An Organic Rankine Cycle as Technology for Smaller Concentrated Solar Powered Systems

Louis Karsten[1], Dr Jaap Hoffmann[2] and Prof Frank Dinter[3]

[1, 2,] Solar Thermal Energy Research Group (STERG), Stellenbosch University, (South Africa)

[3] Fraunhofer Institute, (Chile)

Abstract

This paper analyses and simulates an organic Rankine cycle (ORC) specifically for smaller concentrated solar powered (CSP) systems to produce solar thermal electricity (STE) in the range from 500 kW to 5 MW. The plant efficiency is optimized with the evaporating and condensing temperatures as optimizing variables. A thorough process for selecting the working fluid is presented to help the designer with this monumental task. After considering various aspects n-Pentane is chosen as working fluid for the ORC. It is also the only organic working fluid that has been successfully used in conjunction with CSP on the scale from 500kW to 5MW. The power block is simulated by modelling each component and combining them to form a complete simulation model. The results of the simulation is document and a plant efficiency of about 14.2% is achieved across the power range. The output if the ORC simulation is then compared to a steam Rankine cycle under the same operating conditions and the ORC proved to be more efficient for a power output up to 3000kW.

Keywords: Organic Rankine Cycle (ORC), concentrated solar power (CSP)

1. Introduction

Even though steam Rankine cycles are more commonly used, the usage of organic fluids as working fluid is not a new concept. In 1826, Thomas Howard patented the first concept of an engine using ether as a working fluid (Casati, 2014), and the first operational solar ORC was built by Frank Shuman in Philadelphia USA in 1907 and was rated at 2.5 kW thermal output. With the 100 m^2 collector area, direct vapour of Ether at 115°C was used to drive an irrigation pump (Shuman, 1907). Currently ORC technology is used in a variety of industries mostly because of the modularity and versatility of the technology. Industries developed around the heat sources used for the ORC where the largest by far is the geothermal energy (76.5%), secondly the heat recovery industry (12.7%), thirdly the biomass industry (10.7%) whilst the solar industry only accounts for 0.1% of the total ORC industry (Tartiere, 2016).

Various micro-scale (1 kW – 10 kW) solar ORC test facilities exist and are well documented in literature but the lack of optimized technology in the ORC solar scale up to 5 MW, leads to industry lacking confidence in this technology. A resurgence of interest in the research and development of ORC as a viable small-scale solution for electrical production has developed after the successful completion of the 1 MW APS Saguaro PT plant in Arizona, USA in 2006. The plant has 10 340 m^2 of PT collectors using thermal oil at 300°C as heat transfer fluid in the solar field. The ORC module uses n-pentane as working fluid with a 1 MW_e turbine supplied by ORMAT. A major increase in efficiency was seen with an overall solar to electrical efficiency of 12.1% at design point. To date this is still the largest operating solar ORC plant in the world and a pioneer in solar ORC as it proved the simplicity of an ORC compared to that of a conventional steam Rankine cycle and this plant even allows for unattended operation. All of which are important factors in the economic considerations and commercial acceptance of this technology (Canada et al., 2005), (Quoilin et al., 2013).

CSP plants operating with traditional steam Rankine cycles tend to become unfeasible in the small scale power range (<5 MW) and ORC's might be able to fill that gap. For lower temperature thermodynamic cycles, the ORC have advantages over the steam Rankine cycle. The most promising advantage of ORC's is that less and cheaper components are needed due to lower temperatures and cycle simplicity. Another technical advantage is that typical working fluids used in ORC cycles have higher molecular weight than water which leads to the fact that a higher mass flow rate can be achieved with an ORC for the same size of turbine. This can lead to higher turbine work output with less turbine losses (Drescher & Bruggemann, 2007). This is a major advantage seeing that the turbine is a key component in an ORC having the largest effect on cycle efficiency.

doi:10.18086/swc.2017.04.05 Available at http://proceedings.ises.org

The benefits of successfully deploying small-scale CSP goes far beyond just the climate and environmental benefits. In South Africa the mining, construction, the auto and metals and engineering sectors contribute about 20% of South Africa's gross domestic product, hence it is important to sustain these industries (SEIFSA, 2016). The ORC technology might form part of the energy solutions to these industries once an optimised small-scale CSP technology is proven hence the proposed future output of electricity generating Rankine cycle plants are shown in figure 1.

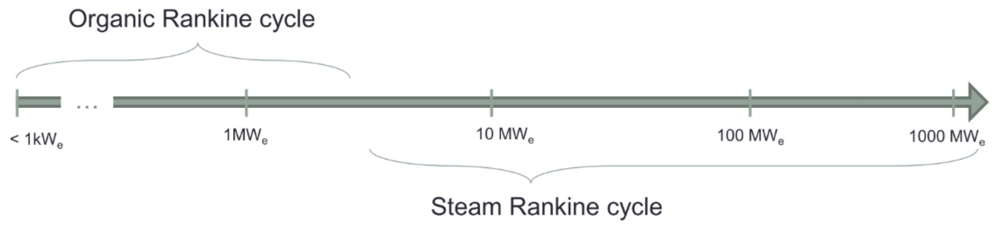

Figure 1: Proposed Thermal range of Rankine Cycles, adapted from (Dickes, 2016)

In this paper a Techno-thermodynamic design optimization is presented for an ORC using CSP as heat source across the gross electrical power output of 500 kW – 5 MW using the MATLAB environment. A specific focus is put on the working fluid selection whilst the paper elaborates on the methodology behind the design optimization, a technical analysis of the plant and components are presented and where after the design optimization results are discussed and conclusions are drawn on the feasibility of such a plant. The exact same simulation is run for a steam Rankine cycle to determine which is better suited for the current conditions.

2. Methodology

A holistic approach is taken on the power plant and its functional units namely the solar field, storage and power block. The main reasons for including storage is to prolong the operating hours of power plant and to eliminate spikes in the ORC caused by solar variations. A technological analysis is done in section 4 in order to model the whole plant consisting out of the various components in each functioning unit. By modelling each component a complete cycle simulation is achieved. For the solar field evacuated tube parabolic trough collectors are considered and Therminol 66 is used as heat transfer fluid. Therminol 66 has been developed especially for solar ORC applications by Eastman Chemicals in Italy. A Thermocline energy storage system will be incorporated in the solar field to balance out the solar variations. Regarding the power block components, it proved to be best to use an axial flow turbine for the wanted power range of 500 kW to 5 MW. A multistage centrifugal pump are commonly used in ORC's and is chosen for the current application (Macchi & Astolfi, 2017). A Plate type heat exchanger is taken for the evaporator due to their compactness and high heat transfer area. The lack of water resources in areas where CSP is implemented necessitates the use of an air cooled condenser. The final proposed ORC power plant that is used for the simulation is shown in figure 2 with the ORC T-S diagram produced by the simulation in figure 3.

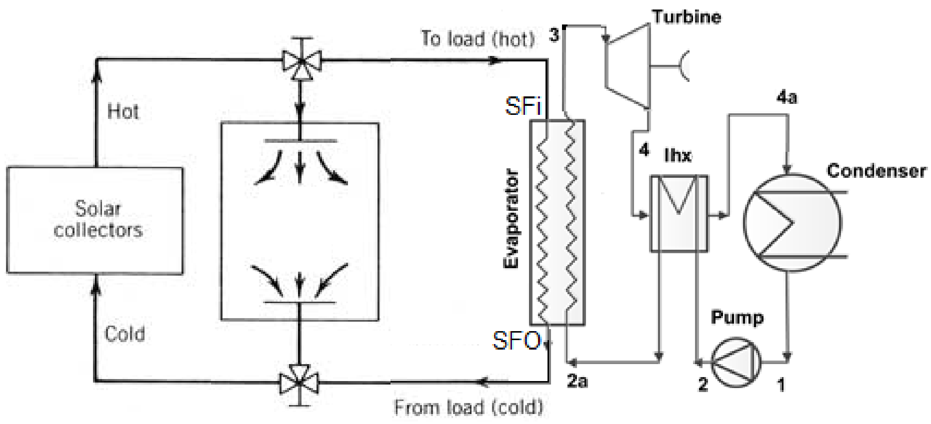

Figure 2: Final plant schematic, adapted form (Stine & Michael, 2001), (Li, G. 2016)

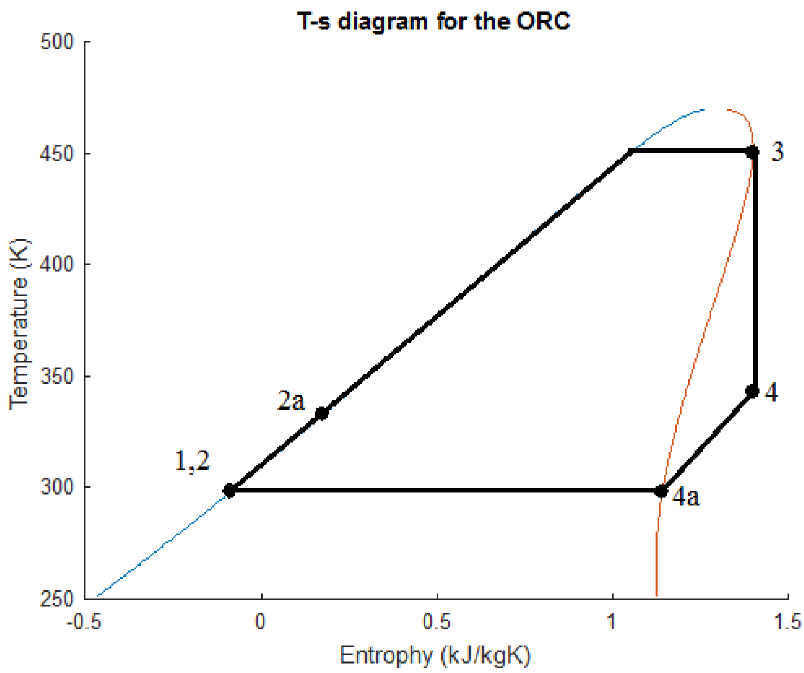

Figure 3: T-S diagram for the simulated ORC plant

A techno-thermodynamic optimization is conducted on the power plant where the total plant efficiency is used as objective function and defined as the ratio between the net power output and the maximum received thermal heat:

$$\eta_{plant} = \frac{W_{net}}{Q_{in}} \qquad \text{(eq. 1)}$$

The independent decision variables that are used as optimizing variables are the condensing and evaporating temperatures. The optimization constraints are due to the working fluid property constraints. Looking at a subcritical cycle, the chosen working fluid's critical temperature is 470K which serves as the upper limit for the cycle evaporating temperature. The maximum condensing temperature is constraint at the boiling temperature at the condensing pressure. Due to the air-cooled condenser the minimum temperature to which the working fluid can be cooled down to during condensation is limited. The incorporation of a recuperator or also referred to as an internal heat exchanger as depicted in figures 2 and 3, enables the working fluid to be cooled down to a lower temperature as the working fluid now enter the condenser at a much lower temperature. A design choice is made for the lower limit of condensation temperature and it is set at 7°C above the average ambient temperature of 20°C to compensate for condenser ineffectiveness.

For each iteration of condensing and evaporating temperatures, all model equations are solved sequentially and with the black-box optimization approach the most efficient plant design with the corresponding temperatures, is determined. The flow chart of the calculation procedure is shown in figure 4:

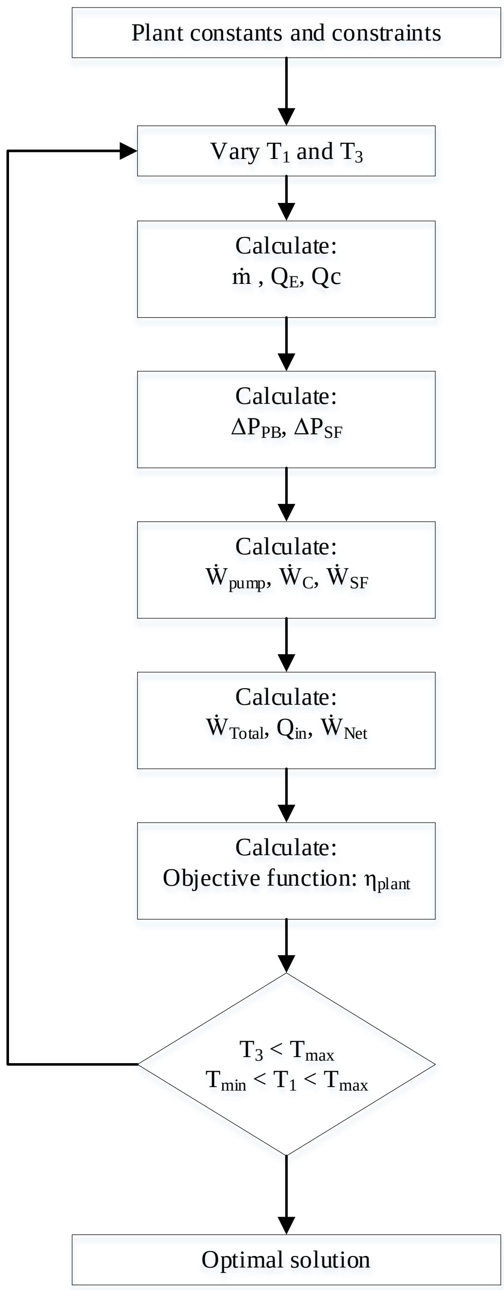

Figure 4: Simulation flow diagram

3. Choice of working fluids

The theory for an ORC is the same as to steam Rankine cycle with only the working fluid properties differing. The selection of the working fluid not only affects the efficiency of the system but also the design and sizes of the system components, stability, safety and environmental concerns and ultimately the cost of the system (Bao & Zhao, 2013). Hence, it is a very important degree of freedom for any ORC design process. Bao and Zhao continues by saying that the selection of an ORC working fluid is more complex than other thermodynamic cycles mainly due to two reasons:

- The heat type sources for ORC varies widely from 80 ^{0}C to 500 ^{0}C
- Excluding a few organic fluids whose critical temperatures are too high or too low, hundreds of fluids are available for usage including hydrocarbons, aromatic hydrocarbons, perfluorocarbons, alcohols, CFS’s, siloxanes, ethers etc.

Due to the working fluid selection being so important, many researchers have carried on the fluid screening method and this method is by far the most used method for working fluid selection in scientific literature (Bao & Zhao, 2013). The method consists of building a steady state simulation model of the proposed ORC plant and run it with different working fluids to determine the most effective fluid for the current application.

Another fluid selection guideline has been developed by (Macchi & Astolfi, 2017) to help ORC designers with this monumental task of screening hundreds of working fluids. The number of applicable fluids can be drastically decreased by considering and comparing different fluids. The obvious requirements a fluid must meet is the fluid should be:

- *Commercially available at a reasonable cost*: with the intended MW range of the proposed system, a large fluid inventory is needed and the fluid cost can mount up to a significant portion of the total plant cost.
- *Non-flammable*: Hydrocarbons are flammable working fluids.
- *Nontoxic:* ammonia is toxic but is still sometimes adopted.
- *Compatible with materials*: the working fluid must be compatible with the lubricating oils, elastomers, metals etc.
- *Environmental benign*: the two main indexes that account for fluid acceptance is the Ozone Depletion Potential (ODP) and the Global Warming Potential (GWP).

Macchi & Astolfi continues to say that it is practically impossible to fulfil all these requirements with a working fluid that is suitable for ORC applications hence ORC manufacturers must overlook some of the qualities listed above but they are still aimed for. The second list of considerations in selection a suitable working fluid regards the thermodynamic considerations. The thermodynamic and physical properties of working fluids are what differentiate working fluids the most. The relationship between working fluid properties and thermodynamic cycle performance are discussed below:

- *Thermal stability:* The thermal stability of the fluid used can limit the temperature of the heat source as fluids can chemically brake down at certain temperatures hence a high thermal stability is desired.
- *Vaporization latent heat:* In terms of work output it has been found that for the same defined temperatures, a larger unit work output is produced with working fluids with a higher vaporization latent heat (Chen et al., 2010).
- *Density:* Bao and Zhao states that a high vapour density is of key importance especially for working fluids with a low condensing pressure. A low density leads to higher volume flow rates which in turn lead to larger pressure drops in the heat exchangers and the bigger turbine sizes.
- *Specific heat:* There is no direct recorded effect of specific heat and total system power output.
- *Critical temperature:* High critical temperature has a beneficial effect on cycle performance but has the adverse effect of lower vapour densities which has a benign effect on cycle performance.
- *Boiling temperature:* When comparing fluids of the same family a higher boiling temperature leads to an increase in cycle efficiency but it is by no means an absolute criteria.
- *Freezing temperature:* It serves as a constraint but has no effect on cycle efficiency.
- *Molecular weight:* Bao and Zhao concluded that high molecular weight has a positive impact on turbine efficiency but it must be noted that fluids with a high critical pressure and high molecular weight require higher heat transfer area which increases the total plant cost.
- *Molecular complexity:* A direct link between molecular complexity and cycle performance is not possible as most properties are affected by the molecular complexity and different properties have different effects on cycle performance.
- *Viscosity:* In order to maintain low friction losses in the pipes and heat exchangers, low viscosity is desired in both the liquid and vapour phase (Bao & Zhao, 2013).

- *Conductivity:* In order to obtain a high heat transfer coefficient in the heat exchangers, a high conductivity is required (Bao & Zhao, 2013).

In the case of varying heat source temperatures, it can be more advantageous to use mixtures of working fluids rather than pure working fluids. In such a system heat is supplied to or rejected at variable temperature but at constant pressure because the boiling temperature varies during phase change and the binary mixture evaporates over a large range of temperatures (Bao & Zhao, 2013). The storage incorporated in this analysis allows the heat addition to the power block at constant temperature and only pure working fluids are considered henceforward.

Working fluids are also categorised according to their saturation curve and the categorisation disregarded the structural point of view and type of atoms. When looking at the latter two, the possible ORC working fluids can be categorised in seven main classes and Bao and Zhao pointed out typical characteristics of each after screening the fluids over a range of applications:

1. Hydrocarbons including linear (n-butane, n-pentane), branched (Isobutane, Isopentane) and aromatic hydrocarbons (Toluene, Benzene)
    - Flammability issues
    - Desirable thermodynamic properties
2. Perfluorocarbons
    - Thermodynamically undesirable
    - Extremely inert and stable
    - Extreme molecular complexity
3. Siloxanes
    - Mostly used as mixtures rather than pure fluids
    - Isobaric condensation and evaporation are not isothermal and exhibit a certain glide
4. Partially flouro-substituted straight chain hydrocarbons
    - Several zero ODP fluids exists which are of interest
5. Ethers and fluorinated ether
    - Thermodynamically undesirable
    - Flammability and toxicity issues
6. Alcohols
    - Thermodynamically undesirable
    - Soluble in water
    - Flammability issues
7. Inorganics
    - Operational problems
    - Small environmental impact
    - Inexpensive

Even though the screening process covers a large number of fluids, only a few fluids are actually used in commercial plants. Hydrocarbons are the fluids with the most desirable thermodynamic properties and the flammability issues are often carefully managed in practice by restricting the operating conditions. One of the hydrocarbon fluids, namely n-pentane, is the working fluid that is used in the 1 MW APS Saguaro PT plant in Arizona. According to Bao and Zhao no other working fluid has been used for commercial solar power plants (>500 kW) before 2013 and no new information has been published according to the author's knowledge since then. This serves as a very strong argument for the acceptance of n-Pentane as working fluid. The screening process further concluded that R134a and R245fa are also possibilities for a solar application and these fluids have been used as working fluids on micro scale (<10 kW) solar applications. Table 1 gives a summary of how n-pentane relates to the selection criteria when compared with R134a and R245fa:

Table 1: n-Pentane regarding the selection considerations

Initial selection	Availability and cost	Locally and affordably available.
	Non-flammable	NFPA 704 rating: 4; readily dispersed in air and will burn readily (Chemistry Reference, 2017).
	Non-toxic	NFPA 704 rating: 1; can cause human irritation (Chemistry Reference, 2017) .
	Compatible with materials	Compatible with all materials
	Environmental benign	Quickly evaporates and biodegrades in soil (National Refrigerants Inc., 2015)
Selection according to thermodynamic properties, (all properties are evaluated at 25°C)	Thermal stability	NFPA 704 rating: 0; very stable
	Vaporization latent heat	n-Pentane: 365 kJ/Kg R134a: 178 kJ/Kg R245fa: 197 kJ/Kg
	Density	n-Pentane: 620 kg/m^3 R134a: 1210 kg/m^3 R245fa:1339 kg/m^3
	Critical temperature	n-Pentane: 197°C R134a: 122°C R245fa: 154°C
	Molecular weight	n-Pentane: 72 R134a: 102 R245fa: 134
	Viscosity	n-Pentane: 0.217 mPa.s R134a: 12.06 mPa.s R245fa: 402.7 mPa.s
	Conductivity	n-Pentane: 0.1112 W/mK R134a: 0.013 W/mK R245fa: 0.0125 W/mK

From table 1 it can be seen that n-Pentane is the best option regarding the vaporization latent heat, viscosity and conductivity. The beneficial higher critical temperature of n-Pentane and the adverse effect thereof can be seen with the lower density. The much higher conductivity of n-Pentane caries a lot of weight as the total heat transfer area is greatly reduced resulting in smaller heat exchangers. Heat exchangers are large contributors to the total power plant cost hence smaller heat exchangers are desired. As a result n-Pentane is the best option for working fluid considering the preceding criteria hence n-Pentane is the working fluid of choice for this analysis.

4. Technical analysis

The numerical calculations were carried out for the gross power output range of 500 kW – 5 MW. Following the procedure of figure 4, the fluid properties at each state point as depicted in figures 2 and 3, are retrieved using compressed liquid, saturated liquid, saturated vapor and superheated vapor property tables for n-Pentane, (NIST, 2017). The heat and mass balance across the devices are used and the procedure follows:

The mass flow rate of the working fluid in the power block is given by:

$$\dot{m}_{PB} = \frac{W_{gross}}{(h_3 - h_4)\eta_T\eta_{gen}} \quad \text{(eq. 2)}$$

The heat supplied to the evaporator and the heat rejected by the condenser is given by:

$$Q_{PB,E} = \dot{m}_{PB}(h_3 - h_{2a}) \quad \text{(eq. 3)}$$

$$Q_{PB,C} = \dot{m}_{PB}(h_{4a} - h_1) \quad \text{(eq. 4)}$$

To calculate the pumping power of the power block pump, the pressure drops in the turbine, condenser and evaporator needs to be accounted for. They are calculated as such:

The pressure increase over the pump to overcome the turbine losses can be calculated by:

$$\Delta P_T = P_2 - P_1 \quad \text{(eq. 5)}$$

The total heat transfer area of the evaporator is given by:

$$A_e = \frac{Q_{PBe}}{U_e \Delta T_{m,E}} \quad \text{(eq. 6)}$$

Where the log mean temperature difference over the evaporator is defined as:

$$\Delta T_{m,E} = \frac{(T_{SFi} - T_3) - (T_{SFo} - T_2\ )}{\ln\left(\frac{T_{SFi} - T_3}{T_{SFo} - T_2}\right)} \quad \text{(eq. 7)}$$

The size of the evaporator is then determined by determining the number of plates to the upper integer:

$$n_E = \frac{A_E}{w_E \times l_E} \quad \text{(eq. 8)}$$

Due to the phase change in the evaporator the pressure drop across the core part of the evaporator can be a tedious procedure to calculate. By taking the pressure drop equation for single phase flow and incorporating the two phase flow effect in the multiplication factor ϑ^2, a satisfying result can be obtained with this equation (Shah & Sekulić, 2003):

$$\Delta P_{core,E} = \left(\frac{dp}{dz}\right) = f_E\left(\frac{4}{D_{h,}}\right)\left(\frac{G^2}{2\rho_E}\right)\vartheta^2 \quad \text{(eq. 9)}$$

Where:

$$\vartheta^2 = (1-x)^2 + x^2\frac{\rho_l}{\rho_g}\frac{f_g}{f_l} + \frac{3.24(x^{0.78}(1-x)^{0.24})\left(\left(\frac{\rho_l}{\rho_g}\right)^{0.91}\left(\frac{\mu_g}{\mu_l}\right)^{0.19}\left(1-\frac{\mu_g}{\mu_l}\right)^{0.7}\right)}{\left(\frac{G^2}{gD_h\rho_{avg}}\right)^{0.045}\left(\frac{G^2D_h}{\rho_{avg}}\right)^{0.035}} \quad \text{(eq. 10)}$$

The pressure losses due to the intake and outlet manifolds are evaluated by the following equation (Shah & Sekulić, 2003):

$$\Delta P_{M,E} = \frac{1{,}5G_E^2 n_{p,E}}{2\rho_2} \quad \text{(eq. 11)}$$

The total pressure loss over the evaporator for the working fluid side is then after neglecting the gravitational effects due to fact that the tubes are horizontal:

$$\Delta P_E = \Delta P_{core,E} + \Delta P_{M,E} \quad \text{(eq. 12)}$$

The total pressure loss over the condenser, ΔP_c , is evaluated exactly the same way as that for the evaporator taking note that the geometrical features of the air-cooled condenser differs.

The work required for the power block pump is then:

$$W_{PB,pump} = \dot{V}(\Delta P_T + \Delta P_E + \Delta P_c) \quad \text{(eq. 13)}$$

To avoid the integrate and iterative design process of an air-cooled condenser in the scope of cycle optimization such as this, a general relationship between the work required by the air-cooled condenser fan and the turbine power output where (O'Donovan, 2013):

$$W_{c,F} = 0.02 \times W_{gross} \quad \text{(eq. 14)}$$

This relationship was confirmed by taking a typical air-cooled condenser fan that is used in a similar application with similar heat rejection requirements than the 5 MW case. The fan curve was available and by submitting the data in the power block simulation a relationship of about 4% was calculated between fan work and turbine power output. The relationship will be used with the more conservative relationship of 4% and by using a relationship the simulation can be used in a modular way across the power output range.

The next step is to calculate the work required by the solar field pump. After calculating the mass flow rate in the solar field with the following equation, the pressure drop in the evaporator, $\Delta P_{SF,e}$ can be calculated as in the case of the evaporator working fluid side.

$$\dot{m}_{SF} = \frac{Q_{PB,E}}{Cp_{SF}(T_{SFi} - T_{SFo})} \quad \text{(eq. 15)}$$

Neglecting minor losses, the frictional pressure drop of the rest of the solar field can be calculated with the following:

$$\Delta P_{SF,fr} = f \frac{L}{D_{SF,p}} \frac{\rho V_{SF}^2}{2} \quad \text{(eq. 16)}$$

The total pressure drop and maximum required work (during day time) of the solar field pump is given below:

$$\Delta P_{SF} = \Delta P_{SF,e} + \Delta P_{SF,fr} \quad \text{(eq. 17)}$$

$$W_{SF,pump} = \dot{V}_{SF}(\Delta P_{SF}) \quad \text{(eq. 18)}$$

The net power output of the plant is then calculated as given below:

$$W_{Net} = W_{gross} - (W_{PB,pump} + W_{C,F} + W_{SF,pump}) \quad \text{(eq. 19)}$$

Following the objective function can be calculated as described previously given that:

$$Q_{in} = \frac{Q_{PB,E}}{\eta_{trough}} \quad \text{(eq. 20)}$$

5. Results and discussion

A techno-thermodynamic optimization was set up for a solar ORC plant where the plant efficiency was used as the objective function. For the 1 MW gross power output case, the power plant reached an optimal objective function at a condensing temperature, T_C = 300K and evaporating temperature T_E = 450K. These values fall within the constraints set by the simulation which necessitated the evaporating temperature below the critical temperature of 470K of n-Pentane, and keeping the condenser outlet temperature above an ambient temperature of 298K. The evaporating pressure reached a value of 2.45 MPa and the condensing pressure reached a value of 67 kPa. The exact same simulation was run under the same conditions for steam Rankine cycle for comparison sake. The resulting plant efficiencies from the objective function for the whole simulated power range can be seen in table 2:

Table 2: Resulting optimized plant efficiencies with net power output

Turbine power output (kW)	Net ORC plant power output (kW)	Optimized ORC plant efficiency (%)	Optimized steam plant efficiency (%)
500	458	14.26	11.10
1000	916	14.26	11.82
2000	1 831	14.25	13.77
3000	2762	14.25	14.17
4000	3660	14.24	14.76
5000	4598	14.24	15.75

As can be seen from table 2, the ORC performs much better at the lower temperature range. The main reason is that steam turbines are very inefficient at low power outputs leading to the whole cycle efficiency being lower. As the turbine output increases above 4000kW, it can be concluded that a steam Rankine cycle would be a better choice. Furthermore the ORC turbine efficiencies remain relatively constant for different power outputs, hence the whole plant efficiency remains more or less constant with change in power output. The turbine and working fluid has the largest effect of all the components on the cycle efficiency. Typically the average temperature at which heat is added in the evaporator must be increased and the average temperature at which heat is being rejected in the condenser needs to be decreased in order to increase cycle efficiency. The first mentioned is directly linked to the turbine inlet temperature and the effect of the turbine inlet temperature on cycle efficiency can be seen in figure 5.

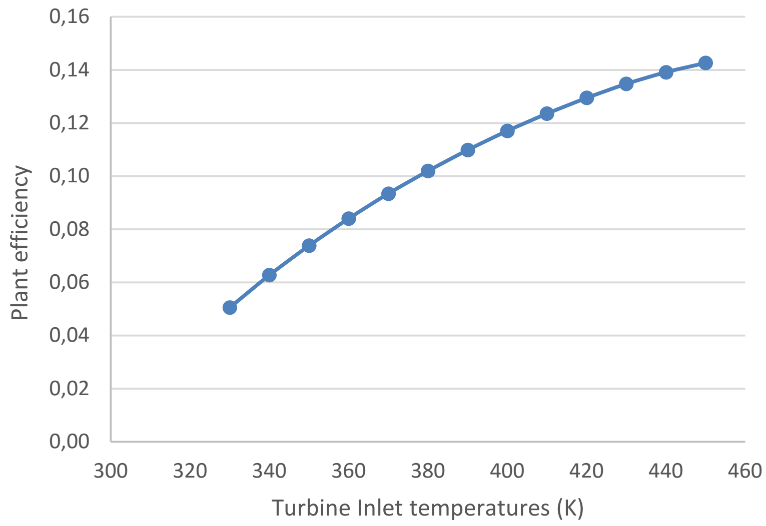

Figure 5: The influence of turbine Inlet temperature on plant efficiency for the 1 MW case

6. Conclusion

The paper optimized an ORC coupled with CSP and obtained a plant efficiency of about 14.2% across the power range of 500 kW – 5 MW. The efficiency correlates with an existing plant. The simulated plant operated at a maximum temperature and pressure of 450K and 2.45 MPa respectively. This is significantly lower than that of traditional Rankine cycles resulting in components needing to have less resilience. This in effect has a great decreasing effect on plant cost and if accurate cost relations can be found one can conclude on the economical competiveness of an ORC. When the steam Rankine cycle was simulated under the same conditions and temperatures than the ORC, the ORC proved to be more efficient up to 3000kW power output. However the lack of optimized technology in the solar ORC field leads to industry not widely accepting the technology as a feasible solution yet but this paper concludes that further research in this topic is justified and that the solar ORC technology is deemed to reach maturity in the future.

7. References

Bao, J., & Zhao, L. (2013). A review of working fluid and expander selections for organic Rankine cycle. Renewable and Sustainable Energy Reviews, 24, 325–342

Canada, S., Brosseau, D., Kolb, G., Moore, L., Cable, R., & Price, H. (2005). Status of APS 1-Mwe Parabolic Trough Project, Size: 5 pp. Retrieved from www.osti.gov/servlets/purl/882807-d8b9mN

Casati, E. I. M. (2014). New Concepts for Organic Rankine Cycle Power Systems. Energy *Procedia*. Delft University of Technology. https://doi.org/10.1016/j.egypro.2014.10.220

Chen, H., Goswami, D. Y., & Stefanakos, E. K. (2010). A review of thermodynamic cycles and working fluids for the conversion of low-grade heat. Renewable and Sustainable Energy Reviews, 14(9), 3059–3067

Dickes, R. (2016). Solar-based ORC power systems. In *ORC-Plus Workshop*

Drescher, U. & Bruggemann, D., 2007. Fluid selection for the Organic Rankine Cycle (ORC) in biomass power and heat plants. *Applied Thermal Engineering*, Volume 27, pp. 223-228.

SEIFSA, 2016. High electricity price increase will have a crippling effect of the already embattled metals and engineering sector, s.l.: s.n.

Shah, R. K. (Ramesh K. ., & Sekulić, D. P. (2003). Fundamentals of heat exchanger design. John Wiley & Sons

Shuman, F., 1907. The Direct Acting Solar Engine: the Prime Mover of the Immediate Future. s.l.:s.n.

Tartiere, T. (2016). Analysis of the Organic Rankine Cycle market. Retrieved November 9, 2016, from http://orc-world-map.org/analysis.html

Quoilin, S., Broek, M. Van Den, Declaye, S., Dewallef, P., & Lemort, V. (2013). Techno-economic survey of organic rankine cycle (ORC) systems. *Renewable and Sustainable Energy Reviews*, *22*, 168–186

8. Appendix: Nomenclature

Symbol	Quantity	Unit
A	Area	m^2
Cp	Specific heat	$J\ kg^{-1}\ K^{-1}$
D_h	Hydraulic diameter	m
f	Friction factor	
g	Gravity	$m\ s^{-2}$
G	Mass velocity	$kg\ m^{-2}\ s^{-1}$
l	Length	m
$\dot{m}$	Mass flow rate	$kg\ s^{-1}$
n	Number of plates	
P	Pressure	kPa
S	Entropy	$kJ\ kg^{-1}\ K^{-1}$
T	Temperature	K
U	Overall heat transfer coefficient	$W\ m^{-2}\ K^{-1}$
$\dot{V}$	Volume flow rate	$m^3 s$
w	Width	m
W	Power	W

Subscript	Definition
C	Condenser
E	Evaporator
F	Fan
fr	Friction
gen	Generator
l	Liquid
M	Manifolds
m	Log mean
PB	Power Block
SF	Solar Field
SFi	Evaporator in
SFo	Evaporator out
T	Turbine
v	Vapour
$1,2,2a,3,4,4a$	State Points

Greek symbol	Definition	Unit
η	Efficiency	
μ	Viscosity	Pa.s
ρ	Density	$kg\ m^{-3}$