Thermodynamic Modeling of the Solar Organic Rankine Cycle with Selected Organic Working Fluids for Cogeneration

Suresh Baral and Kyung Chun Kim

ABSTRACT

Fifteen (15) organic fluids were thermodynamically modeled to evaluate their fitness and performance as working fluids in an Organic Rankine Cycle (ORC) based cogeneration system. This article presents the exergy efficiency, thermal efficiency, solar power cycle efficiency, cogeneration efficiency, mass flow rate, heat input, required area of the solar collector and hot water production for the evaluated working fluids the low-temperature (90 and medium-temperature (125 solar organic Rankine cycles. Thermodynamic modeling was carried out using a commercial 1 kW scroll expander, two compact heat exchangers, a diaphragm pump and a solar collector. The article also describes the use of solar ORC technology for electricity generation and producing hot water as cogeneration. Commercial software, Engineering Equation Solver (EES), was used to calculate the operating parameters of the solar ORC. Of the 15 fluids investigated, R134a and R245fa were found to be the most appropriate working fluids for low-temperature and medium-temperature solar ORC cogeneration systems, respectively. RC318 and R123 offer attractive performance but require environmental precautions owing to their high ozone depletion potential (ODP) and high global warming potential (GWP). The article also estimates the hot water production from different working fluids for a period of one year in Busan, South Korea.

Keywords

Thermodynamic modeling, Solar organic Rankine cycle, Working fluids, Cogeneration, Heat source temperature, Hot water

Nomenclature

Q = heat (kW)
W = work done (kW)
A = area (m^2)

E = exergy (kJ)
I = solar insolation (W m^{-2})
T = temperature (°C)
Cp = constant pressure specific heat capacity (kJ kg^{-1} K^{-1})
h = enthalpy (kJ kg^{-1})
s = entropy (kJ kg^{-1} K^{-1})
V = volume of hot water (L)

Greek Symbols
η = Efficiency (%)

Subscripts
i = inlet
th = thermal
in = inlet
out = outlet
c = collector
t = turbine
p = pump
hw = hot water
sun = hour of sunshine
o = ambient
exg = exergy
spc = solar power cycle
cog = cogeneration

INTRODUCTION

The organic Rankine cycle (ORC) is similar to the cycle of a conventional steam turbine power plant but it uses a high molecular organic fluid as the working fluid. The high molecular mass allows the exploitation of effectively low temperature heat sources to produce electricity in a wide range of power outputs ranging from a few watts to several megawatts in many applications, such as geothermal, solar, desalination, ocean thermal and biomass power plants. Currently, research has been carried out extensively focusing on the full utilization of waste heat from industrial plants. Owing to environmental concerns and the depletion of fossil fuels, this energy utilization concept from low heat sources has been used widely.

More attention is being paid to organic working fluids, which must be risk less in terms of: human health, ozone depletion, global warming contribution. This way, ORC fluids can enhance their sustainable potential as key components of distributed generation (DG) plants capable of utilizing low and medium quality heat sources (80 to 150. The selection of suitable organic fluids in the ORC plants should have the following desirable characteristics, as described by appropriate low critical temperature and pressure: small specific volume, low viscosity and surface tension, high thermal conductivity, suitable thermal stability, non-corrosive, non-toxic and compatible with engine material and lubricating oil [1,2].

Solar energy is available everywhere and is completely renewable. In remote locations, the conversion of solar energy to electricity could be an important option for enhancing the development of rural communities. The low energy density and the fluctuations of the source availability are the main obstacles in solar energy applications. Photovoltaic modules, which are commercially available technology for converting solar energy directly into electricity, have very low efficiency and require batteries for storage, which limits its application in developing countries.

Few studies have reported the experimental data from an operational solar ORC system. A performance and design optimization study by Quoilin et al. [3] based on low cost solar ORCs provided the modeling results for solar thermal electricity generation in the remote off-grid areas of developing countries, and revealed an overall thermal efficiency of 8%. Saitoh et al. [4] reported a solar ORC efficiency of 7% and with the working fluid, R113, and an expander efficiency of 63%. Jing et al. [5] analyzed a combination of ORC with compound parabolic concentrators (CPCs) for electricity generation with the working fluid, HCFC-123. They suggested that heat exchangers with two thermal oil cycles can improve the collector efficiency by 8.1-20.9% with a maximum evaporating temperature of the working fluid of 120°C. He et al. [6] simulated a parabolic trough solar energy generation system, and examined the optimal feeding fluid temperature, suitable number of heat recovery series, thermal storage capacity, and system performance with three different working fluids, R113, R123 and pentane. Torres and Rodriguez [7] provided a detailed analysis of low power solar-driven Rankine cycles with working fluids, such as toluene, octamethylcyclotetrasiloxane (D4) and hexamethyldisiloxane (MM) in direct solar vapor generation configuration of solar ORC applied in the desalination process, and have a global efficiency of 17.3%, 15.7% and 15%, respectively when the condensation temperature was set to

35°C. Gang et al. [8] examined the regenerative and without regenerative solar ORC cycle, and reported an efficiency of approximately 8.6% and 4.9% respectively for irradiance 750 W/m^2. Wang et al. [9], using R245fa as the working fluid and a flat-plate solar collector, reported a maximum system efficiency of approximately 3%, where the time-averaged Rankine cycle efficiency was approximately 2.8% in the morning and 5.3% in the afternoon. Twomey et al. [10] simulated a solar ORC and revealed an efficiency of 3.4% with a maximum expander isentropic efficiency of 59%. Manolakos et al. [11] provided the design and experimental results from a 2 kWe solar ORC system using evacuated tube collectors, scroll expanders and R134a as the working fluid, and reported an efficiency of 4% applied in a reverse osmosis process in desalination. Pedro et al. [12] examined the exhaust waste heat recovery potential of a micro-turbine using an ORC, where different dry organic fluids were considered to find the exergy efficiency.

Currently, 1.6 billion people all around the world still have no access to electricity. Huge communities in underdeveloped countries do not have a centralized grid connected to their main lines of electricity. Therefore, conversion techniques from low heat source to electricity in the underdeveloped countries, which require rural electrification, are needed. Several groups including Solar Turbine Group (STG) have tried to implement small scale solar thermal technology using medium temperature collectors aimed at replacing or complementing diesel generators in the off grid areas in underdeveloped countries, thereby generating clean and green power at very low installation and maintenance cost.

The present work aims to select most suitable working fluids for low-temperature (90 and medium-temperature (125 solar ORC systems to produce both electricity and hot water. Characteristics of 15 potential working fluids are evaluated and compared for a 1 kW micro-power system.

SYSTEM MODELING

The present solar ORC system consisted of a working fluid pump (diaphragm type), two compact heat exchangers and a commercial scroll expander, as shown in Figure 1. The working principle is explained as the hot water obtained by the solar collector is passed through the evaporator. The ORC working fluid is pumped and passed through the evaporator, where it changes its phase. The working fluid is expanded into the scroll expander to generate electricity. The working fluid at the turbine outlet is condensed in the condenser by cold water supplied from a tap and flows back into the circulation pump to begin a repeated cycle. The condensing temperature determines the temperature of hot water production as cogeneration.

A simulation and thermodynamic modeling of the present system was carried out to predict the performance characteristics of the solar organic Rankine cycle under a range of conditions and parameters. The graphical representation of computation modules organized by direction of energy flow can be shown in Figure 2. The system equations were derived easily from the energy and mass balance for a control volume. The exergy of the ORC system was calculated using the enthalpy and entropy of the evaporator inlet and the dead state (To = 25°C). The system equations obtained can be expressed as

Evaporator: $$Q_{in} = h_4 - h_1 \tag{1}$$

Condenser: $$Q_{out} = h_3 - h_2 \tag{2}$$

Turbine: $$W_t = \dot{m}(h_1 - h_{2s})\eta_t \tag{3}$$

Pump: $$W_p = \dot{m}(h_3 - h_{4s})/\eta_p \tag{4}$$

Power output: $$W_t - W_p = \dot{m}(h_1 - h_{2s})\eta_t - \dot{m}(h_3 - h_{4s})/\eta_p \tag{5}$$

Thermal efficiency: $$\eta_{ORC} = \frac{W_{out}}{Q_{in}} \tag{6}$$

Solar power cycle efficiency: $$\eta_{SPC} = \eta_{th} \times \eta_c \tag{7}$$

Cogeneration efficiency: $$\eta_{cog} = (\dot{W}_{net} - \dot{W}_p + \dot{Q}_{out})/\dot{Q}_{in} \tag{8}$$

Exergy state point: $$E_i = \dot{m}\left[\left(h_i - \dot{h}_o\right) - T_o\left(s_i - s_o\right)\right] \tag{9}$$

Exergy efficiency: $$\eta_{exg} = \dot{W}_{net}/\dot{E}_i \tag{10}$$

Area of collector: $$Ac = Q_{in}/(\eta_c \times I \times W_{net}) \tag{11}$$

Hot water production: $$V_{hw} = \left[\frac{\eta_c \times A_c \times I}{c_p \times (T_c - T_o)}\right] \times T_{sun} \tag{12}$$

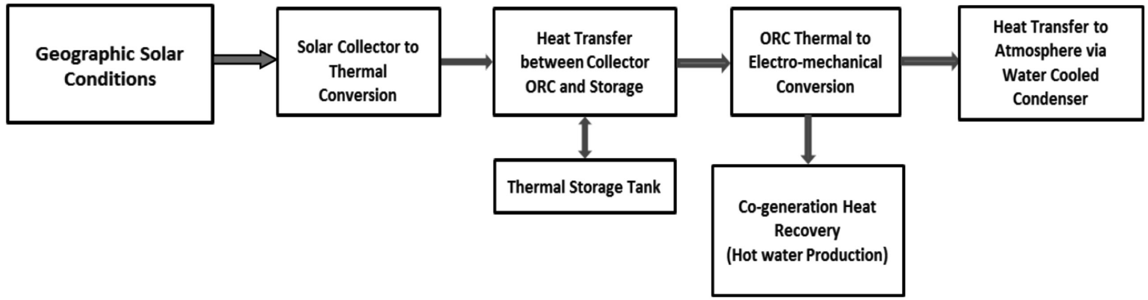

Figure 2. Flow chart of simulation with input parameters and output quantities.

Selectable Input Parameters

Latitude	Collector size	Hot water flow	Working Fluid	Condenser Type
Longitude	Heat Transfer Fluid	Storage Tank	Expander efficiency	Cold water
Altitude	Collector Materials		Pump Efficiency	Amb.Temp
			Pinch Temperature	

Calculated Quantities

Solar Insolation	Collector Efficiency	Power Delivery	Cycle Efficiency	Condenser Size
Ambient Temperature	Solar Input Power	Size of Evaporator	Power Output	Water flow rate
		Tank Size	Solar Power Efficiency	
			Exergy Efficiency	
			Co-generation Efficiency	
			Hot water Production Volume	

The present work compared several ORC working fluids for system efficiencies, appropriate area of the solar collector and production of hot water as cogeneration for 1 kW power output. Several authors [13-18] worked on analyzing the characteristics of the different working fluids for the conversion of low grade heat. For the simulation and analysis of ORC working fluids as potential candidates in solar applications, the critical temperature of fluids, ranging from 95°C to 240°C, were selected. For low-temperature solar ORC using a flat-plate solar collector, the critical temperature of the working fluids ranged from 95°C to 135°C, whereas the critical temperature for medium-temperature solar ORC using a vacuum tube type solar collector ranged from 150°C to 240°C. These critical temperature ranges were taken as a fundamental basis because of its thermophysical behavior at various heat source temperatures. For the simulation, the desired thermodynamic properties of the ORC working fluid at different state points should be known. The thermodynamic properties of an ORC working fluid can be described best by the energy equation and calculated using software known as an Energy Equation Solver (EES). Therefore, all the thermodynamics properties of ORC working fluids in this study were obtained using commercial software EES.

For simplicity, several reasonable assumptions were implemented by thermodynamic modeling. The assumptions are listed as follows:

a. The electric power at the expander is fixed at 1 kW.
b. The evaporating temperature for low and medium temperature solar ORC are supposed to be 85°C and 120°C, respectively, with a pinch point temperature difference of 5°C.
c. The condensation temperature is fixed to 45°C.
d. The efficiency of the expander is 70%.
e. The efficiency of the pump is assumed to be 70%.
f. The solar collector efficiency is presumed to be 70%.
g. The internal irreversibility is ignored.
h. The pressure drops in the components other than the expander are ignored.

For low-temperature and medium-temperature solar ORCs, the heat source is expected to be 90°C and 125°C, respectively.

RESULTS AND DISCUSSION

In total, 15 pure organic working fluids were selected as potential candidates, as listed in Table 1. Only one criterion was considered at this first step, a critical temperature above 90°C. Hot water was provided from the solar collectors. The condenser was cooled with tap water at 25°C. In this study, the vapor at the turbine inlet was saturated. A re-heater was not introduced because it could add too much expense to the system. Table 2 lists the system performance results for a design of 1kW power output. This study examined the ORC system located in Busan, South Korea (latitude and longitude of 35.15°N and 129.06°E, respectively).

Efficiencies

The ORC efficiency is the most important index used to evaluate the performance of the system. Nine and six working fluids emerged as potential candidates for the low-temperature and medium-temperature solar ORC respectively. The exergy efficiency is an indicator of how close the thermal efficiency is to the highest value permissible depending on the temperature range (Carnot efficiency). The system exergy efficiency ranged from 29.29% to 50.03%. The maximum exergy efficiency was obtained using RC318 and R123, as shown in Table 2. This is because these

Table 1. Properties of the organic fluids along with the safety and environmental parameters

Working Fluid		Molecular mass (kg/kmol)	Critical Temperature (°C)	Critical Pressure (MPa)	ASHRAE safety group	Atmospheric life time(yr)	ODP	GWP (100 yr)
1	R22	86.5	96.14	4.9	A1	12.1	0.055	1500
2	R290	44.1	96.68	4.2	A3	0.041	0	20
3	R134a	102.03	101	4.059	A1	14	0	1430
4	R227ea	170	101.7	2.9	A1	34.2	0	3220
5	R500	99.3	105.6	4.17	A1	74	0.74	6010
6	R12	120.91	112	4.11	A1	100	0	10890
7	RC318	200.03	115.2	2.79	A1	3200	0	10250
8	R717	17.03	132.3	11.33	B2	0.01	0	1
9	R600a	58.12	134.7	3.64	A3	12	0	n.a
10	R245fa	134.05	154.2	3.64	B1	8.8	0	820
11	R123	153	183.7	3.668	B1	1.3	0.02	77
12	R11	137.4	198	4.4	A1	50	1	5800
13	R141b	116.95	204.2	4.25	n.a	9.3	0.12	725
14	Methanol	32.04	240.2	8.1	n.a	n.a	n.a	n.a
15	Ethanol	46.07	240.8	6.14	n.a	n.a	n.a	n.a

fluids rise at low working pressures as the enthalpy difference in the expander increases. Lower exergy efficiency means higher system irreversibility. Normally, in an ORC system, the evaporator makes the largest contribution to the overall irreversibility followed by the expander. R500, the lowest exergy efficiency fluid, requires the largest heat input due to the large irreversibility in the evaporator.

The thermal efficiency and solar power cycle efficiency ranged from 5.38% to 12.58% and 3.67% to 8.8%, respectively, as shown in Figures 3 and 4. The thermal efficiency was based on the first law of thermodynamics and is the typical efficiency of an ORC. The solar power cycle efficiency considers the efficiency of the solar collectors. Therefore, the solar power cycle efficiency is always 2 ~ 3% lower than the thermal efficiency. This efficiency can be compared directly with photovoltaic devices in terms of electric power generation. On the other hand, the system efficiency of a solar ORC with cogeneration should be considered using hot water from the condenser. The cogeneration efficiency ranged 73.11% to 77.92% for the low-temperature ORC whereas 80.24% to 82.24% for the medium-temperature solar ORC. A cogeneration unit should utilize the high thermal efficiency of solar thermal collectors to provide water heating, while generating electricity during the off-peak periods when the sun's potential would otherwise be wasted.

Table 2. Comparison of the performances of different working fluids for a 1 kW power output.

Working Fluid	P_{max} (kPa)	P_{min} (kPa)	PR	Q_{in} (kW)	$\dot{m}_{wf}$ (kg/s)	η_{exg} (%)	η_{th} (%)	η_{cog} (%)	η_{SPC} (%)	$\dot{V}$ (L/s)	$A_c(m^2)$
R22	4037	1730	2.33	17.99	0.12	31.04	5.55	73.55	3.89	0.52	32.13
R290	3437	1534	2.24	18.57	0.06	30.08	5.38	73.41	3.77	0.63	33.16
R134a	2928	1161	2.52	17.41	0.1	32.08	5.74	74.36	4.02	0.59	31.09
R227ea	2075	638	3.25	15.04	0.12	37.14	6.65	73.9	4.65	0.58	26.85
R500	3015	1282	2.35	19.07	0.13	29.29	5.24	73.11	3.67	0.73	34.05
R12	2536	1084	2.34	17.88	0.14	31.23	5.6	73.85	3.91	0.82	31.93
RC318	1500	289	5.19	11.59	0.08	48.19	8.63	77.92	6.04	0.54	20.69
R717	4609	1782	2.58	14.74	0.02	37.88	6.78	76.11	4.75	0.37	26.32
R600a	1462	469	3.11	13.25	0.04	42.13	7.54	76.85	5.28	0.9	23.67
R245fa	1929	181	10.66	8.51	0.04	46.82	11.76	81.08	8.23	0.3	15.18
R123	1201	111	10.82	7.95	0.04	50.03	12.58	82.24	8.8	0.5	14.2
R11	1228	202	6.08	9.09	0.04	43.76	11	80.51	7.69	0.68	16.24
R141b	1031	126	8.18	8.49	0.03	46.84	11.77	81.44	8.23	0.68	15.17
Methanol	631	44	14.34	8.18	0.006	48.62	12.22	80.24	8.55	0.95	14.62
Ethanol	429	23	18.65	8.37	0.006	47.55	11.95	82.05	8.36	1.24	14.95

Sources Temperature	Decision rating criteria for working fluid selection in solar ORC co-generation system											
	P_{max} (kPa)	P_{min} (kPa)	PR	η_{exg} (%)	η_{th} (%)	η_{spc} (%)	A_c (m^2)	Safety Factor	ODP	GWP	V_{hw}(L/ day)	Remarks
Low-temperature	3000	1200	4	30	5.5	4	32	A1	0	1500	3700	Single Expander
Medium-temperature	1300	120	12	45	11.5	8	15	A1	0	1500	1800	Double Expander

Figure 3. System exergy, thermal and solar power cycle efficiency for low-temperature solar ORC working fluids at condensing temperature of 45°C.

Sources Temperature	Decision rating criteria for working fluid selection in solar ORC co-generation system											
	P_{max} (kPa)	P_{min} (kPa)	PR	η_{exg} (%)	η_{th} (%)	η_{spc} (%)	A_c (m^2)	Safety Factor	ODP	GWP	V_{hw}(L/ day)	Remarks
Low-temperature	3000	1200	4	30	5.5	4	32	A1	0	1500	3700	Single Expander
Medium-temperature	1300	120	12	45	11.5	8	15	A1	0	1500	1800	Double Expander

Figure 4. System exergy, thermal and solar power cycle efficiency for medium-temperature solar ORC working fluids at a condensing temperature of 45°C.

Variations of the exergy efficiencies of the low-temperature and the medium-temperature solar ORC with respect to the turbine inlet temperature are shown in Figures 5 and 6. The exergy efficiencies are increased with increasing turbine inlet temperature. The pinch point temperature was maintained as a constant and the vapor at the expander inlet became saturated. In the case of low-temperature solar ORC, RC318 shows the maximum exergy efficiency up to 47% at 90°C of the turbine inlet temperature followed by R600a and R717. Other six fluids show similar exergy efficiency in the range of 27 to 33% at 90°C of the turbine inlet temperature. In the case of medium-temperature solar ORC, R123 shows the maximum exergy efficiency. However, the difference is not pronounced compared with the low-temperature case. The exergy efficiency of R245fa shows a unique pattern. The maximum exergy efficiency is saturated when the turbine inlet temperature exceeds 120°C. This nature suggests the existence of an optimal operating condition.

Effects of the turbine inlet pressure on the exergy efficiency can be seen in Figures 7 and 8. The system exergy efficiency increased with increasing turbine inlet pressure. In the case of low-temperature solar ORC,

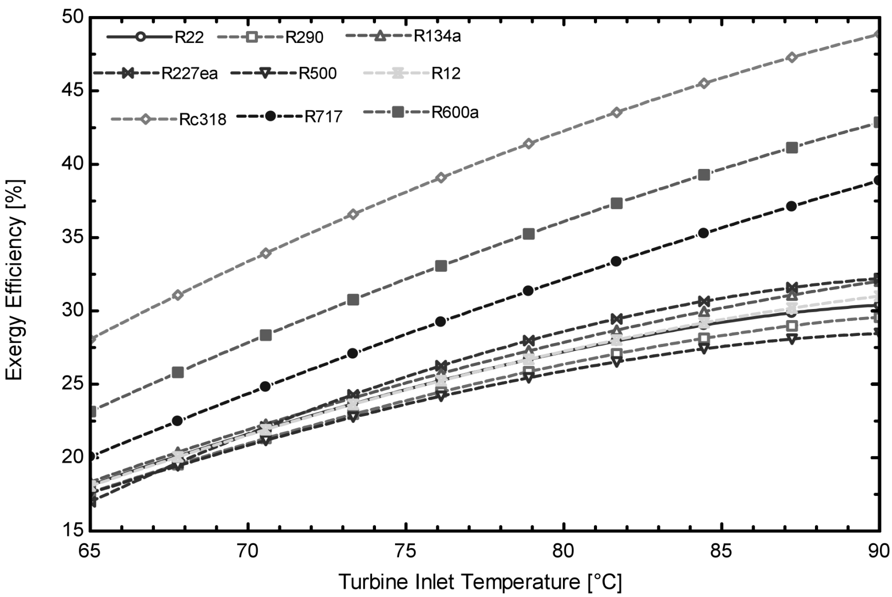

Figure 5. Variation of the exergy efficiency for low-temperature solar ORC working fluids with the turbine inlet temperature at the condensing temperature of 45°C.

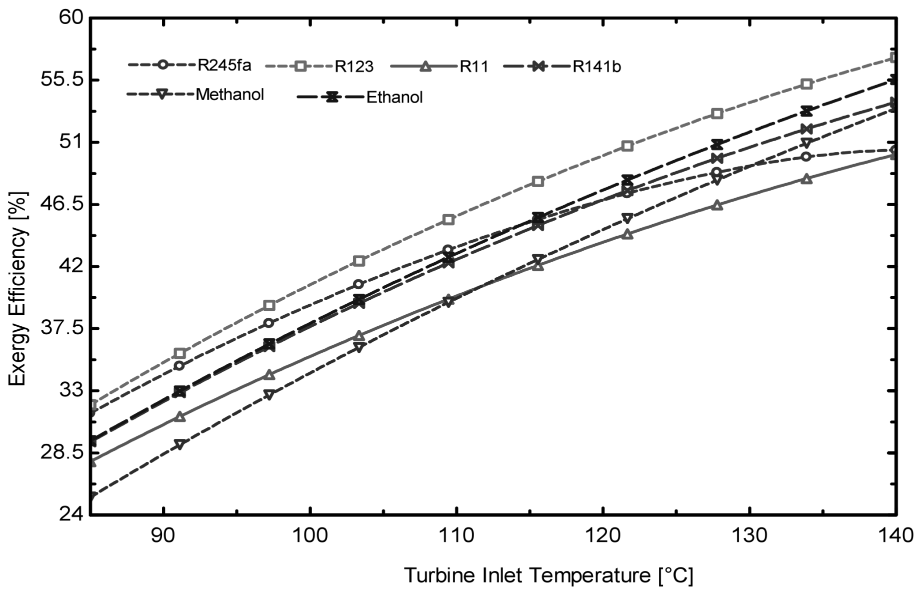

Figure 6. Variation of the exergy efficiency for medium-temperature solar ORC working fluids with the turbine inlet temperature at a condensing temperature of 45°C.

RC318 shows the maximum exergy efficiency 38 to 55% for the range of turbine inlet pressure from 1000 to 2500 kPa. The high exergy efficiency in the low-temperature ORC with RC318 attributes to high molecular mass compared to other working fluids. Other eight working fluids show a similar range of exergy efficiency with different turbine inlet pressure ranges as shown in Figure 7. In the case of medium-temperature solar ORC, R123 shows the best performance for the range of 500 to 1800 kPa turbine inlet pressure. Interestingly, the exergy efficiencies of R245fa are gradually increased with increasing turbine inlet pressure for the wide range of pressure variation, 870 to 2900 kPa as shown in Figure 8. The exergy efficiencies of methanol and ethanol increase in a short range 0.2 to 1.0 MPa.

Pressure Ratio

The cycle pressure ratio determines the type of expander size and number in the ORC system. A high pressure ratio also increases the efficiency of the system. On the other hand, a higher pressure ratio accounts for the installation of a number of expanders arranged in series or parallel

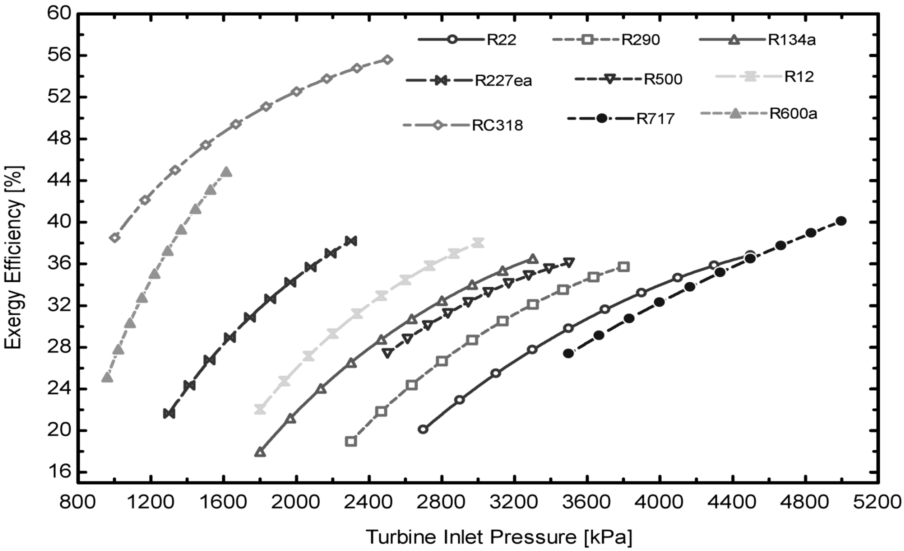

Figure 7. Exergy efficiency versus turbine inlet pressure for low-temperature solar ORC working fluids at a condensing temperature of 45°C

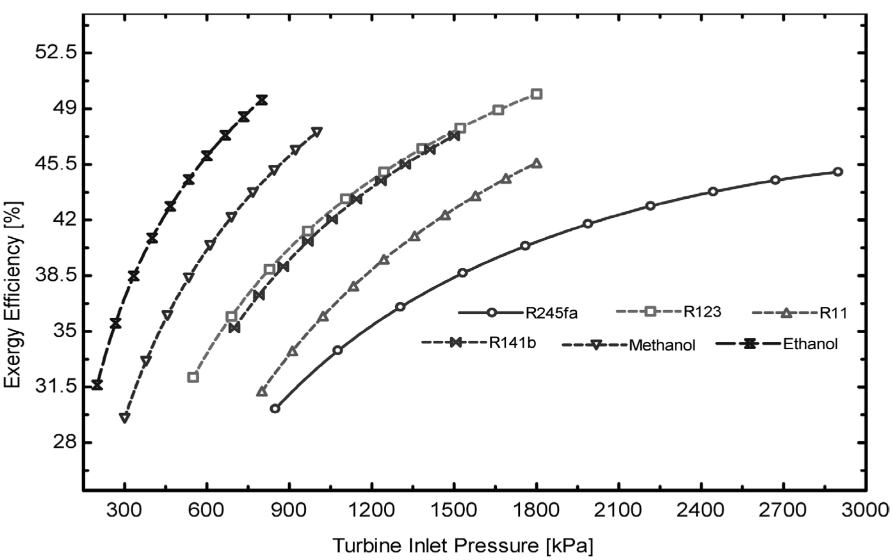

Figure 8. Exergy efficiency versus turbine inlet pressure for medium-temperature solar ORC working fluids at a condensing temperature of 45°C.

in an ORC system. An excessively high pressure in the evaporator and an excessively low pressure in the condenser should be avoided due to structural problem and leakage. According to Tchanche et al. [18], good pressure values were in the range of 0.1 – 2.5 MPa, and a pressure ratio of approximately 3.5 is reasonable for a single stage expander. From Table 2, R227ea, RC318 and R600a, R123, methanol and ethanol have low condensing pressure, meaning that they have a higher pressure ratio. R22, R290, R500, and R717 have higher pressure above 3 MPa in the evaporator. A high turbine inlet temperature yields a high pressure ratio, which can be observed from Figures 9 and 10 when the temperature is varied from 65°C to 140°C. The ORC fluids with good condensing and evaporating pressures are R134a, R227ea, RC318, and R600a for the low-temperature solar ORC system, and R245fa, R123, R11, and R141b for the medium-temperature solar ORC system.

Flow Rates

For economic aspects, the turbine outlet volume flow rate plays an important role because it determines the size and cost. As shown in Table 2, R12, R500, R600a, methanol and ethanol exhibited a high volume flow rate. Fluids with a low volume flow rate are preferable for decreasing the pump size and friction loss in a pipe system. Among these are R245fa,

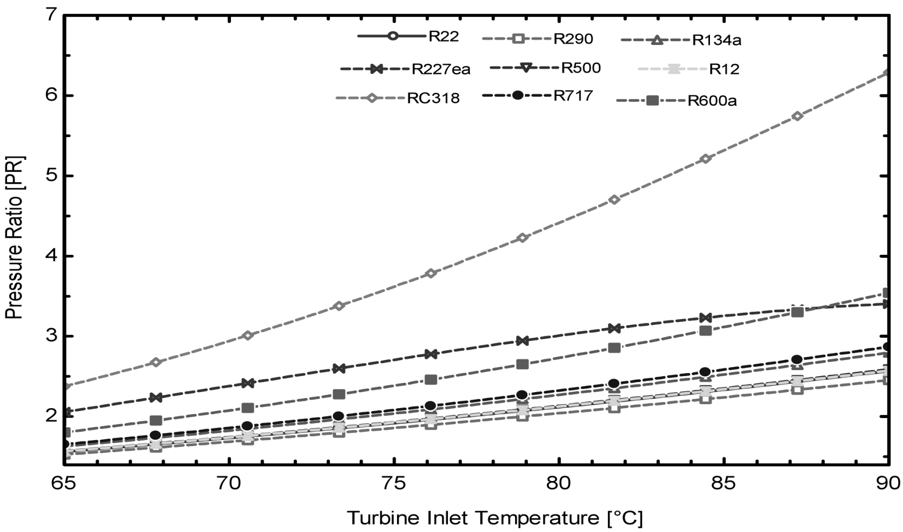

Figure 9. Variation of the pressure ratio for low-temperature solar ORC working fluids with turbine inlet temperature at a condensing temperature of 45°C.

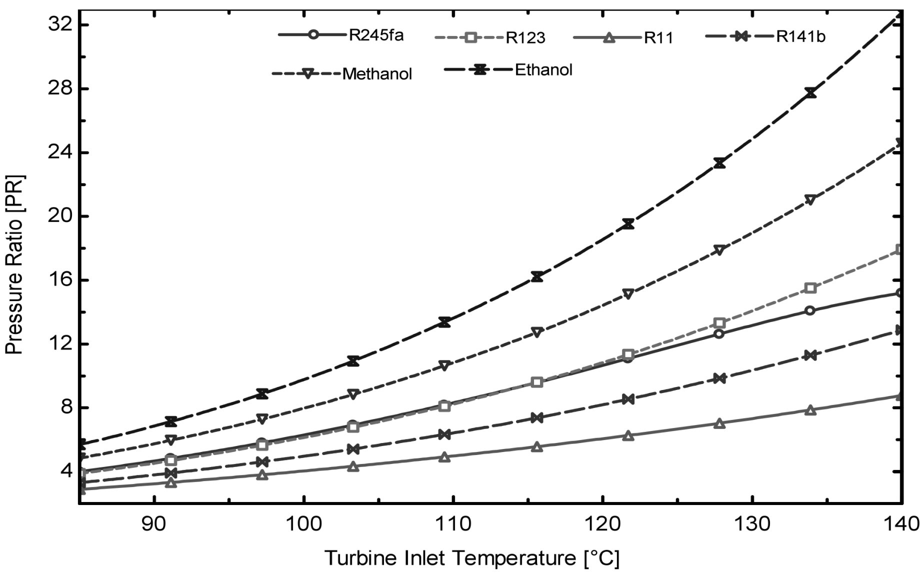

Figure 10. Variation of the pressure ratio for medium-temperature solar ORC working fluids with turbine inlet temperature at a condensing temperature of 45°C.

R717, R123, R22, RC318, R227ea, and R134a. In general, the turbine outlet volume flow rate is inversely proportional to the turbine inlet temperature, which can be illustrated in Figures 11 and 12. For the low-temperature solar ORC, R717 shows the best performance followed by R22, RC318 and R134a. For the medium-temperature solar ORC, R245fa shows the lowest turbine inlet volume flow rate followed by R123 and R11.

The proper introduction of a working fluid mass flow rate is necessary for the expansion process. The pump selection is also another fundamental aspect for appropriate mass flow rate pumping. From Table 2, R290, RC318, R717 and R600a yielded a low mass flow rate for a low-temperature solar ORC. This is useful because they also require a low heat input. For medium-temperature solar ORC, methanol and ethanol exhibited a similar low mass flow rate because of their similar properties. From Table 2, methanol and ethanol yielded the lowest maximum pressures and highest enthalpy heat of evaporation. For the economic condition, R290, RC318, R600a, methanol and ethanol are feasible for a large capacity ORC system.

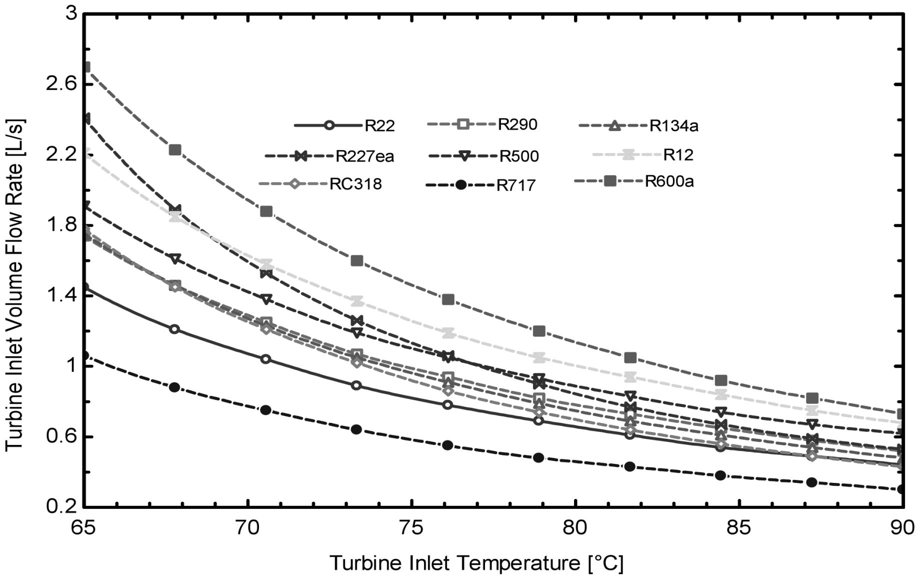

Figure 11. Variation of turbine outlet volume flow rate for low-temperature solar ORC working fluids with a turbine inlet temperature at a condensing temperature of 45°C.

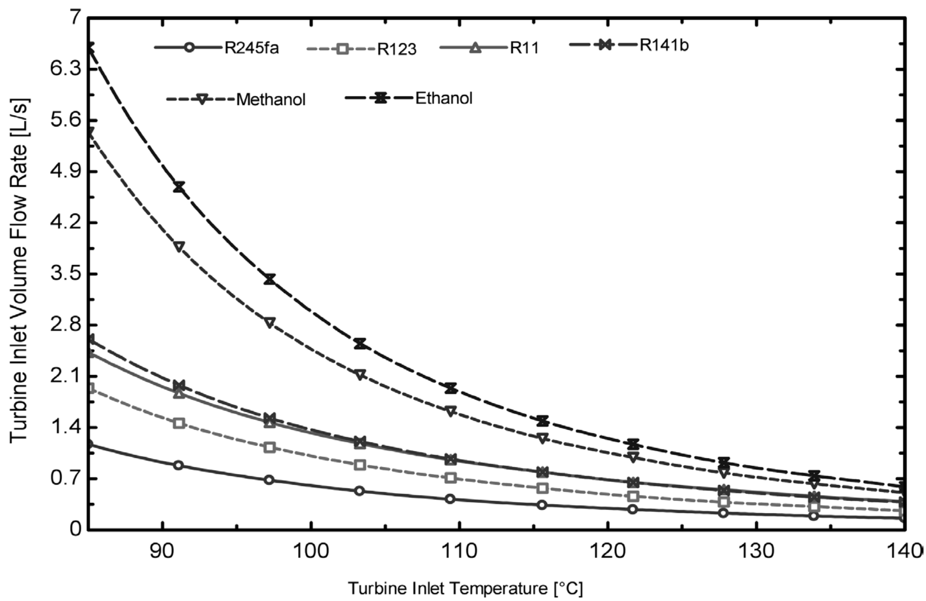

Figure 12. Variation of turbine outlet volume flow rate for medium-temperature solar ORC working fluids with a turbine inlet temperature at a condensing temperature of 45°C.

Solar Collectors

The heat input to the system is significant in a solar ORC that determines the size of the collector and establishes a major part of the system cost. Therefore, solar applications will be more viable with fluids for which the amount of heat required is small. From Table 2, the heat required for a 1 kW power output falls in the range between 12-20 kW and 6-19 kW for the low-temperature and medium-temperature solar ORC system, respectively. In addition, at a higher turbine inlet temperature, the system requires less heat input, which can be explained by Figures 13 and 14. High temperature saturated vapors reduce the heat input. When designing a solar ORC, one could choose between a system with a larger collector area-low temperature and a system with a small collector area-high operating temperature depending on the application.

The area of solar collector in ORC system is determined by the heat input, net work done and solar insolation of the location for installation. From Figures 13 and 14, a high turbine inlet temperature requires less area for the solar collector for the system. For a 1 kW power output, the required area of the collector is in the range, 23-34 m^2, for the low-

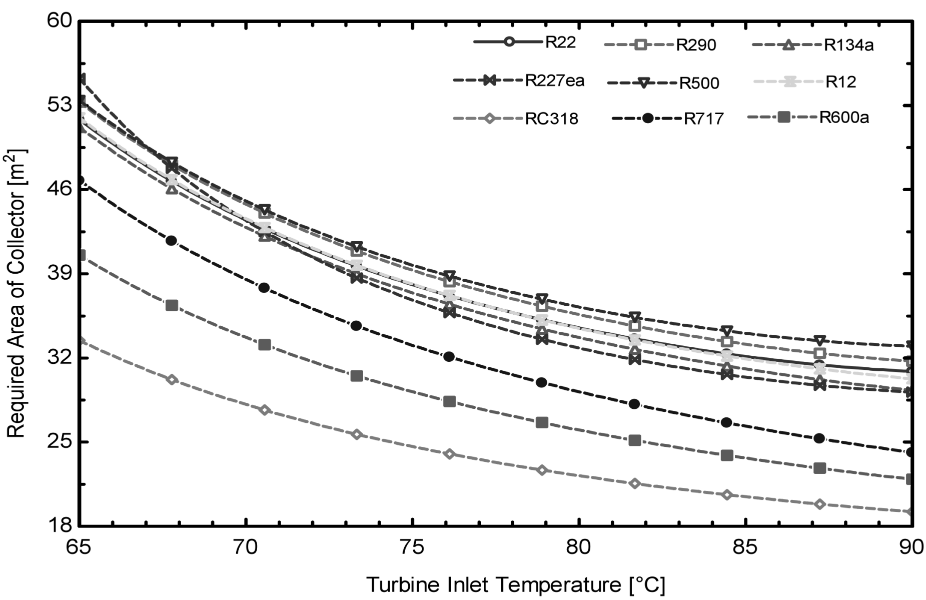

Figure 13. Variation of the area of collector for low-temperature solar ORC working fluids with the turbine inlet temperature at a condensing temperature of 45°C.

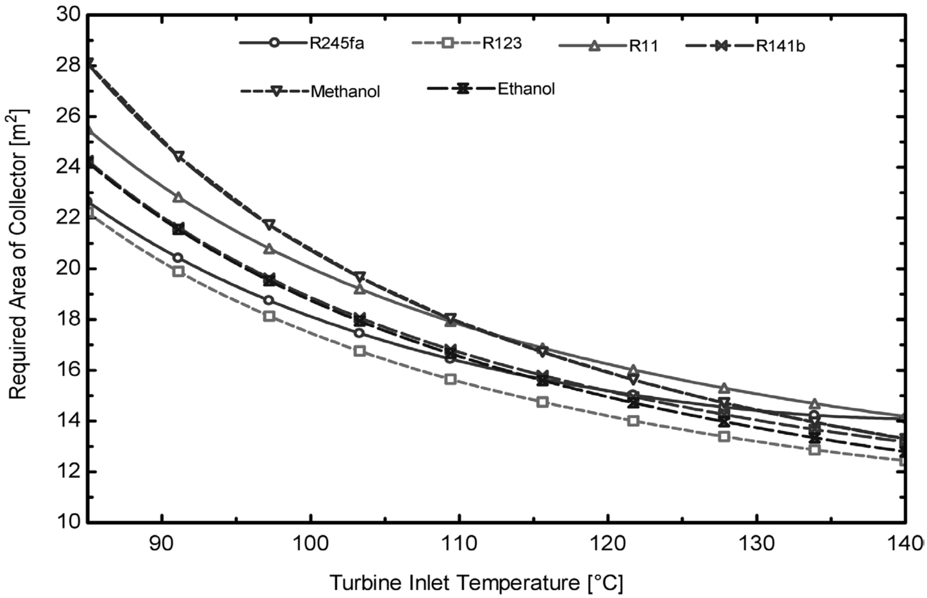

Figure 14. Variation of the area of collector for medium-temperature solar ORC working fluids with the turbine inlet temperature at a condensing temperature of 45°C.

temperature, whereas for the medium-temperature solar ORC, the range is between 14-17 m^2, as listed in Table 2. The collector area also constitutes a major part of the system cost. Figure 13 shows that the lowest required area of solar collector is in order RC318, R600a, and R717 in the low-temperature ORC. The lowest heat input fluid is R123 followed by R245fa, methanol, ethanol, R141b, and R11 for the medium-temperature solar ORC system as shown in Figure 14.

Hot Water Production

For the production of hot water, the area of the collector plays a vital role. The other factors include solar radiation, ambient temperature and solar collector efficiency. Hot water can be provided with a condensing temperature from the condenser of an ORC system. This concept of cogeneration can be applied widely when there is plenty of water in the located site of the ORC. In the present study, the obtained hot water temperature was fixed to 45°C. Table 3 lists estimates of the average daily hot water production at different months of the year from the required area of solar collector for working fluids. The meteorological conditions data of Busan,

Table 3. Hot water production for different working fluids in a year in Busan, South Korea.

Meteorological conditions for Busan, South Korea			Average daily hot water production during the month of year for various working fluids														
			Heat source at 90 °C									Heat source at 125 °C					
Month	Solar insolation (W/m^2)	Amb. temperature(°C)	R22	R290	R134a	R227ea	R500	R12	RC318	R717	R600a	R245fa	R123	R11	R141b	Methanol	Ethanol
Jan.	483	3	2795	2885	2705	2336	2962	2778	1800	2290	2059	1321	1235	1413	1320	1272	1301
Feb.	592	4	3422	3532	3311	2860	3626	3401	2203	2803	2521	1617	1512	1730	1616	1557	1592
March	703	8	4068	4198	3936	3399	4311	4042	2619	3332	2997	1922	1798	2056	1921	1851	1893
April	877	13	5070	5233	4906	4237	5373	5039	3265	4153	3735	2395	2241	2563	2394	2307	2359
May	928	17	5369	5541	5195	4487	5690	5336	3457	4398	3955	2537	2373	2714	2535	2443	2498
June	842	21	4868	5024	4710	4068	5159	4837	3135	3987	3586	2300	2151	2460	2298	2215	2265
July	733	24	4241	4377	4104	3544	4495	4215	2731	3474	3124	2004	1874	2144	2002	1930	1973
Aug.	718	25	4154	4288	4020	3472	4403	4129	2675	3403	3061	1963	1836	2100	1961	1890	1933
Sept.	620	22	3586	3701	3470	2996	3800	3563	2309	2937	2642	1694	1585	1812	1693	1632	1668
Oct.	593	17	3431	3541	3320	2868	3637	3410	2210	2811	2528	1621	1517	1734	1620	1561	1597
Nov.	472	12	2728	2815	2640	2280	2891	2711	1757	2235	2010	1289	1206	1379	1288	1241	1269
Dec.	440	6	2545	2626	2462	2127	2697	2529	1639	2085	1875	1202	1125	1286	1201	1158	1184
Daily average hot water production (Liter/ day)			3856	3980	3732	3223	4087	3832	2483	3159	2841	1822	1704	1949	1821	1755	1794

South Korea were obtained from Korea Meteorological Administration [19]. From this, a larger collector area and higher solar insolation produces more hot water. The production reached a maximum at April, May, June, July and August. The highest production is for R500, which has a collector area of 30.05 m^2 and produces 4087 liters per day of hot water followed by R290, R22 and R134a in a low-temperature solar ORC, whereas in the medium-temperature solar ORC, the highest production was obtained using R11, which is 1949 liters per day followed in order by R245fa, ethanol, methanol and R123. The hot water produced is stored in the insulated storage tank. The application of hot water in rural areas of developing countries are washing utensils, bathing, washing clothes as well as in small community-based hospitals, health posts and clinics.

Verification and Validation

For the verification and validation of simulation model the results were compared with the experimented data. The experimental data were obtained from a small scale ORC system using a scroll expander developed in our laboratory [20]. The ORC system used R245fa as the working fluid. The experiment was carried out with the evaporating pressure of 2520 kPa and condensing temperature of 45°C. The experiment was carried out with the setting mass flow at different rates. The heat source was hot water generated by an electric heater whose temperature was 130°C [20]. Figure 15 depicts a comparison of the simulated results and the experimental data. In general, both results agree well within the uncertainty range of experiment. However, the simulated efficiency is slightly higher than that of experiment. It can be explained that the internal losses of the real piping system and heat exchanger pressure losses are not taken into accounted in the system while carrying out the simulation. The other reason attributed to the expander's efficiency being fixed for whole calculation. In the real case, the expander's efficiency is not always constant. Additionally, there occurs irreversibility in the system. The trend of the simulated ORC system thermal efficiency versus working fluid mass flow rate shows a good agreement with the experimental data. The average deviation between experiment and simulation is 3.5% while the maximum deviation is 4.5% in correspondence of the higher values of working fluid mass flow rate. Moreover, it can be observed that ORC system efficiency values are in the range 5%-8%, in agreement with literature data for ORC systems characterized by a net electric power output in order of few kW. The experimental and simulated thermal efficiency follow the same behavior, even though the simulation model has a few simplifying assumptions as discussed earlier. Furthermore, it was revealed that the increase in working fluid mass flow rate resulted in higher thermal efficiency and shaft power output.

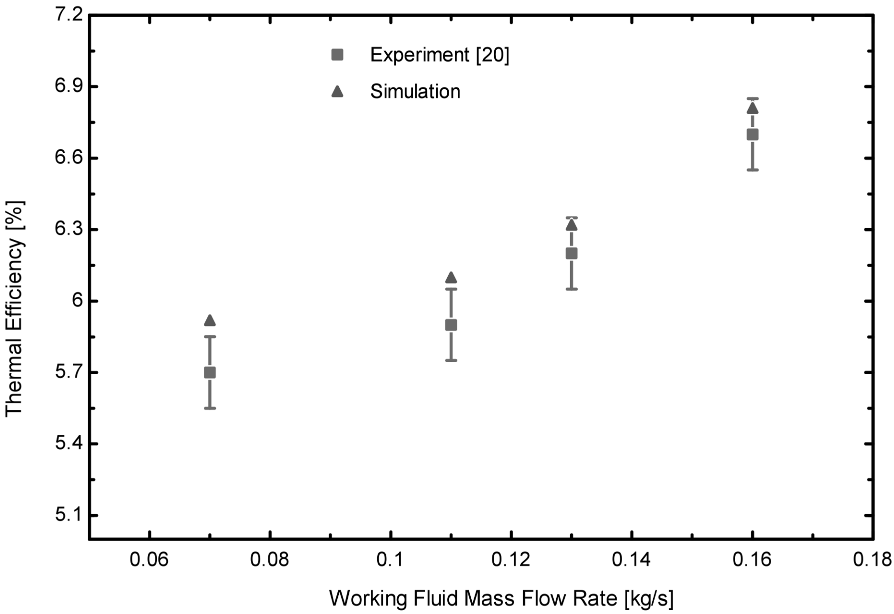

Figure 15. Comparison of thermal efficiency predicted by simulation and experimental data obtained in a small scale ORC system [20] for the validation of mathematical models.

Working Fluids Selection

According to the operating conditions and various parameters for the estimation of a 1 kW solar power output ORC, most of the working fluids were not accepted, even though the simulated results yielded high thermal efficiencies and pressure ratio. This is because of safety, toxicity, flammability and environmental characteristics of the working fluids. The safety criteria cannot be neglected. The terms of safety according to the ASHRAE are A1 (non-flammable and non- toxic), A2 (lower flammability and non-toxic), A3 (flammable and non-toxic), B1 (non-flammable but toxic), B2 (toxic and low flammability) and B3 (high toxic and high flammability). In Table 4, the decision rating criteria for working fluids have been recommended according to the manufacturer's operating conditions of a 1 kW power scroll expander, average value of exergy, thermal, solar cycle efficiencies, area of the solar collector, hot water production along with the minimal safety and environmental considerations. For the value of the parameter preferring the working fluid, it is marked as "√," and "x" vice versa. According the decision criteria rating in Table 4, the following working fluids were rejected, which are listed in Table 5: high ozone depletion potential (ODP) (R22, R500, R123, R11, R141b), high global warming potential (GWP) (R227ea, RC318), low safety (R290), high pressure ratio (Methanol, Ethanol), large collector area (R12), and low hot water production (R717,R600a).

Only two working fluids (R134a and R245fa) were finally acceptable for operating in a low-temperature and medium-temperature solar ORC system, whose heat source temperature are 90°C and 125°C, respectively.

CONCLUSIONS

A 1 kW solar organic Rankine cycle system was studied and modeled thermodynamically with 15 selected organic working fluids for low-temperature and medium-temperature heat sources based on its critical temperature, ranging from 150°C to 240°C. The analysis involved comparing various parameters, such as exergy, thermal, solar power cycle efficiencies in addition to the pressure ratio, mass flow rate, heat input, turbine inlet volume flow rate, the required area of solar collector and hot water production. The maximum exergy efficiency was 48.19% and 50.03% for RC318 and R123, respectively. The maximum turbine inlet pressure was 4037 kPa and 1929 kPa for R22 and R245fa, respectively. The maximum pressure ratio accounts for RC318 and ethanol. RC318 and R123 had the minimum heat inputs required. Ideally, a high efficiency, high pressure ratio, high hot water production, low mass flow rate and small area of collector are appropriate. On the other hand, many working fluids are rejected because of other factors, such as flammability, toxicity, and environmental conditions. Therefore, the most suitable working fluids are R134a and R245fa for low-temperature and medium-temperature solar ORCs, respectively. These recommended working fluids require an appropriate collector area size and the production of hot water is reasonable. This solar ORC can be mostly suitable for remote places of developing countries that lack electricity. Therefore, this technology is expected to be

10% Online Discount
Use Code JR10

HIGH PERFORMANCE BUILDINGS: A GUIDE FOR OWNERS & MANAGERS

Anthony Robinson

High Performance Buildings: A Guide for Owners and Managers, is a template - a blueprint for action for those making decisions about how to improve the energy efficiency and performance of new or existing buildings. It is designed to have broad appeal, both for the seasoned veteran facility or energy manager and for the new manager alike, but can also be utilized as a practical desk reference by professionals such as architects, engineers, and construction managers. The book provides an objective and orderly approach to what is often a complex, costly and time-consuming process. The full spectrum of topics relevant to achieving optimum building performance is addressed, including analysis of overall building energy use and performance, building commissioning, applicable codes, standards and rating systems, building envelope, onsite power generating options, optimizing performance of building mechanical and electrical equipment, and importance of effective building operation and maintenance practices. It is thorough in its topical scope, technically accurate, yet concise. Fundamental principles are discussed and illustrated with case studies.

ISBN: 0-88173-646-5

6 x 9, 396 pp., Illus.
Hardcover

$145
Order Code 0686

CONTENTS

BOOK ORDER FORM

1 Complete quantity and amount due for each book you wish to order:

Quantity	Book Title	Order Code	Price	Amount Due
	High Performance Buildings: A Guide for Owners and Managers	**0686**	**$145.00**	
			Applicable Discount	
			Georgia Residents add 6% Sales Tax	
			Shipping $10 first book $4 each additional book	**10.00**
			TOTAL	

2 Indicate shipping address: CODE: Journal 2013

NAME (Please print) BUSINESS PHONE

SIGNATURE (Required to process order) EMAIL ADDRESS

COMPANY

STREET ADDRESS ONLY (No P.O. Box)

CITY, STATE, ZIP

MEMBER DISCOUNTS—A 15% discount is allowed to AEE members (discounts cannot be combined).
☐ **AEE Member (Member No.________________)**

3 Select method of payment:
☐ **CHECK ENCLOSED**
☐ **CHARGE TO MY CREDIT CARD**
☐ *VISA* ☐ *MASTERCARD* ☐ *AMERICAN EXPRESS*

CARD NO.

Expiration date *Signature*

Make check payable in U.S. funds to:
AEE ENERGY BOOKS

4 Send your order to:
AEE BOOKS
P.O. Box 1026
Lilburn, GA 30048

INTERNET ORDERING
www.aeecenter.org/books
(use discount code)

TO ORDER BY PHONE
Use your credit card and call:
(770) 925-9558

TO ORDER BY FAX
Complete and Fax to:
(770) 381-9865

INTERNATIONAL ORDERS
Must be prepaid in U.S. dollars and must include an additional charge of $10.00 per book plus 15% for shipping and handling by surface mail.

Table 4: Decision rating criteria for the selection of working fluids

Sources Temperature	Decision rating criteria for working fluid selection in solar ORC co-generation system											
	P_{max} (kPa)	P_{min} (kPa)	PR	η_{exg} (%)	η_{th} (%)	η_{spc} (%)	A_c (m^2)	Safety Factor	ODP	GWP	V_{hw}(L/ day)	Remarks
Low-temperature	3000	1200	4	30	5.5	4	32	A1	0	1500	3700	Single Expander
Medium-temperature	1300	120	12	45	11.5	8	15	A1	0	1500	1800	Double Expander

Table 5: Decision criteria table for the selection of working fluids

Working Fluid	P_{max} (kPa)	P_{min} (kPa)	PR	η_{exg} (%)	η_{th} (%)	η_{spc} (%)	A_c (m^2)	Safet y Factor	ODP	GWP	V_{hw} (L/day)	Decision
R22	x	x	√	√	√	x	x	√	x	√	√	Rejected
R290	x	x	√	√	x	x	x	x	√	√	√	Rejected
R134a	√	√	√	√	√	√	√	√	√	√	√	**Accepted**
R227ea	√	√	√	√	√	√	√	√	√	x	x	Rejected
R500	x	x	√	x	x	x	x	√	x	x	√	Rejected
R12	√	√	√	√	√	x	√	√	√	x	√	Rejected
RC318	√	√	x	√	√	√	√	√	√	x	x	Rejected
R717	x	x	√	√	√	√	√	x	√	√	x	Rejected
R600a	√	√	√	√	√	√	√	x	√	n.a	x	Rejected
R245fa	x	√	√	√	√	√	√	x	√	√	√	**Accepted**
R123	√	x	√	√	√	√	√	x	x	√	x	Rejected
R11	√	√	√	x	x	x	x	√	x	x	√	Rejected
R141b	√	x	x	√	√	√	x	n.a	x	√	√	Rejected
Methanol	√	x	x	√	√	√	√	n.a	n.a	n.a	x	Rejected
Ethanol	√	x	x	√	√	√	√	n.a	n.a	n.a	x	Rejected

used for small distributed power generations systems and producing hot water with the same unit.

Acknowledgements

This study was supported by the National Research Foundation of Korea (NRF) grant funded by the Korea government (MSIP) through GCRC-SOP (No. 2011-0030663) and by the Korea Institute of Energy Technology Evaluation and Planning (KETEP) grant funded by MSIP (No. 2011201010030-11-2-300).

References

1. Chen H, Goswami DY, Stefanakos EK. A review of thermodynamic cycles and working fluids for the conversion of low-grade heat. *Renewable and Sustainable Energy Reviews 2010*; 14:3059-3067.
2. Tchanche BF, Lambrinos G, Frangoudakis A, Papadakis G. Low-grade heat conversion into power using organic Rankine cycles -A review of various applications. *Renewable and Sustainable Energy Reviews 2011*; 15:3963-3979.
3. Quoilin S, Orosz M, Hemond H, Lemort V. Performance and design optimization of a low-cost solar organic Rankine cycle for remote power generation. *Solar Energy 2011*; 85:955-966.
4. Saitoh T, Yamada N, Wakashima SI. Solar Rankine cycle system using scroll expander. *Journal of Environment and Engineering* 2007; 2:708-719.
5. Jing L, Gang P, Jie J. Optimization of low temperature solar thermal electric generation with Organic Rankine Cycle in different areas. *Applied Energy* 2010; 87:3355-3365.
6. He YL, Mei DH, Tao WQ, Yang WW, Liu HL. Simulation of the parabolic trough solar energy generation system with Organic Rankine Cycle. *Applied Energy* 2012; 97:630-641.
7. Delgado-Torres AM, García-Rodríguez L. Preliminary assessment of solar organic Rankine cycles for driving a desalination system. *Desalination* 2007; 216:252-275.
8. Pei G, Li J, Ji J. Analysis of low temperature solar thermal electric generation using regenerative Organic Rankine Cycle. *Applied Thermal Engineering*; 30: 998-1004.
9. Wang XD, Zhao L, Wang, JL. Experimental investigation on the low-temperature solar Rankine cycle system using R245fa. *Energy Conversion and Management* 2011; 52:946-952.
10. Twomey B, Jacobs PA, Gurgenci H. Dynamic performance estimation of small-scale solar cogeneration with an organic Rankine cycle using a scroll expander. *Applied Thermal Engineering* 2013; 51:1307-1316.
11. Manolakos D, Papadakis G, Kyritsis S, Bouzianas K. Experimental evaluation of an autonomous low-temperature solar Rankine cycle system for reverse osmosis desalination. *Desalination* 2007; 203:366-374.
12. Mago PJ, Luck R. Energetic and exergetic analysis of waste heat recovery from a microturbine using organic Rankine cycles. *International Journal of Energy Research* 2013; 30:926-938.
13. Bao J, Zhao L. A review of working fluid and expander selections for organic Rankine cycle. *Renewable and Sustainable Energy Reviews* 2013;24:325-342.
14. RayeganR, Tao YX. A procedure to select working fluids for Solar Organic Rankine Cycles (ORCs). *Renewable Energy* 2011; 36:659-670.

15. Gao H, Liu C, He C, Xu X, Wu S, Li. Performance Analysis and Working Fluid Selection of a Supercritical Organic Rankine Cycle for Low Grade Waste Heat Recovery. *Energies* 2012; 5:3233-3247.
16. Saleh B, Koglbauer G, Wendland M, Fischer J. Working fluids for low-temperature organic Rankine cycles, *Energy* 2007; 32:1210-1221.
17. Maizza V, Maizza A. Unconventional working fluids in organic Rankine-cycles for waste energy recovery systems. *Applied Thermal Engineering* 2001; 21:381-390.
18. Tchanche BF, Papadakis G, Lambrinos G, Frangoudakis A. Fluid selection for a low-temperature solar organic Rankine cycle, *Applied Thermal Engineering* 2009; 29:2468-2476.
19. Korea Meteorological Administration. Monthly and seasonal climate summary, 2013. Available online: http://www. web.kma.go.kr (accessed 08.13.13).
20. Yun E, Kim HD, Yoon SY, Kim KC. Development and characterization of small-scale ORC system using scroll expander. *Applied Mechanics and Materials* 2013; 291-294;1627-1630.

ABOUT THE AUTHORS

Suresh Baral is a PhD student in the School of Mechanical Engineering, Pusan National University, South Korea. His advisor is Prof. KC Kim. Mr. Baral received his bachelor and master degree in Mechanical Engineering from Institute of Engineering, Pulchowk Campus and Kathmandu University, Nepal respectively. He is permanent faculty member in Pokhara University, Nepal as a Lecturer. His areas of interests are renewable energy, energy conversion technology and Finite Element Methods. He can be reached at baral@pusan.ac.kr

Kyung Chun Kim (corresponding author) is a Professor in the School of Mechanical Engineering of Pusan National University in Korea. He obtained the Ph.D. degree from the Korea Advanced Institute of Science and Technology (KAIST), Korea, in 1987. He was selected as a member of the National Academy of Engineering of Korea in 2004. His research interests include 3D3C Micro-PIV, Bio-MEMS, turbulent flow measurements based on PIV/LIF, biomedical engineering, POCT development, wind turbines, and organic Rankine cycle system. He can be reached at kckim@pusan.ac.kr