic Rankine Cycle for Rural Electrification

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**ABSTRACT**

The use of Organic Rankine Cycle (ORC) in decentralized applications is linked with the fact that this process allows using low temperature heat sources and offers an advantageous efficiency in small-scale applications. The investigation of supercritical parameters in ORC applications seems to bring promising results in decentralized energy production. This paper presents the results from the simulation of the ORC process in normal and supercritical fluid parameters and discusses the efficiency variation in various applications. Performance calculation is presented for small scale power system based on ORC, in order to investigate the potential of this technology. A low-temperature solar Rankine system utilizing R245fa as the working fluid is proposed. The system consists of parabolic trough solar collector (PTSC) array, an ORC engine, and hot vapor condenser.
**Keywords:** Rural Electrification, Heat Sources, ORC Applications.

## I. INTRODUCTION

Most of the currently installed Concentrating Solar Power (CSP) plants use a steam Rankine cycle in the power block. This technology to be competitive and involve high collector temperature requires power of a few MWe. Rankine cycles using an organic fluids instead of water, operates at a lower working temperature. This makes it applicable at small and medium power scale. This advantage makes ORC technology more economically attractive. ORC technology is economically feasible and attractive when used on a small or medium power scale. Solar ORC have been studies both theoretically and experimentally science early 1970s with reported overall efficiency 2.52% and 7%. [1].

Recent studies have tended to emphasize optimization of the overall system efficiency. Twomey *et al*. [2] evaluated a small scale solar ORC with cogeneration, where the maximum isentropic efficiency of the scroll expander was 59% but the ORC efficiency was only 3.4%. In addition to the maximum instantaneous power developed was 676 W and 2540 L/day of hot water production. Wang *et al*. [3] examined a 1.6 kWe solar ORC unit using a rolling piston expander, which had an overall efficiency of 4.2% and 3.2% using evacuated tube and flat-plate collectors, respectively. Ksayer E. [4] evaluated a solar ORC for electricity and domestic hot water production where the working fluid selected was R245fa. The predicted efficiency during the peak solar hours was 14.5%. Tchanche *et al*. [5] studied theoretically in 2 kW micro-solar ORC for desalination of sea water by reverse osmosis process. The study found that the conversion of solar energy into mechanical energy to be less than 5% using three different working fluids namely R134a, R245fa and R600a.

## II. ELEMENTARY RURAL AREAS ELECTRIFICATION

In many of the non industrialized countries, centralized approaches to provide electricity based on central station generation and electric grid. Extension (often requiring substantial subsidies) and a monopoly franchise has been the only viable option for electrification of rural areas. In fact, it has been successful enough to make electricity services such as electric light, electric motive power, and electronics essential to modern industrial society, however many rural areas, face high transmission and distribution costs that make the centralized approach prohibitively expensive, for several reasons, long distances, low densities, and low demand levels; inefficiently use of the capacity of power lines due to low population; high losses in power lines; and peak demand profiles of villages. Because of the problems of supplying grid electricity for small, scattered, peaky loads, decentralized electricity generation is becoming more attractive. With decentralized systems, the high costs of transmission and distribution networks can be avoided. Electricity is typically highly valued by local populations because of the enormous improvements in living standards that it brings. But while high - cost electricity may be acceptable for satisfying basic needs in households and for some agricultural and cottage industry applications, lower costs are needed to attract greater job - generating industrial base to rural areas [6].

The alternative in decentralized approach for ruralelectri fication is the use of technologies based on renewable energy in stand-alone off-grid systems. This will identifies several advantages of decentralized renewable energy technologies over grid extension:

- ✓ They can be located closer to the demands so distribution and transmission cost and consequently energy and capacity loss are reduced.
- ✓ They avoid the need for high cost utility generation.
- ✓ They do not require fuel; their operations independent from fuel supply availability.
- ✓ They create more employment, especially local jobs.
- ✓ They are clean.

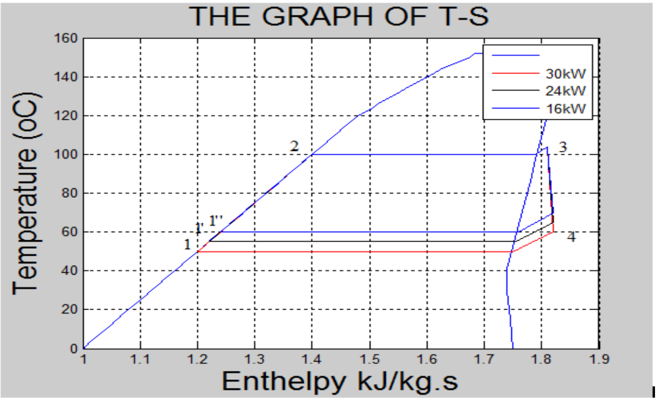


**Figure 1:** The graph of T-S chart of the unit

In these types of systems the turbine output power changes as the function of the saturated vapor temperature at the evaporator. Figure (1) shows the T-s curve for the various saturated vapor temperature. Increased temperature cause the pressure rising. For the same turbine output power, the pressure and temperature at turbine exit simultaneously increase for the increase inlet temperature. When the saturated vapor temperature reaches the critical temperature, the input power for the evaporating will reduced. Therefore the specific input power will decrease with the increased temperature. The low temperature at turbine inlet has the advantage on the system efficiency. However the low temperature requires more work on the condenser. This causes troubles when the system is operating in hot weather. So the turbine configuration must be designed based on the saturated vapor temperature 100$^{o}$C. For the same turbine inlet pressure and temperature, the exit temperature varies with the turbine power. Both increase with increase of the turbine power. Therefore, the turbine of designed max power can be operated at different off power as required by reduce the mass flow rate.

In this work, an alternative technology is proposed, consisting in the coupling between a low-cost solar thermal concentrator and an ORC engine. The aim of the technology is to integrate within a distributed generation framework to provide rural areas of the country with a micro-grid platform that can be manufactured and assembled locally and can replace or supplement Diesel generators in off grid areas, by generating clean power at a lower levelized cost. Currently, there is a wide range of viable and cost-competitive renewable energy alternatives that can be powered by solar energy. Among the different forms of solar energy conversion, the solar organic Rankine cycle system is a good option for meeting the demands for rural electrification in remote villages. ORC technology is similar to the conventional steam Rankine cycle but instead of water, its working fluid is either pure or a mixture of organic compounds.

## III. THE PROPOSED LOW-TEMPERATURE SOLAR RANKINE CYCLE SYSTEM

The use of solar energy for generating electricity on a micro- and small-scale using organic liquids as the working fluid in the solar Rankine cycle system is important and is expected to be popular in developing countries. Before powering rural communities and villager's homes using ORC technology, several practical challenges should be addressed. First, the cost of the ORC system should be competitive with other alternative rural electrification technologies, such as photovoltaic's, micro-hydro plants and small scale diesel plants. Other challenges include the development of compact units with a leakage free expansion device that has acceptable overall efficiency. In addition, the system should be lubrication free, easy to control, robust and reliable in different climatic and geographic regions.

Out of many solar technologies for electrical power generation is the solar thermal system using solar collectors. They have been divided into two main types:

1. Flat plate collectors
2. Concentrators' collectors

The flat plate collector is most popular solar collector. Their construction and operation are simple. They are applied for low temperature solar systems up to 125 $^{o}$C. Flat plate collectors have the advantage of absorbing both the direct and diffused solar energy. They are seldom tracked to follow the sun's daily path across the sky. However their fixed mounting usually provide a tilt of the site latitude angle toward the equator to minimize the angle between the sun's rays and the surface at noontime.

When higher temperatures are required, concentrating solar collectors are used where solar energy falling on a large reflective surface is reflected onto a smaller area before it is converted into heat. Therefore the absorbing surface can attain higher temperatures. Most concentrating collectors can only concentrate the direct insolation from the sun. Therefore must follow the sun's path across the sky. Four types of solar concentrator are in common uses which are:

1. Parabolic trough
2. Parabolic dish
3. Central receiver
4. Fresnel lenses

Parabolic trough technology is currently the most proven solar thermal electrical technology.

## IV. PARABOLIC TROUGH GEOMETRY

A parabola is the locus of a point that moves so that its distance from a fixed line and a fixed point are equal, (Figure 2), where the fixed line is called the directrix, and the fixed point the focus. The length FR equals the length RD. The line perpendicular to the directrix and passing through the focus F is called the axis of the parabola. The parabola intersects its axis at point V called the vertex which is exactly midway between the focus and the directrix, (FV=VO).

The surface formed by moving a parabola along the axis normal to a plane is called a parabolic cylinder and are often called parabolic trough for solar concentrators with this type of reflecting surface, sometimes called line focus concentrators When the plane containing the axes of the parabola is aligned parallel to the rays of the sun, the rays are focused on the focal line. For a parabolic cylinder of length l and aperture a, the aperture area *Aa* is given by:

$$A_a = l.a \qquad (1)$$

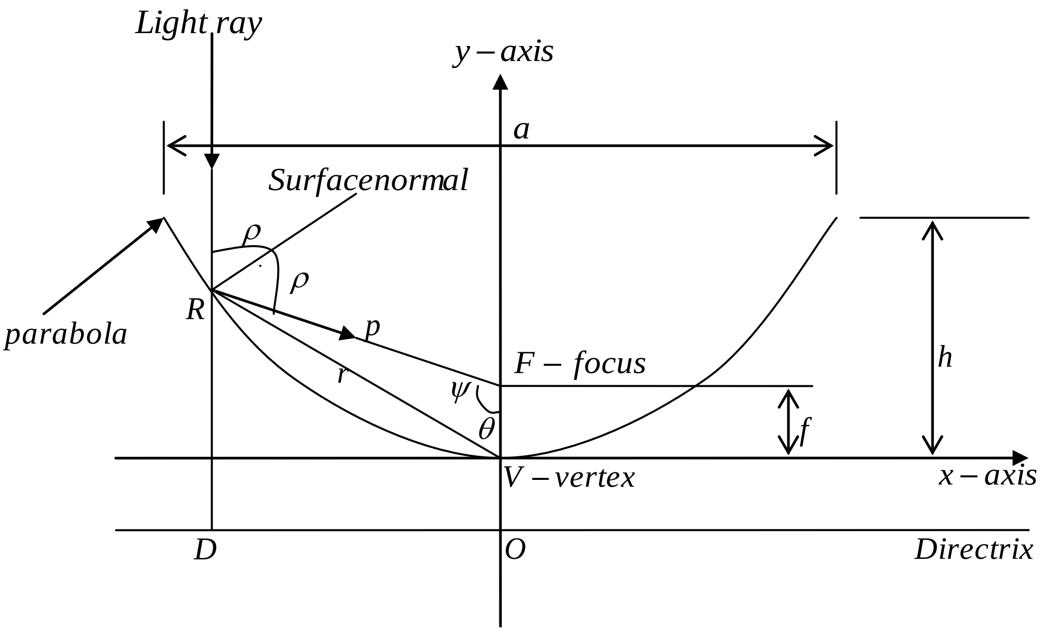


**Figure 2:** The *Parabolic* Geometry

## V. GEOMETRICAL DATA FOR THE TROUGH COLLECTOR

The parabolic trough collector design depends geometrically on:

The aperture width a
The length of the collector l
The rim angle $\psi_{rim}$
The parabolic trough depth h
The focal length f

The geometrical relation for these parameters can be found as follows:

From (Figure 2), if the origin is taken at the vertex V and the y-axis along the axis of the parabola, the equation of the parabola in a Cartesian coordinate is:

$$x^2 = 4fy \quad (2)$$

Where f is the focal length, (the distance VF from the vertex to the focus).The angle ψ measured from the line VF and the parabolic radius p (the distance from the focus F to the curve) is found by:

$$r = \frac{2f}{1+\cos\psi} \quad (3)$$

The parabolic shape has a property that, for any line parallel to the axis of the parabola, the angle ρ between it and the surface normal is equal to the angle between the normal and a line to the focal point. The solar radiation arrives at the earth is essentially parallel rays. From small surface reflection law, the angle of reflection equals the angle of incident (Snell's law)[5]. Therefore all radiation parallel to the axis of the parabola will be reflected to a single point F which is the focus. From the geometry of the parabola, (Figure 2):

$$\psi = 2\rho \quad (4)$$

If the point R is at the aperture end of the parabola, the angle ψ is known as the rim angle $\psi_{rim}$ or $\psi_{max}$,, which is the ratio of focal length to the aperture diameter f/a.

Once a specific portion of the parabolic curve has been selected, the height of the curve, h may be defined as the maximum distance from the vertex to a line drown across the aperture of the parabola. From equation (2), the height of the parabola in terms of the focal length and aperture diameter is:

(a /2 )2 = 4f h (5)

Or:

$$h = \frac{a^2}{16f} \quad (6)$$

As ψ varies from 0 to ψrim, r increases from f to R

$A_a$ = (a-d) l (7)

$A_r$ = π d l (8)

Where: -

$A_a$ = the effective aperture area of the collector = $m^2$

$A_r$ = the receiver area = $m^2$

d = the receiver outside diameter = *m*

l = the concentrator and receiver length = *m*

From equation (3)

$$R = \frac{2f}{1+\cos\psi_{rim}} \quad (9)$$

$$a/2 = R\sin\psi_{rim} \quad (10)$$

$$R\cos\psi_{rim} = f - R \quad (11)$$

The rime angle ψrim is given by:

$$\tan\frac{\psi_{rim}}{2} = \frac{a}{4f} \quad (12)$$

## VI. DESIGN OF THE PARABOLIC TROUGH COLLECTOR

The thermal output of the parabolic trough collector depends on the absorbed solar radiation incident on the collector. The solar intensity incident on the reflector surface of the trough collector is reflected on the heat collecting element (HCE). Its value is a function of optical efficiency ηo.

$$I_c = \eta_0 . I_b \quad W/m^2 \quad (13)$$

The absorbed heat varies with the direct solar irradiation $I_b$, the effective reflectors area $A_a$, the optical efficiency $\eta_o$, the collector cleanliness factor $f_c$ and the incident angle modifier *K*.

$$Q_{abs} = I_b . A_a . \eta_o . K . f_c \quad (14)$$

Part of this absorbed radiation is loss as heat losses of the collector. Therefore the rate of useful energy leaving the absorber

$$\dot{Q}_{opt} = \dot{Q}_{abs} - \dot{Q}_{los} \quad (15)$$

The solar energy collection efficiency $\eta_c$ is defined as the ratio of the rate of the useful thermal energy leaving the collector to the useable solar irradiance falling on the aperture area.

$$\eta_c = \frac{\dot{Q}_{usf}}{Q_{inc}} \quad (16)$$

The incident solar resource is:

$$\dot{Q}_{inc} = I_b . A_a \quad (17)$$

Where: -

- $I_b$ = solar irradiance entering the collector aperture - $W/m^2$.
- $A_a$ = aperture area of the collector - $m^2$

This solar irradiance as it passes from the aperture of the collector to the absorber is reduced by a number of losses depend on the type of collector. The most important loss is the optical losses. The loss mechanisms that reduce the amount of solar energy that is incident on the receiver of the collector aperture area are: imperfect reflectance, imperfect geometry, imperfect transmission, and imperfect absorption.

The optical efficiency of a collector $\eta_o$ could be identified by:

$$\eta_o = \frac{\dot{Q}_{opt}}{\dot{Q}_{inc}} = \Gamma\tau\alpha\rho \quad (18)$$

From equation (12), optical energy reaching the absorber or receiver is:

$$\dot{Q}_{opt} = \Gamma\rho\tau\alpha I_b A_a \quad (19)$$

Where: -

- $\Gamma$ = fraction of reflected energy entering or impinge on receiver
- $\rho$ = reflectance of the reflecting surface
- $\tau$ = transmittance of any intermediate glass or plastic cover sheets or windows
- $\alpha$ = absorptance of absorber or receiver surface

The rate of useful energy output from the thermal collectors is the heat addition to a heat transfer fluid. The incoming irradiance falling on the collector aperture $I_b$ multiplied by the collector aperture area represents the maximum amount of solar energy that could be captured by the collector.

Once the net optical energy reaches the absorber or receiver of the collector, it raises the temperature above the ambient temperature. This starts a process of heat losses to the surrounding. These loss mechanisms are convection, radiation, and conduction dependent on the collector's type.

$$Q_{los} = Q_{los.conv} + Q_{los.rad} + Q_{los.cond} \quad (20)$$

The energy balance on a solar collector absorber or receiver $Q_{useful}$ is

$$\dot{Q}_{useful} = \dot{Q}_{opt} - \dot{Q}_{los} \quad (21)$$

Where: -

$\dot{Q}_{useful}$= rate of useful energy leaving the absorber -$W$

$\dot{Q}_{opt}$ = rate of optical radiation incident on the absorber -$W$

$\dot{Q}_{los}$ = rate of thermal energy loss from the absorber -$W$

The useful energy is the rate of thermal energy leaving the collector, usually described in term of the rate of energy being added to a heat transfer fluid passing through a receiver or absorber.

$$Q_{useful} = \dot{m}c_p(T_{out} - T_{in}) \quad (22)$$

Where: -

- $\dot{m}$ = Mass flow rate of the heat transfer fluid - $kg/s$
- $c_p$ = Specific heat of the heat transfer fluid $J/kg\text{-}K$
- $T_{out}$ = Temperature of the heat transfer fluid leaving the absorber- $K$
- $T_{in}$ = Temperature of the heat transfer fluid entering the absorber -K

Since heat loss increases with temperature, the balance heat removal by the heat transfer fluid and heat loss defines the design operating temperature of the collector.

## VII. CONVECTION LOSS

The convective heat loss of a solar collector receiver is given as [7]:

$$\dot{Q}_{con} = h_c A(T_r - T_a) \quad (23)$$

Where: -

- $h_c$ = Average overall convective heat transfer coefficient - $W/m^2\text{-}K$
- $A_r$ = Surface area of the receiver or absorber - $m^2$
- $T_r$ = Average temperature of the receiver - $^oK$
- $T_a$ = Ambient air temperature -$^oK$

For parabolic trough collectors the average temperature of the receiver is not fixed. For these imperfections, the convective heat loss is considered as being proportional to the surface area and difference between some average temperature and ambient temperature. A convective heat loss is the major heat loss term for most solar collectors. Therefore, many systems for reducing this term of loss have been invented.

## VIII. RADIATION LOSS

Heat transfer by radiation between two surfaces is a function of emission $\varepsilon$, the Stephan – Boltzmann constant $\sigma$, and the temperatures of the surfaces $T_r$. The rate of radiation heat loss is proportional to the emitance of the surface and the difference in temperature to the fourth power [7].

$$\dot{Q}_{rad} = \varepsilon.\sigma.A_r.(T_r^{\,4} - T_a^{\,4}) \qquad (24)$$

Where: -

$\varepsilon$ = emittance of the absorber surface

$\sigma$ = the Stefan- Boltzman constant ($5.67x10^{-8}\ W/m^2K^4$)

A category of surfaces called selective surfaces has been developed by the solar collector’s designers to optimize these parameters for the collection of energy. To maximize the useful heat collected by a solar collector, the absorber or receiver of a solar collector should have a high absorptance and low emittance.

## IX. CONDUCTION LOSS

This is generally described in terms of material constants, the thickness of the material and the cross section area.

$$\dot{Q}_{cond} = k.\Delta x.A_r(T_r - T_a) \qquad (25)$$

Where: -

$k$ = equivalent average conductance – $W/m.^oK$

$\Delta x$ = the average thickness of the insulating material – $m$

In solar systems conduction loss is usually small compared to convection and radiation losses and therefore is combined with convection loss term in most analysis. Therefore the heat energy is lost from the HCE through convection and radiation.

For parabolic trough collectors there is a fourth mode of mechanism to be considered called the end loss. This is the fraction of energy being reflected from the trough that falls beyond the receiver. This loss is proportional to the angle of incidence.

## X. THERMAL ENERGY BALANCE

For an energy balance of the receiver or absorber, the combined of the equations of the three modes of losses, the rate of thermal energy loss from collector's receiver is: -

$$\dot{Q}_{loss} = \dot{Q}_{conv} + \dot{Q}_{rad} \qquad (26)$$

The heat loss from a linear receiver is the heat loss rate from the outside surface of the tube. This includes the sum of the convection and radiation losses. From equations (22), (24), and (25), the heat loss rate from the outside surface of the receiver's tube is:-

$$Q_{los} = h_c A_r (T_r - T_a) + \sigma\varepsilon F_{ra} A_r (T_r^{\,4} - T_a^{\,4}) = UA_r (T_r - T_a) \qquad (27)$$

Where $F_{ra}$= radiation shape factor and U the overall heat transfer coefficient. Therefore the thermal output of the collector $Q_{usf}$ is;

$$Q_{usf} = F_R \{\eta_o A_a I_b - UA_r (T_r - T_a)\} \qquad (28)$$

The selected trough for the proposed system is the model designed and tested by the author of the below model equation (28). [7].

$$\eta_{th} = -7.8576\left(\frac{t_i - t_a}{I_b}\right) + 0.6508 \qquad (29)$$

For our proposed system we chose a low cost parabolic trough module that was designed and tested at the mechanical department, University of Khartoum for PHD program by the author at [7]. The trough collectors use stainless steel reflective sheeting operating at 150°C. At a 150°C outlet temperature, a 57% thermal efficiency can be obtained at previous test results. The important feature of the collectors is the use of readily available materials and adaptability to manufacturing methods suitable to remote worksite construction. Table [1] gives geometrical details of the collector. The trough unit is shown as photo in Fig. 3 where the performance curve of a typical parabolic trough collector is given in (Figure 4).

**Table 1:** Geometrical data of the parabolic trough model

| | |
|---|---|
| Length | 5m |
| Aperture | 2.35m |

| | |
|---|---|
| Rim angle | 72 $^o$ |
| Focal length | 0.78m |
| Receiver diameter | 4.67cm |
| Geometrical C.R | 50.3 |
| The concentrator height | 0.42m |

**Figure 3.** The developed trough model

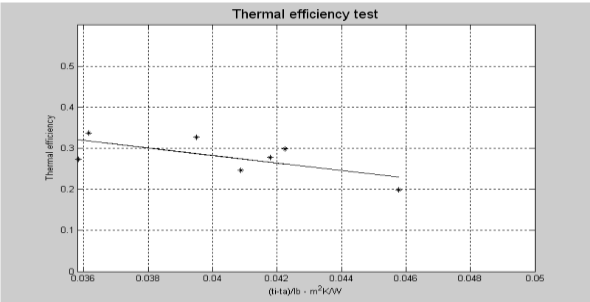


*Figure* 4. *Thermal efficiencytest ofthe troughunit*

## XI. DESCRIPTION OF ORC SYSTEM AND THE CYCLE ANALYSIS

The refrigerant, R245fa, which is non-flammable, non-toxic with zero ozone depletion potential, is to be use as the working fluid for the ORC. The working principle of the ORC system is as follows:

The working fluid passes through the trough collector and extracts the heat from the receiver (heat source), which is produced by reflected solar radiation. The working fluid in its superheated form is directed to the expander for useful work. The expanded fluid after leaving the expander is cooled by cooling water operated from a chiller. The refrigerant, R245fa, which is a saturated liquid, is pumped back to the evaporator to begin its cycle. This system can be applied as a low-temperature heat source (100–150 °C), which is obtained from a solar collector.

The total enthalpy $h_i$ at the turbine inlet can be obtained from the total pressure and temperature, where the static enthalpy *h* depends on the velocity at the duct. From the decided output power *P* based on the available heat source, the cycle analysis can be conduct. However the cycle could be operated at the condition of the output power less than the decided power P when the amount of available heat source is reduced, where the turbine to be operated at off-design point. Therefore, the turbine should be designed to be impulse tube turbine which works well in partial admission.

When the pressure loss within the nozzle has considerable value of nozzle efficiency, then:

$$h_{t2} = h_2 + \frac{V_2}{2\eta_N} \qquad (30)$$

For a nozzle of exit diameter $d_i$, the mass flow rate

$$\dot{m}_n = \rho V \frac{\pi d_2^2}{4} \qquad (31)$$

The total enthalpy $h_{fs}$ at the turbine exit is obtained as:

$$h_{t3} = h_{t1} - \frac{\Delta h_t}{\eta_{t-t}} \qquad (32)$$

In order to operate the turbine, the turbine should be pressurized $P_{pump}$ power is evaluated with the pump efficiency $\eta_{pump}$.

$$\eta_{pump} = \frac{P - P_{pump}}{h_{it} - h_6} \qquad (33)$$

The system can be illustrated schematically in (Figure 5). The cycle is entirely in the sub-critical region of the T-S chart utilizing phase change heat transfer processes for both energy addition and rejection. The cycle consists of an evaporator (4-1) in which the energy from the solar collector source is transferred to the ORC working fluid. The fluid leaves the evaporator in the dry saturated condition and enters an expansion device (1-2). The expansion process drives an electrical generator. On leaving the expander the fluid is fully condensed (2-3) leaving the condenser as a low temperature, low pressure liquid. It is then compressed to the evaporator pressure and the cycle is repeated. The cycle can also be shown on a T-S diagram, (Figure 6).

The analysis of the cycle consists of applying mass and energy balances to each of the processes mentioned above. By choosing each component as a control volume, each process in the system cycle can be written as follows:

Process 1 -2 is the actual expansion of the working fluid through the turbine: $W = (h_1 - h_2)\eta_m \cdot \eta_s$

Process 2-3 is the condensation process which occurs within counter flow heat exchanger using cooling water so the heat transferred to the cooling water (heat out) is:

$$Q = h_2 - h_3 \qquad (34)$$

Process 3 - 4 is the work done by the working fluid pump:

$$W_p = (h_3 - h_4)/\eta_p \qquad (35)$$

Process 4 - 1 is the heating process in the evaporator where thermal heat transferred to the working fluid:

$$Q = h4 - h1 \qquad (36)$$

The net electrical power produced by the ORC unit is:

$$W_{unit} = W - W_p \qquad (37)$$

The thermal efficiency of the ORC unit is:

$$\eta_{net} = W_{unit}/Q_{Evap} \qquad (38)$$

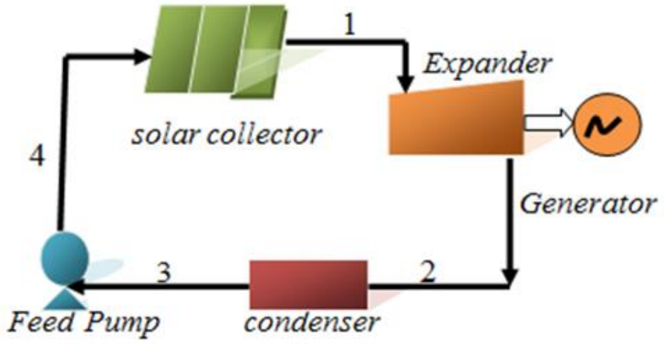


**Figure 5.** Schematic diagram of the ORC unit

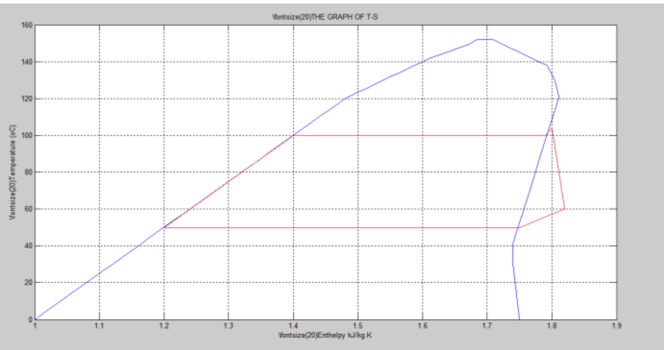

**Figure 6:** Schematic T–S diagram of the solar Rankine cycle system

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