THERMO-ECONOMIC ANALYSIS OF A MIXTURE OF RC-318 AND PENTANE AS A WORKING FLUID IN A HIGH TEMPERATURE ORC

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ABSTRACT

In this paper, the potential of realizing a low cost power block in concentrated solar power applications is investigated. A conventional steam Rankine cycle operates under vacuum pressures in the condenser which manifests in considerably high volumetric flow rates at the expander exhaust. This calls for large equipment size and makes the power block expensive. This paper presents the idea of substituting the conventional power block by a high temperature (~ 350 °C) and equally efficient organic Rankine cycle (ORC) which works on a mixture of pentane and RC-318 (70 % and 30 % on a molar basis). A comparison is made on a one-to-one basis with identical operating conditions for a given solar field equipment. It is found that the proposed mixture has an initial investment cost ~ 25 % lower than steam in the Rankine cycle due to its significantly lower volumetric flow rates.

1. INTRODUCTION

As demand for energy increases across the world along with proliferation of awareness about humankind's responsibility towards nature, major advancements have been made in the search for cleaner sources of energy. Amongst the renewable energy resources, a lot of impetus has been put on concentrated solar thermal (CST) and as a consequence, it has witnessed great technological leaps in the past decade.

Conventional steam Rankine cycles used in fossil fuel based power plants have been directly adapted to CST since it has evolved as an efficient power block over the last two centuries. Despite this, steam suffers from low densities at the expander exhaust resulting in prodigious size of the equipment. Furthermore, issues like wet expansion render the turbine systems complicated. These limitations have led to other cycles being suggested for CST technology. Garg et al. (2013a, 2014) investigated and proposed CO_2 cycles for solar thermal applications, suggesting that the compactness of such cycles could be explored as means of lowering the levelized cost of energy (LCOE). Contemporarily, organic fluids have also been suggested to supersede steam as a working fluid and reviewed by Tchanche et al. (2011), albeit for low temperature heat sources such as waste heat recovery or geothermal. Significant amount of work by Quoilin exists on thermo-economic evaluation of organic

Rankine cycles (Quoilin et al. 2009, Quoilin et al. 2011a and Quoilin 2011b). Furthermore, different ORC working fluids have been investigated by Mikielewicz and Mikielewicz (2010), Wang et al. (2011) and Garg et al. (2013c). Low Carnot efficiencies of these cycles when coupled to a comparatively expensive solar field result in higher investment costs and encourage investigation of high temperature ORCs. Buoyed by this motivation, present paper suggests a new working fluid for a high temperature ORC which can compete with the solar steam Rankine cycle on thermo-economic platforms.

2. SOLAR-ORGANIC RANKINE CYCLE DETAILS

Figure 1 shows a schematic of a solar-ORC power block which consists of two closed loops, namely a) heat transfer fluid (HTF) loop and b) working fluid loop. Parabolic troughs concentrate sunlight onto tube-like receivers carrying cold HTF from a (Thermal Energy Storage) TES tank where it gets heated up and makes its way back to the TES tank. The heat from the hot HTF is thus stored in the TES tank and is then transferred to the working fluid in a heat exchanger to produce pressuriz ed vapor, which is further used to drive turbines to generate power.

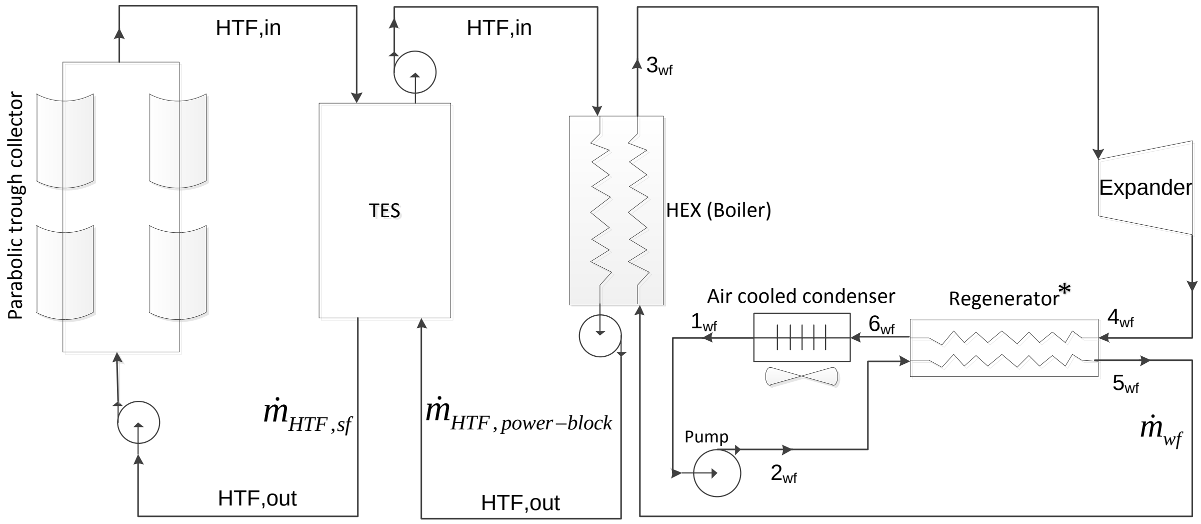

Figure 1: Schematic of an ORC. *Regenerator is not present in case of steam Rankine cycle

In an ORC, low pressure and low temperature liquid at state 1_{wf} is pumped to a higher pressure state, 2_{wf} which is then preheated in a regenerator to state 5_{wf} by recovering heat from cooling the expander exhaust from state 4_{wf} to state 6_{wf}. Remaining heat addition from state 5_{wf} to 3_{wf} occurs in a heater via a heat transfer fluid. Working fluid is then further expanded in a turbine till state 4_{wf}. Regenerator outlet on low pressure side at state 6_{wf} is then cooled in an air cooled condenser till state 1_{wf} to complete the cycle. The power block of a steam Rankine cycle is very similar to the one above except that it does not incorporate a regenerator.

2.1 Choice of working fluid for high temperature ORCs

NIST REFPROP database is exhaustively searched and 18 fluids are shortlisted based on the criteria of having zero ozone depletion potential (ODP) and/or pressure inside the condenser at 45 °C above the atmospheric pressure. These fluids are listed in Table 1. The maximum temperature limit up to which the equation of state (EOS) used in REFPROP is valid is also mentioned. Very few fluids have EOS valid beyond 300 °C, out of which propane and R143a have high saturation pressures at 45 °C (>15 bar) demanding very high upper cycle pressure ($p_{2,wf}$) and hence cannot be used economically for ORC applications. On the other hand butane, pentane and RC318 have manageable condenser pressures but suffer from issues like flammability or high global warming potential (GWP). Based on the conclusions drawn from our previous studies (Garg et al., 2013b), mixture of RC318 (high GWP) with butane or pentane (flammable) can be made non-flammable if its molar concentration of flammable component is equal to or less than 70 % (Zabetakis, 1924). It turns out that these mixtures have acceptable GWPs as well. The mixture of RC-318 and pentane was chosen over the RC-318 and butane mixture since it exhibited a greater temperature glide in the condenser. The rationale was that a greater temperature glide facilitates the use of cheaper and more compact condenser heat exchangers by increasing the LMTD. Also, alkanes with five or more carbon atoms tend to result in high pressure systems.

Table 1: Shortlisted fluids from the NIST REFPROP database

Fluid	*Saturation pressure at 45 °C (bar)*	*Maximum temperature (°C)*	*Maximum pressure (bar)*
R-1234yf	11.5	137	300
R-1234ze	8.8	147	200
R-218	14.4	167	200
R-245fa	2.9	167	2000
R-134a	11.6	182	700
R-142b	6.0	197	600
R-227ea	8.0	202	600
R-365mfc	1.2	227	350
R-245ca	2.1	227	600
R-236ea	3.9	227	600
R-152a	10.4	227	600
R-125	22.6	227	600
R-141b	1.6	227	4000
Butane	4.3	302	2000
Pentane	1.3	327	1000
RC-318	5.7	350	600
R143a	20.6	377	1000
Propane	15.3	377	10000

3. METHODOLOGY

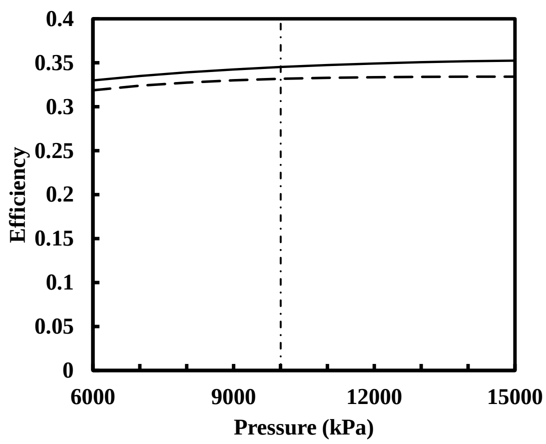

Figure 2. Effect of increasing the evaporator pressure on the efficiency of the steam Rankine cycle
Legend: ——— steam Rankine cycle, — — — ORC.

To facilitate comparison between a solar steam Rankine cycle and an ORC on a one-to-one basis, both the cycles are modelled with the same solar field aperture area. Hence, the thermal heat input for the both the cycles is the same and the cost associated with the solar equipment is not considered in this paper. Further, the evaporator pressure and lowest temperature of the both the cycles is fixed at 100 bar and 45 °C respectively using the same rationale. 100 bar was chosen as the evaporator pressure for both the cycles because it was observed that with increasing the pressure beyond 100 bar, the efficiency increased only marginally. Figure 2 illustrates the decreasing benefits, in terms of efficiency, of increasing the evaporator pressure.

3.1 Solar field

The solar field is assumed to provide a supply of HTF with inlet and outlet temperatures of 293 °C and 393 °C, respectively to either cycle which is in accordance with a majority of parabolic trough CST plants across the world. The place chosen for the case study is Ahmedabad, India, situated at 23.07 °N, roughly near the Tropic of Cancer and the day assumed is vernal equinox, i.e. 21st March. Table 2 lists all the solar field details.

Table 2. Solar side details assumed for the case study. DNI stands for direct normal insolation.

	Steam	*ORC*
DNI (kW-hr$_{th}$/m^2-day)	5.3	5.3
Solar-field aperture area (m^2)	40,000	40,000
Collector efficiency (%)	70	70
Storage efficiency (%)	95	95
Thermal energy storage (hours)	12	12
Thermal heat input (MW$_{th}$)	143	143

3.1. Power cycles

T-s and *p-h* charts for the steam Rankine cycle and ORC are shown in Figures 3 and 4, respectively. Note that the pumping processes $1_{wf} \rightarrow 2_{wf}$ are not visible on the *T-s* charts of either cycle since the temperature and entropy changes across the pump are relatively negligible. Modeling details for such an ORC could be found in Garg et al. (2013b) and for the sake of completeness are repeated here.

i. The ambient temperature is assumed to be 30 °C.

ii. Turbine and pump isentropic efficiencies are assumed to be 0.9 i.e.

$$\eta_{turbine} = \frac{h_{3'_{wf}} - h_{4'_{wf}}}{h_{3'_{wf}} - h_{4_{wf}}} = 0.9 \tag{1}$$

$$\eta_{pump} = \frac{h_{2'_{wf}} - h_{1_{wf}}}{h_{2_{wf}} - h_{1_{wf}}} = 0.9 \tag{2}$$

iii. All heat exchangers are modelled as counter flow heat exchangers with approach temperature of 20 °C. Mathematically,

$$T_{3,wf} = T_{HTF,in} - T_{approach} \tag{3}$$

and for the ORC regenerator,
$$T_{6,wf} = T_{2,wf} + T_{appraoch} \tag{4}$$

iv. Thermodynamic property data is extrapolated by 25 °C for the sake of comparison.

3.2 Cost functions

To compute the initial investment cost of the power blocks, appropriate cost functions are formulated for the major cycle components, namely heat exchangers, turbines and pumps and condensers. The expressions are summarized in detail in Table 3.

For the pump, the cost function listed by Arsalis (2008) is used. For all heat exchangers other than condensers, Ho et al. (2015) is referred to and appropriate cost function is used. Turbine costs have been adapted from Silveira and Tuna (2003) which agree well with numbers extrapolated from Black & Veatch, NREL (2012). Finally, condenser cost numbers are based on quotations received from one of the leading air cooled condenser manufacturers. Correction factors have been applied wherever applicable.

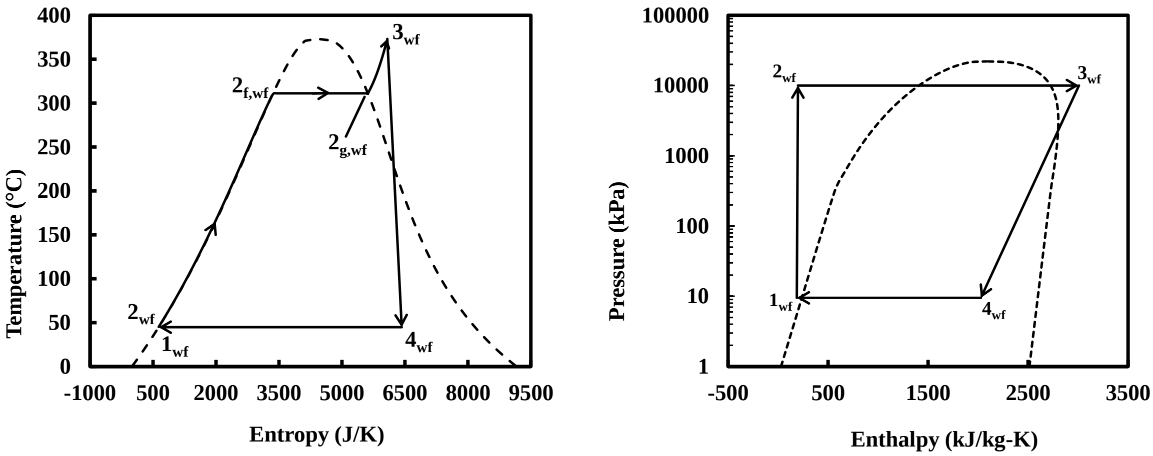

Figure 3: Steam Rankine cycle on *T-s* and *p-h* charts (100 bar, 373 °C).
Legend: — — — saturation curve, —— cycle processes.

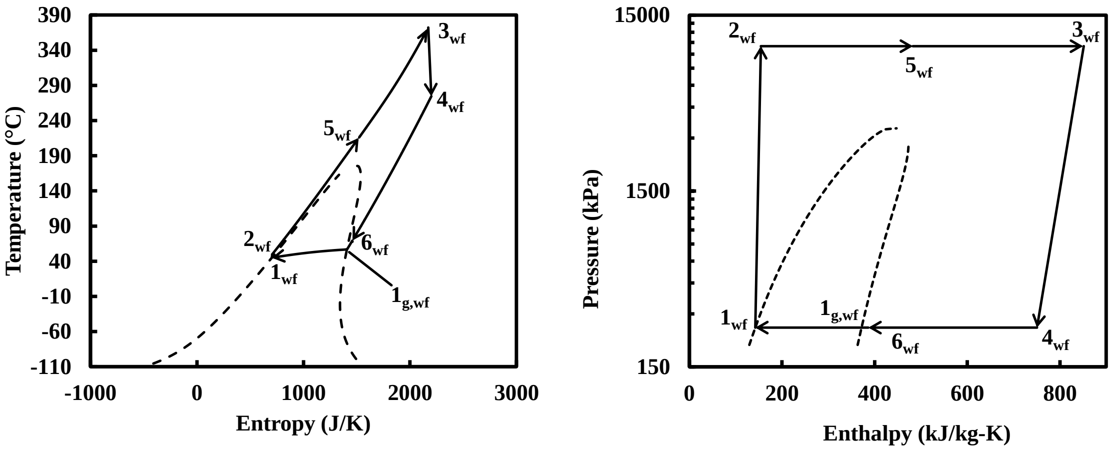

Figure 4: ORC on *T-s* and *p-h* charts (100 bar, 373 °C).
Legend: — — — saturation curve, —— cycle processes

Table 3. Component cost models.

	Variable description	Model equation
C_{pump}	Pump component cost	$C_{pump} = 442 \cdot (\dot{W}_{pump})^{0.71} \cdot 1.41 f_{\eta}$ (5)
f_{η}	Efficiency correction factor	$f_{\eta} = 1 + (\frac{1-0.8}{1-\eta_{pump}})$ (6)
C_{boiler}	Boiler component cost	$C_{boiler} = f_1 \cdot A^{f_2} \cdot f_M \cdot f_p + f_3$ (7) where f_1, f_2, f_3 depend upon the type of heat exchanger, f_M is the material correction factor and f_p is the pressure correction factor
$C_{turbine}$	Turbine component cost	$C_{turbine} = 3540 \cdot \dot{W}_{turbine}^{0.71}$ (8)
$C_{condenser}$	Condenser component cost	$C_{condenser} = 876.6 \cdot A_{frontal} + 1.311 \times 10^{6}$ (9)

3.3 Performance indicators

The parameters of interest are the overall thermal efficiency of cycle defined as

$$\eta_{th,steam} = \frac{\left(h_{3_{wf}} - h_{4'_{wf}}\right) - \left(h_{2'_{wf}} - h_{1_{wf}}\right)}{h_{3_{wf}} - h_{2'_{wf}}} \tag{10}$$

$$\eta_{th,orc} = \frac{\left(h_{3_{wf}} - h_{4'_{wf}}\right) - \left(h_{2'_{wf}} - h_{1_{wf}}\right)}{h_{3_{wf}} - h_{5_{wf}}} \tag{11}$$

volumetric flow rate for a heat addition of $\dot{Q}$ in the boiler heat exchanger, calculated as

$$vfr_{steam} = \frac{\dot{Q}}{\left(h_{3_{wf}} - h_{2'_{wf}}\right) \times \rho_{steam}} \tag{12}$$

$$vfr_{mixture} = \frac{\dot{Q}}{\left(h_{3_{wf}} - h_{5_{wf}}\right) \times \rho_{mixture}} \tag{13}$$

and temperature glide in the ORC, calculated as

$$\Delta T_{glide} = T_{1_{g,wf}} - T_{1_{wf}} \tag{14}$$

where $1_{g,wf}$ corresponds to the intersection of condenser isobar and saturation vapor line. As pointed out earlier, a greater temperature glide in the condenser increases the LMTD and thereby, reduces the heat transfer area required. This allows for more compact and cheaper heat exchangers.
The final performance parameter is the cost of cycle per unit of electricity generated ($/$W_e$)
All calculations are carried out on MATLAB 8.3.0.532 (R-2014a) platform, which is programmed to invoke REFPROP for all thermodynamic property calculations.

4. RESULTS AND DISCUSSIONS

4.1 Efficiency

For identical thermal input from the solar field for the steam Rankine cycle and the ORC, the efficiencies computed are 34.52 % and 33.18 %, respectively which manifest into net-work outputs of 49.43 MW and 47.51 MW, respectively. Hence, on a one-to-one thermodynamic comparison, steam Rankine cycle is slightly more efficient and economic analysis needs to be performed. On further study of the costs involved, evidence in support of ORC begins to accumulate.

4.2 Component Costs

4.2.1 Pump

The pressure ratios in the steam Rankine cycle and mixture based ORC are around 1000 and 40, respectively. However, the pump work in the case of the ORC is roughly 8 times more than the same in steam Rankine cycle. This is expected since the specific volume of water is less as compared to the mixture. Arsalis (2008) performed a thermo-economic analysis of a hybrid solid oxide fuel cell-gas turbine-steam turbine power plants. Correspondingly, cost functions for pumps have been adopted for this paper which are applicable to power scales considered here.

4.2.2 Regenerative heat exchanger

The regenerative heat generator is absent in the case of steam Rankine cycle while in ORC, its heat duty is 127.3 MW. The hot working fluid is at a pressure of 2.5 bar with inlet and outlet temperatures of 272 °C and 68 °C, respectively while the cold fluid is at a pressure of 100 bar with inlet and outlet temperatures of 49 °C and 215 °C, respectively. The log mean temperature difference (LMTD) computed is 37 °C. The temperature profile of the ORC regenerator is shown in Figure 5. An appropriate pressure correction factor is considered while modifying the cost function for ORC application suggested by Taal et al. (2003).

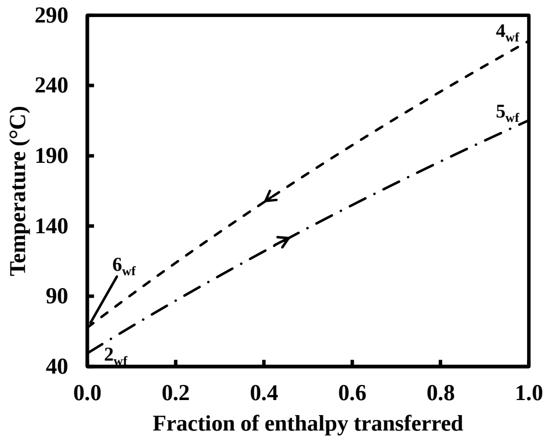

Figure 5. Temperature profile in the ORC regenerator. Legend: – – – *hot working fluid*, – · – · –*cold working fluid*

4.2.3 Boiler

Since the thermal heat input from solar field is identical for both the cycles, the heat duty of the boilers are identical as well at 143.2 MW. The LMTD for the steam Rankine cycle is found to be 83 °C while for the ORC, it is 49 °C. Figure 6 shows the temperature profiles in the boilers for both the cycles. Note that phase change occurs inside the boiler for the steam Rankine cycle whereas there is no phase change inside the ORC boiler. The ORC is transcritical. Boiler cost estimated is in good agreement with estimations from Taal et al. (2003).

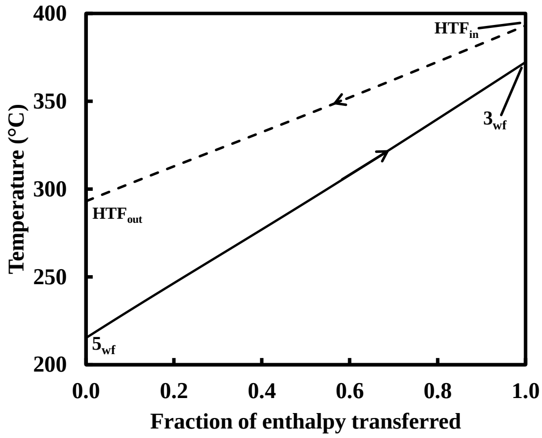

Figure 6a: Temperature profile in the ORC boiler. Legend: – – – *HTF*, —— *working fluid*

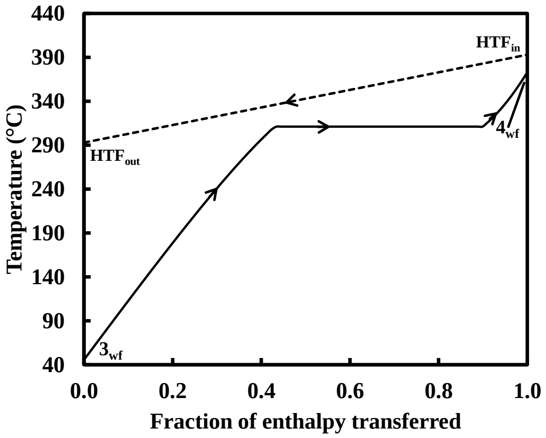

Figure 6b: Temperature profile in the steam Rankine cycle boiler. Legend: – – – *HTF*, —— *working fluid*

4.2.4 Turbine

The volumetric flow rates at the turbine exhaust for the ORC is about one-tenth of the same for the steam Rankine cycle. This flow rate promises decisive savings in equipment size for the ORC. A cost estimate was first made using the function suggested by Silveira and Tuna (2003) and cross validated with numbers suggested by Black & Veatch (contracted by NREL, 2012) for steam turbines employed in a pulverized coal fired Rankine cycle in a 606 MW plant. Furthermore, additional care is taken to scale the cost thus obtained by a factor of 2 since steam turbines are rated according to temperatures achieved in coal fired Rankine cycles, which are generally around 600 °C. For a solar steam Rankine cycle working at a maximum temperature of about 370 °C, the specific work output is 2 times the specific work output for a coal powered Rankine cycle. The corresponding cost for a turbine operating in the ORC cycle is taken to be one-fifth of the cost of a turbine operating in the solar Rankine cycle. The rationale behind this number is the fact that the volumetric flow rates in the ORC is one-tenth the volumetric flow rates in the steam Rankine cycle. Hence, ideally, the turbine cost would be close to one-tenth but following a conservative approach and allowing for non-linear variation of cost of the turbine with flow rates, one-fifth fraction is set as the maximum limit.

4.2.5 Condenser

Air cooled condenser (ACC) is considered to reject the cycle waste heat to the ambience. Turchi et al. (2010b) proposed that air cooled condenser are more cost-effective for solar Rankine cycles than evaporative cooling. A detailed model for ACC can be found in Vidhi et al. (2014). The algorithm to optimize the ACC area and hence cost is different in both the cycles. In steam, the criterion is to minimize the pressure drop on the steam side as the work output is highly sensitive to steam pressure drop in condenser. For example, a 3 % pressure drop in steam condenser is found to drop the cycle efficiency by 1 %. However, in case of ORC, the parameter to optimize is fan power which is set at 1 % of the plant output. Furthermore, in case of mixture based ORC, temperature glide across the condenser increases the effective LMTD resulting in lower condenser area. Table 4 lists the condenser details for either cycle.

Table 4. Condenser design details for steam Rankine and ORC.

Condenser details	*Steam*	*ORC*
Frontal area (m^2)	23600	8200
Fan power (kW)	15.52	64.9
Pressure drop on working fluid side (kPa)	0.61	1.8
LMTD (°C)	15	27.2
Temperature glide (°C)	0	11.17

Table 5 compares the operation parameters and the component costs for each cycle and the resulting $/kW$_e$. The 916 $/kW for the steam Rankine cycle is found to be in good agreement with the cost of the power block (940 $/kW in 2010) suggested by Turchi et al. (2010a).

Although the ORC heat exchangers are more expensive than their steam counterpart due to lower LMTDs, the cost savings achieved in a more compact turbine (due to lower volumetric flow rates) and condenser (due to temperature glide) sufficiently compensate with the result that the mixture based ORC has a significantly lower initial investment cost for a 50 MW$_e$ plant studied in this paper. This suggests a scope of realizing lower energy generation costs.

6. CONCLUSIONS

A thermo-economic evaluation of a steam Rankine cycle and an ORC for similar operation conditions and power outputs suggests the promise of organic working fluids for high temperature ORCs. Key conclusions are listed below:

1. The thermal efficiency of steam Rankine is only marginally better than that of the ORC.
2. The volumetric flow rate at the exhaust of turbine in the ORC is one-tenth of the same in steam Rankine. Hence, significant cost savings can be realized in the ORC turbine.
3. Temperature glide of the working fluid across the ORC condenser manifests into a higher LMTD and thus lesser heat transfer area facilitating a more compact and cheaper condenser. The cost savings in condenser are decisive in realizing its lower initial investment costs.
4. Although regenerator heat exchanger contributes to additional cost for the ORC, the significant difference in the $/kW$_e$ for the two cycles suggests that high temperature ORCs are viable substitutes for the steam Rankine cycle and worth investigating further in detail.

It must be appreciated that although the imposed operating conditions constrained the cycles from operating at their thermo-economic optima, the cost numbers generated nevertheless encourage further investigation of high temperature ORCs. Future scope of work may include comparison of the two cycles operating at their thermo-economic optima, use of more accurate functions or database for thermodynamic data points and LCOE computations. Inhibiting feature is the thermal stability of organic fluids at high temperatures.

Table 5. Steam Rankine cycle and ORC design, performance

	Steam	***ORC***
T_{max}(°C)	373	373
T_{min}(°C)	45	45
P_{max}(MPa)	10	10
P_{min}(MPa)	0.009	0.250
Pressure ratio	1111	39.8
$\eta_{turbine}$(%)	0.9	0.9
η_{pump} (%)	0.9	0.9
T_{amb} (°C)	30	30
ΔT_{htf} (°C)	100	100
vfr (m^3/s)	604.6	62.95
$\dot{W}_{turbine}$ (MW_e)	50	52.2
$\dot{W}_{pump}$ (MW_e)	0.6	4.7
Net Power (MW_e)	49.4	47.5
Efficiency (%)	34.52	33.18
Boiler ($/$kW_e$)	157	261
Regenerator ($/$kW_e$)	N/A	110
Cooling ($/$kW_e$)	445	175
Compression ($/$kW_e$)	4	16
Expansion ($/$kW_e$)	311	130
Total ($/$kW_e$)	917	692

NOMENCLATURE

Symbols

C	component cost	US $
c_p	specific heat at constant pressure	kJ/kg K
f	efficiency correction factor	–
$\dot{m}$	mass flow rate	kg/s
p	pressure	bar
$\dot{Q}$	heat transfer rate	kW
T	temperature	K
$\dot{W}$	power output	kW
vfr	volumetric flow rate	m^3/s

Greek letters

η	efficiency	–
ρ	density	kg/m^3

Subscripts

amb	ambient

f	saturated liquid
g	saturated vapor
htf	heat transfer fluid
in	inlet
out	outlet
η	efficiency
th	thermal
wf	working fluid
1 to 6	states on ideal thermodynamic cycle

Superscripts

'	states on real thermodynamic cycle

REFERENCES

Arsalis, A., 2008, Thermo-economic modeling and parametric study of hybrid SOFC-gas turbine-steam turbine power plants ranging from 1.5 to 10MW_e, *Journal of Power Sources,* vol. 181: p. 313-326

Black & Veatch, 2012, Cost and performance data for power generation technologies, *Prepared for National Renewable Energy Laboratory*

Garg, P., Kumar, P., Srinivasan, K., 2013a, Supercritical carbon dioxide Brayton cycle for concentrated solar power, *The Journal of Supercritical Fluids,* vol. 76: p. 54-60

Garg, P., Kumar, P., Srinivasan, K., Dutta, P., 2013b, Evaluation of isopentane, R-245fa and their mixtures as working fluids for organic Rankine cycles, *Applied Thermal Engineering,* vol. 51: p. 292-300

Garg, P., Kumar, P., Srinivasan, K., Dutta, P., 2013c, Evaluation of carbon dioxide blends with isopentane and propane as working fluids for organic Rankine cycles, *Applied Thermal Engineering,* vol. 52: p. 439-448

Garg, P., Srinivasan, K., Dutta, P., Kumar, P., 2014, Comparison of CO_2 and Steam in Trans-critical Rankine Cycles for Concentrated Solar Power, *SolarPACES 2013 International Conference,* Energy Procedia: p. 1138-1146

Ho, C.K., Carlson, M., Garg, P., Kumar, P., 2015, Cost and performance tradeoffs of alternative solar-driven S-CO_2 Brayton cycle configurations, *ASME 2015 Power and Energy Conversion Conference*

Mikielewicz, D., Mikielewicz, J., 2010, A thermodynamic criterion for selection of working fluid for subcritical and supercritical domestic micro CHP, *Applied Thermal Engineering*, vol. 30: p. 2357-2362

Peard, M.G., Stubbs, F.J., Hinshelwood, C., Danby, C.J., 1952, The kinetics of thermal decomposition of branched-chain paraffin hydrocarbons, *Proceedings of the Royal Society of London. Series A, Mathematics and Physical Sciences,* vol. 214: p. 330-338

Quoilin, S., Declaye, S., Tchanche, B.F., Lemort, V., 2011a, Thermo-economic optimization of waste heat recovery Organic Rankine Cycles, *Applied Thermal Engineering,* vol. 31: p. 2885-2893

Quoilin, S., Lemort, V., 2009, Technological and Economical Survey of Organic Rankine Cycle Systems, *Economics and management of energy in industry*

Quoilin, S., Orosz, M., Hemond, H., Lemort, V., 2011b, Performance and design optimization of a low-cost organic Rankine cycle for remote power generation, *Solar Energy*, vol. 85: p. 955-966

Silveira, J.L., Tuna, C.E., 2003, Thermo-economic analysis method for optimization of combined heat & power systems. Part 1, *Progress in Energy and Combustion Science,* vol. 29: p. 479-485

Taal, M., Bulatov, I., Klemes, J., Stehlik, P., 2003, Cost estimation and energy price forecasts for economic evaluation of retrofit projects, *Applied Thermal Engineering,* vol. 23: p. 1819-1835

Tchanche, B.F., Lambrinos, Gr., Frangoudakis, A., Papadakis, G., 2011, Low-grade heat conversion into power using organic Rankine cyles – A review of various applications, *Renewable and Sustainable Energy Reviews,* vol. 15: p. 3963-3979

Turchi, C., 2010, Parabolic Trough Reference Plant for Cost Modeling with the Solar Advisor Model (SAM), *Technical Report NREL/TP-550-47605*

Turchi, C., Mehos, M., Ho, C.K, Kolb, G.J., 2010, Current and Future Costs for Parabolic Trough and Power Tower Systems in the US Market, *SolarPACES 2010 International Conference,* NREL/CP-5500-49303

Vidhi, R., Garg, P., Orosz, M.S., Goswami, D.Y., Kumar, P., 2014, Ground Cooling System for Improving the Efficiency of Low Temperature Power Generation, *ASME 8th International Conference on Energy Sustainability*, doi. 10.1115/ES2014-6618

Wang, E.H., Zhang, H.G., Fan, B.Y., Ouyang, M.G., Zhao, Y., Mu, Q.H., 2011, Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery, *Energy,* vol. 36: p. 3406-3418

Zabetakis, M.G., 1924, Flammability Characteristics of Combustible Gases and Vapors, *U.S. Dept. of Interior, Bureau of Mines,* TN23.U4 NO. 627 622.06173

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