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Solar Driven Organic Rankine cycle (ORC) – A Simulation Model

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Abstract

In this work, an ORC-process driven by solar heat is simulated, analyzed and dimensioned for reliable electricity generation in remote areas, especially for refugee camps. The most common fluid for ORC-systems is R245fa. In 2022 this fluid will not be available because of the f-gas decree from 2015. Butane, pentane, R1233zd(e), R1234ze(z) are fluids with low global warming potential and are simulated in the ORC-process to analyze the performance and potential to replace R245fa. From the performance results, R1233zd(E), R1234ze(z) and n-butane have the potential to replace R245fa in organic Rankine cycles with low temperatures. Evacuated tube collector with the heat pipe system in low flow is found to be the best collector type for this application. 60 m² ETC collector area is needed for the generation of 2 kW electrical power. The increasing thermal efficiency of the ORC with higher source temperatures has more influence than the decreasing collector efficiency. In conclusion, the solar driven organic Rankine cycle is a potential alternative off-grid power technology to generate reliable electricity in remote areas in the sunbelt area. The whole system as a container concept is simple, robust, low-maintenance, mobile and can power, for example, a medicate centrum and public light with 2kW.

Keywords: *Organic Rankine cycle, low temperature waste heat, working fluid, solar thermal energy*

1. Introduction

In many remote areas exist no electricity grid or it is very unreliable. The international energy agency reported 1.2 billion people without access to electricity in the world (WEO, 2015). According to the general secretary Ban Ki Moon, all people should have universal access to modern energy sources. People in developing countries should have access to renewable and environmentally friendly energy sources (se4all, 2016). Electricity is necessary to develop the economy and to improve the health and education situation. Many people without electricity live in the rising refugee camps. In the end of 2015, 63.5 million people refuge from war, poverty and religious or political persecution, increasing tendency (UNHCR, 2015). A big part of the refugees live in camps, with until 400000 people in one camp. The greatest camps are located in countries like Kenya, Ethiopia, Jordan, Gaza Strip, Mauritania, South Sudan, Uganda, Tanzania and India. In refugee camps 90 % of people have no access to electricity. (Leach, 2015) Big problems in these camps are the medical care without a reliable electricity supply and the dark nights without light and high criminal violation.

The most less-developed countries are located equatorial respectively in the sunbelt of the earth, therefore solar energy represents a big potential for these countries. A big problem by using the sun for electricity generation with photovoltaic is the unreliability in times without sun and high priced power stores. An alternative is to heat up water by solar collectors and store it in a boiler, to save energy for the few hours without sun. An organic Rankine cycle (ORC) generates reliable electricity with the heated water out of the storage. ORC-processes in connection with solar heating and a heat reservoir have the potential for safe and environmental friendly electricity-generation in these camps and remote villages. With the right dimensioning of collector field and heat reservoir it is possible to generate enough electricity for medical equipment and light supply. Also a thinkable task is the stability of isolated electricity grids (rotated masses and black start) and energy supply in public buildings, server rooms and hospitals. Many new projects exist at the moment in modernization and new building of hospitals in less-developing countries in Africa.

A study on a 3kW solar driven ORC with concentrated collectors and temperatures around 150°C is published by Orosz et al. (2009). Organic Rankine cycles have usually small thermal efficiencies, because of the usage of

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low temperature heat sources with a small exergetic part of energy like solar thermal systems. The design of the components and the choice of a suitable working fluid are important to ensure a proper working process without any losses in the efficiency. Studies on the fluid selections for ORC-processes especially driven by a solar heat source are made from researchers like Tchanche et al. (2008), Rayegan and Tao (2011), Delgado-Torres and Garcia-Rodrigez (2010). They all did not analyzed new environmental friendly refrigerants like R1233zd(e) and R1234ze(z).

The biggest challenge to design an ORC-process is the selection of the working fluid and the design of the heat exchanger for the application in dependence of specified temperature levels. In this work, the heat source is solar energy and usually bounded in temperatures between 20°C and 180°C, depending on collector type, without considering concentrated systems.

In this paper, an ORC driven by solar heat is simulated, analyzed and dimensioned for reliable electricity generation in remote areas especially for refugee camps. The whole system should be simple, robust, with a low-maintenance and mobile with 2kW electric power for a medicate centrum and public light. The most common fluid for ORC-systems is R245fa. In 2022 this fluid will not available because of the f-gas decree from 2015 (EG-VO 517, 2014). Different working fluids in the ORC-process for the temperatures of a designed solar system are simulated to analyze the performance of low GWP working fluids for the potential replacement of R245fa.

2. System description and dimensioning

The whole system is dependent from the climate and weather conditions. Refugee camps located in the sunbelt and offer a high potential in the usage of solar energy. As reference in this paper, Aleppo is chosen from meteonorm software. The ambient temperature and the radiation in hour values are relevant. The system schematic is shown in figure 1. On the right side, the ORC is illustrated; on the left side the solar system. The systems are connected by the evaporator as heat exchanger and a storage tank for the compensation of hours without sun. It is also possible to operate directly without storage in hours with maximum turbine power output and high sun radiation to raise the efficiency.

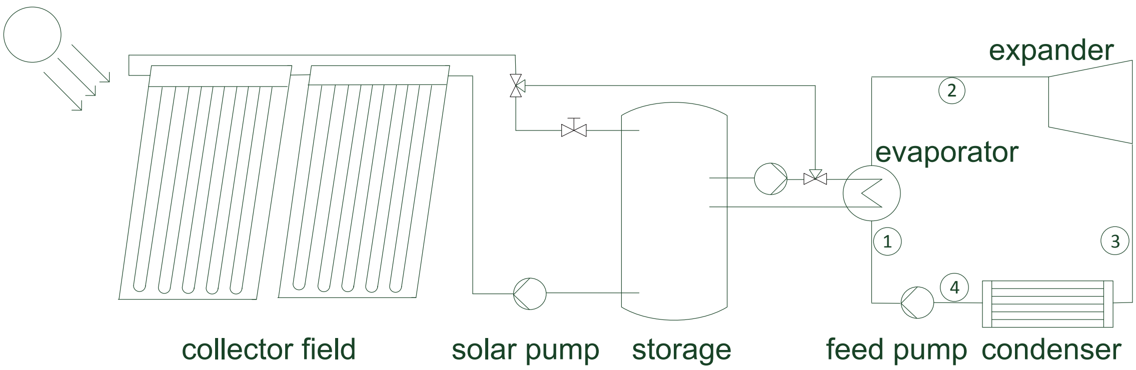


Fig. 1: Overview of the electricity-generation-system with solar heat and ORC

Solar collector:

Because of the limitation in Carnot efficiency and the dependence of exergetic and thermal efficiency on temperature [Theede and Luke, 2016], higher temperatures are preferable for the ORC. To avoid complex collectors with concentrated systems and to use water for a simple system, an evacuated tube collector (ETC) with the heat pipe system is chosen. This collector type has a high operation safety because of the easy replaceable tubes, need less space than flat collectors, has a freezing protection and has a higher performance at high temperatures, because of the better isolation. Collector outlet temperatures are in the rage of 80°C and 140°C. The usage of water at temperatures of 140°C results in a pressure level of maximum 4 bar to avoid evaporation with low heat transfer coefficients. The low flow operation is chosen to generate high temperatures in a big solar system. In this work, the mass flow is calculated to 0.28kg/s in the simulation with an assumed flow rate of 20 l/m²h and a first evaluated collector area of 55m². The fluid velocity is fixed at 1 m/s to avoid high pressure losses in the pipes. The commercial solar collector named Vitosol-T SPA is chosen for the dimensioning and simulation. It is an ETC working on the heat pipe principle. Other collector data with the optical efficiency and heat loss coefficients for the calculation of the collector efficiency are listed in table 1.

The collector efficiency is calculated in

$$\eta_{Coll} = \eta_{op} - \frac{a_1 \cdot (T_m - T_a)}{G} - \frac{a_2 \cdot (T_m - T_a)^2}{G} \qquad \text{(eq. 1)}$$

and depends on the collector a_1, a_2 and η_0 data on the mean average fluid temperature in collector T_m, the ambient temperature T_a and the sun radiation G.

In Figure 2, the characteristic of the ETC is shown. The efficiencies are calculated with radiations between 200 W/m² and 1200 W/m² and an average ambient temperature of 18°C in Aleppo. The collector efficiency decreases with higher collector temperatures and lower sun radiation. The less strong decreasing efficiency of the ETC in comparison to the decreasing efficiency with higher collector temperature in common flat collectors is conspicuous. Because of the good insulation with the vacuum, this is a big advantage of this collector type.

The product of the hour values of sun radiation from meteonorm and calculated collector efficiencies are taken and summarized to the solar thermal heat energy per square meter. In Aleppo, this value was 1875 kWh/m²a in 2003. This value shows the big potential in the usage of solar thermal energy in this region.

Organic Rankine Cycle:
The ORC-process consists of four components similar to the conventional Rankine cycle, but uses a refrigerant as working fluid to enable the generation of electricity from heat with low temperatures. The feed pump raises the pressure for the evaporation. The fluid is heated up, evaporated completely and superheated in the evaporator. The turbine expands the vapor and generates work, which is transferred in the generator in electricity. Finally, the vapor is condensed at the lower pressure.

The ORC-process should have 2kW of electrical power and generate electricity with a source temperature between 80°C and 130°C. The temperature spread in heat exchangers are 10 K and the evaporator is designed with superheat of 5 Kelvin. The evaporator uses a coaxial heat exchanger because of the simple construction. A tube bundle heat exchanger is not worth because of the slow fluid velocity in the solar process.

Tab. 1: collector data (VITOSOL 200-T SPA)[Q]

gross surface	4.62 m²
absorber surface	3.03 m²
aperture surface	3.13 m²
width	2.061 m
height	2.241 m
optical efficiency η_0	0.81
heat loss coefficient a_1	1.103 W/m²K
heat loss coefficient a_2	0.007 W/m²K
permissible pressure	6 bar

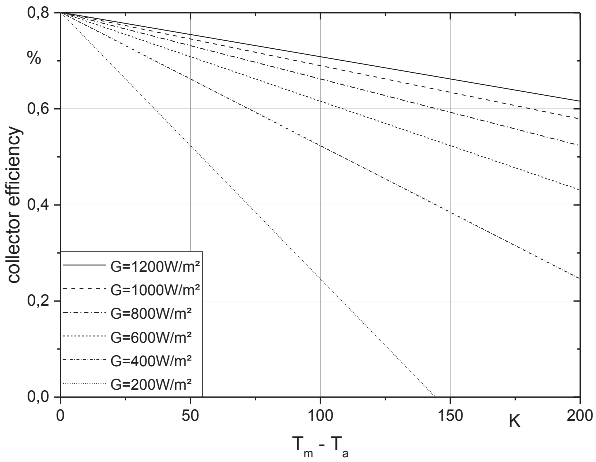
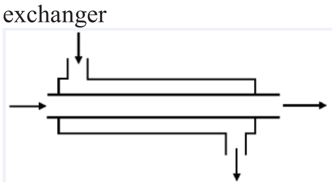
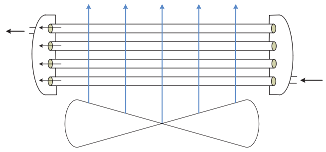


Fig. 2: collector efficiency characteristic of ETC

turbine inlet temp.: 130°C power output: 2 kW isentropic efficiency: 80%	evaporator type: coaxial heat exchanger 	condenser type: tube bundle with cross streamed air 
feed pump isentropic efficiency: 70%		
heat source (solar) fluid: water inlet temp.: 140°C pressure: 4 bar velocity: 1 m/s mass flow: 0.28 kg/s	super-heat: 5K velocity: 2 m/s ambient conditions temp.: 18°C pressure: 1 bar	number of tubes: 100 (shifted) inner diameter: 0.5 cm distance between tubes: 1 cm Condensation temp.: T_a+10K air velocity: 20 m/s

A tube bundle heat exchanger with cross streamed air is chosen for the condenser, because of the lack of water as a coolant in remote areas. So the re-cooling temperature is the average ambient air temperature (18°C in Aleppo) plus the temperature spread in the heat exchanger. Many fluids are reviewed in the temperature range until 90°C [Theede]. The best fluids are found to be R245fa and n-butane. This fluids, n-pentane and two new environmental friendly fluids R1233zd(e) and R1234ze(z) with a low global warming potential are analyzed in this paper. The design conditions are summarized in table 2.

3. Simulation tool

The simulation, realized with the engineering equation solver (EES), is based on change of state in the thermodynamic cycle and is calculated with energy balances, equations of states and the definitions of the isentropic performance of expander and feed pump in equation (3) until (8).

turbine: $P_T = \dot{m} \cdot (h_3 - h_2)$ (eq. 3) $\eta_{s,T} = \frac{h_3 - h_2}{h_{3s} - h_2}$ (eq. 4)

feed pump: $P_P = \dot{m} \cdot (h_1 - h_4)$ (eq. 5) $\eta_{s,P} = \frac{h_{1s} - h_4}{h_1 - h_4}$ (eq. 6)

condenser: $\dot{Q}_C = \dot{m} \cdot (h_3 - h_4)$ (eq. 7)

evaporator: $\dot{Q}_E = \dot{m} \cdot (h_2 - h_1)$ (eq. 8)

where P is power of turbine or feed pump, $\dot{Q}$ is heat transfer rate, $\dot{m}$ is the mass flow of the working fluid, h is the enthalpy in the points of the process, calculated with the equation of states of the fluids. The pressure losses are neglected and evaporation and condensation are assumed to be isobaric. The heat exchangers are simulated according to the kinetic. The simplified simulation structure for the main process is shown in figure 3. The code calculates all working conditions and the parameters of the components.

The results of the thermodynamic process in equation (3-8) are used to calculate the thermal and exergetic efficiency:

$$\eta_{th} = \frac{P_T}{\dot{Q}_E + P_P} \quad \text{(eq. 9)}$$

$$\eta_{ex} = \frac{P_T}{\frac{T_3 - T_a}{T_3} \cdot \dot{Q}_E + P_P} \quad \text{(eq. 10)}$$

T_3 is the turbine inlet temperature and T_a is the ambient temperature. The other simulation parameters are reported in table 2. The operating conditions of the heat source are taken from the solar collector calculations.

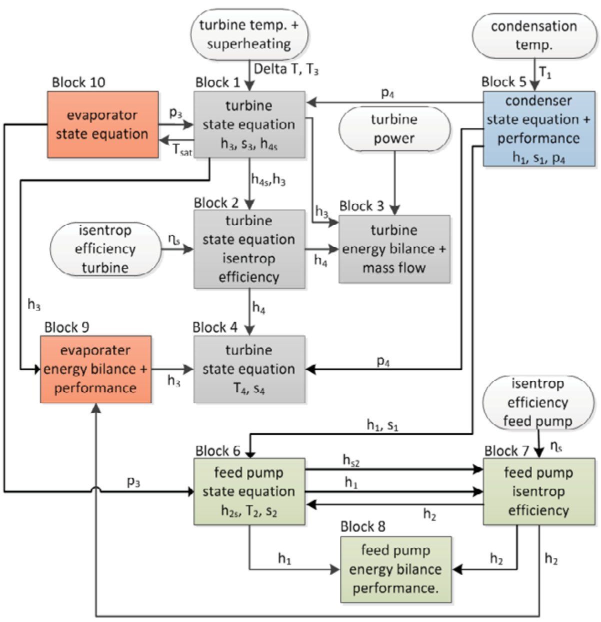


Fig. 3: simplified simulation structure of the ORC

A coaxial heat exchanger is set for pre-heating, evaporation and super-heating and a tube bundle condenser with cross streamed air for desuperheating and condensation. The thermodynamic parameters of the heat exchangers are numerically calculated with a segment-method. The different heat transfer coefficients are calculated and generate the overall heat transfer coefficient for the cell with

$$\frac{1}{k} = \frac{d_o}{d_i \cdot \alpha_{prim}} + \frac{d_o}{2 \cdot \lambda_{Co}} \cdot ln\frac{d_o}{d_i} + \frac{1}{\alpha_{sec}} \qquad \text{(eq. 11)}$$

for each cell with a default length, where k is the overall heat transfer coefficient, d_o outer diameter, d_i inner diameter, α_{prim} heat transfer coefficient inside the pipe, α_{sec} heat transfer coefficient on the outside of the pipe and λ_{Co} the thermal conductivity of copper. At the same time, conditions for the next segment like the new temperatures, vapor quality and the heat flux are calculated. The heat flux in the first segment is simulated with a loop of the first cell that compares the heat fluxes of two consecutive loop cycles until the error is negligible.

In this work, correlation of Steiner (Kind and Saito, 2013) are chosen to calculate the heat transfer coefficients for flow boiling inside the evaporator and the correlation of Gnielinski (2013) for pre- and super-heating plus the heat transfer in the gap (solar fluid) in the coaxial heat exchanger are chosen. The condensation correlation from Thome et al. (2003) is used in the inner tube flow condensation and Gnielinski (2013) for the circulated flow of the air around the tubes. The desuperheater is calculated by Gnielinski (2013) for flow in tubes.

4. Results and discussion

Some primary results of the simulation and conditions of the fluids are reported in table 3. The critical point of the fluids is between 150.1°C and 196.5°C. The differences in the critical temperature of R245fa, n-butane, R1233zd(e) and R1234ze(z) lie in a small range of 15.5 K. R245fa is in many reviews the best and common refrigerant for ORC-process with low temperature heat sources. The F-gas decree from the January 1st, 2015 [EG-VO 517, 2014] prohibits the use of refrigerants with a GWP higher than 150 at 2022. So R245fa will not available for ORC projects in future. The GWP from n-butane, n-pentane, R1233zd(e) and R1234ze(z) are 5 or lower and could be an alternative to replace R245fa.

The pressure levels in evaporation and condensation in the design point are presented in table 3. Pressure levels in evaporation in dependence of the turbine inlet temperature are shown in figure 4. The evaporation and condensation pressure level of the fluids are similar. Only n-pentane has a significant lower vapor pressure curve. The mass flow of process and of different fluids in design point are between 25 g/s (pentane) and 89 g/s (R1234ze(z)). The differences in mass flow are due to the enthalpy difference in the turbine between the given condensation and evaporation pressure level in equation 3. The enthalpy range of the hydrocarbons is higher than the other fluids. R1234ze(z) with the highest mass flow has a significant small enthalpy difference in the turbine. A high mass flow requires bigger components and more use of material, but in this case all mass flows are very low. The feed pump depends on the pressure difference between low and high pressure and the mass flow.

The heat exchanger are the biggest elements of the ORC. Depending on the working fluid, the evaporator and condenser have different heat transfer areas. In figure 5 the evaporator heat transfer surface in dependence of the turbine inlet temperature is shown. The differences in the volume of the evaporator depend mainly in the heat transfer coefficient. In evaporation, the pressure level has the biggest effect in the value of heat transfer coefficient. The increasing evaporation pressure with higher turbine inlet temperatures (see fig. 4) results in a decreasing heat transfer surface in the evaporator with increasing temperatures (see fig. 5). Butane and R1234ze(z) with the highest evaporation pressures have the smallest evaporator. The fluid properties, in particular the thermal conductivity, have the biggest influence on heat transfer coefficient in condensation.

Tab. 3: ORC results of the simulation and fluid conditions on the design point

Fluid [-]	critical temp. [°C]	pressure evap. [bar]	pressure cond. [bar]	massflow [kg/s]	pump power [W]	HX(evap.) surface [m^2]	HX(cond.) surface [m^2]	GWP [-]
R245fa	154	21.25	1.65	0.052	53.1	0.181	3.723	1030
n-butane	152	24.13	2.68	0.027	70.1	0.175	3.695	4
n-pentane	196.5	9.98	0.77	0.025	26.2	0.199	3.274	5
R1233zd(e)	165.6	17.33	1.44	0.051	45.2	0.194	3.662	1
R1234ze(z)	150.1	22.32	1.98	0.089	60.3	0.175	3.603	< 1

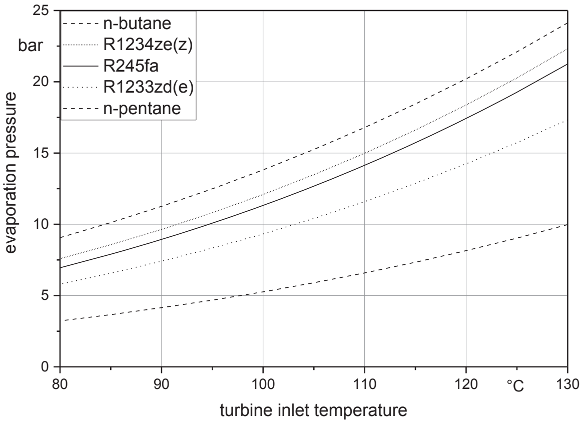


Fig. 4: evaporation pressure in dependence of turbine inlet temperature

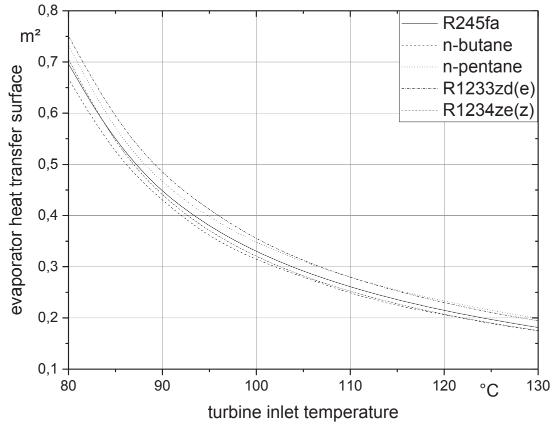


Fig. 5: evaporator heat transfer surface in dependence of the turbine inlet temperature

The design point of the evaporator can be fixed at medium collector temperatures. In ORC-operation driven by the heat of the boiler with lower temperatures, the extra pump can produce (see fig. 1) a higher heat transfer coefficient on the secondary side of the heat exchanger by raising the mass flow.

In table 4 the efficiency results of the ORC, the needed thermal power and energy for 2kW turbine power plus the efficiency of the whole system and the resulting collector area in the design point at 140 degrees source temperature is shown. The thermodynamic efficiency of the ORC-process in dependence of the turbine inlet temperature is shown in figure 6. The thermal efficiency depends on the turbine power, feed pump power and the transferred heat in the evaporator and increases with a slight digressive trend. The difference between the thermal efficiency of the best fluid (R1234ze(z)) and the worst fluid (n-pentane, R245fa) in design point is about 3.85%. These small differences are due to the similar physical characteristics of the fluid. R1234ze(z) has the best thermal efficiency (15.6%) because of the smallest difference of the evaporation temperature to the critical point. So the relation between specific heat capacity and the evaporation enthalpy is the smallest.

Tab. 4: The results of the simulation on the design point

Fluid	thermal efficiency (ORC)	exergetic Efficiency (ORC)	heat flux evaporator	thermal energy	system efficiency* (ORC+solar)	collector area
[-]	[%]	[%]	[kW]	[MWh/a]	[%]	[m²]
R245fa	15.0	53.5	13.27	116.25	9.59	61.99
n-butane	15.2	53.9	13.10	114.76	9.71	61.19
n-pentane	15.0	53.8	13.28	116.32	9.61	62.02
R1233zd(e)	15.4	54.8	12.98	113.68	9.82	60.62
R1234ze(z)	15.6	55.5	12.76	111.80	9.97	59.61

* sun radiation 800 W/m²

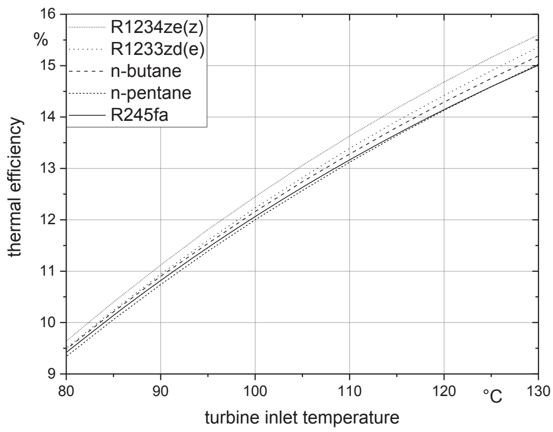


Fig. 6: thermal efficiency of the ORC-process in dependence of the turbine inlet temperature

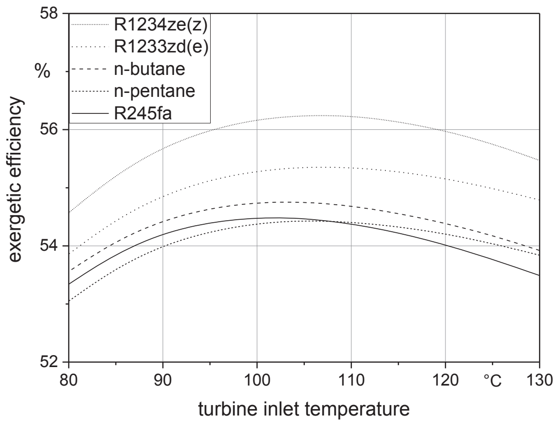


Fig. 7: exergetic efficiency of the ORC-process in dependence of the turbine inlet temperature

The exergetic efficiency of the fluid in dependence of the turbine inlet temperature is shown in figure 7. The exergetic efficiency in equation 10 includes the quality of the heat source (solar) in relation of the ambient temperature. With increasing temperature, the exergetic efficiency increases first, because the increasing exergetic part of the transferred heat is higher than the efficiency loss than the increasing pump power. After the point with the maximal exergetic efficiency at ca. 105°C, the increasing pump power has more influence in the exergetic efficiency and the value decreases. All differences in exergetic efficiency of the fluids with fixed turbine power and fixed ambient temperature depend on the feed pump power and the transferred heat in the evaporator (preheating, evaporation, superheating). The difference between the exergetic efficiency of the best fluid (R1234ze(z)) and the worst fluid (R245fa) in design point is about 3.60%. R245fa has the worst exergetic efficiency because of a combination of a higher relation of specific heat capacity and evaporation enthalpy, a higher feed pump power and a high enthalpy difference between high and low pressure combined with more preheating. The performance data of R1233zd(E) and R1234ze(z) in the respective temperature range are comparable to or better than the performance data of R245fa. R1233zd(E) and R1234ze(z) shows well performance results in figure 6 and 7 to have the potential to replace R245fa in ORC future.

The solar thermal energy of the chosen collector in Aleppo (2003) calculated with hour values in the design point equals 1875.35 kWh/m²a. Other thermal energy values for different collector outlet temperatures are listed in table 5. The thermal energy increases with lower temperatures, because of the rising collector efficiency (see in equation 1).This quantity of thermal energy is in good agreement with other works [Kalogirou, 2002]. The required heat to generate 2kW electricity over the whole year is reported in table 4. The quotient of the thermal energy and the solar thermal energy per m² equals the needed collector area. The collector area in dependence of the source temperature is plotted in figure 8. The collector area decreases with higher temperatures because of the rising thermal efficiency of the ORC. The trend is digressive because of the decreasing collector efficiency with higher collector temperatures.

Tab. 5: solar thermal energy

temperature [°C]	solar thermal energy [kWh/m²a]
140	1875.35
130	1899.33
120	1921.90
110	1942.97
100	1962.54
90	1980.61

Tab. 6: collector and system efficiency in design point for R1234ze(z)

sun radiation [W/m²]	collector efficiency [%]	system efficiency [%]
600	59.61	9.13
800	63.92	9.97
1000	67.15	10.48

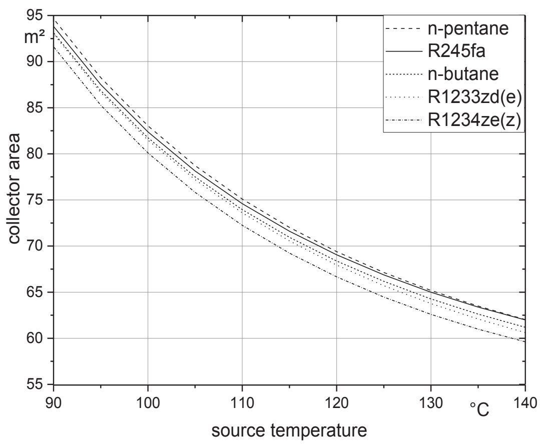


Fig. 8: collector area in dependence of the source temperature

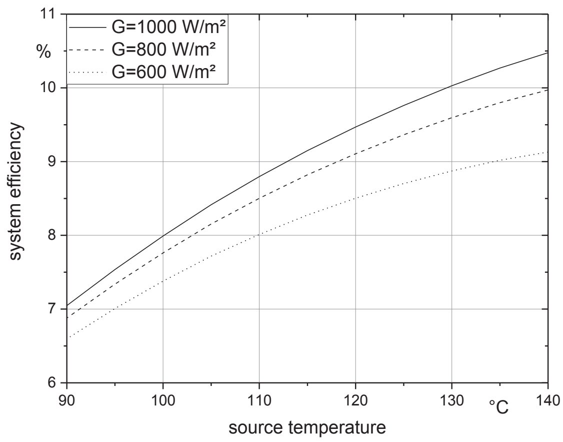


Fig. 9: system efficiency in dependence of the source temperature

In table 6 the collector efficiencies and the system efficiencies in three cases of sun radiation are listed. The values are generated in the design point and for the fluid R1234ze(z). Values for other fluids are listed in table 4 with a fix sun radiation of 800 W/m². The product of the collector efficiency and the thermal efficiency of the ORC equals the system efficiency. The system efficiency increases with higher sun radiations because of increasing collector efficiency. The dependence of the system efficiency with varying source temperature is plotted in figure 9. The system efficiency increases with the source temperature.

The trend is digressive because the collector efficiency is the product of the collector efficiency and the thermal efficiency of the ORC equals the system efficiency. The system efficiency increases with higher sun radiations because of increasing collector efficiency. The dependence of the system efficiency with varying source temperature is plotted in figure 9. The system efficiency increases with the source temperature. The trend is digressive because the collector efficiency is decreasing with higher temperatures. So the influence of the increasing thermal efficiency of the ORC in figure 6 is higher than the decreasing collector efficiency in figure 2. The trend of the system efficiency is more digressive with lower radiations because of the relation of the strong degreasing collector efficiency with lower sun radiation (figure 1). The trend of the system efficiency with 600 W/m² radiation is more decreasing and reaches nearly the maximum point.

In the design point with the selected ETC the system needs 59.6 m² collector absorber area with the fluid R1234ze(z). The absorber area of a ETC has 3.03 m² with 24 heat pipes (see table 1). So the whole system needed 20 ETCs, respectively 480 heat pipes to generate 2 kW electricity over the year. That's a value of 29.8 m²/kWe.

The whole system should be designed simple, robust, low-maintenance and mobile for refugee camps in remote areas. Solar thermal energy and the orc system are much tested and both systems need, with a good control engineering system, no supervision personnel. To make the system simple and mobile an ISO-container is most suitable (see in figure 10). The solar system is installed on the roof of the container and the ORC-process and water boiler are installed in the container. The transport goes by ship and truck. An ISO-container has a volume of 67.4 m² and a roof area of 28.3 m² (12.032 m length, 2.352 m width, 2.382 m high).

An optimal collector slope is the geographic latitude minus ca. 10 degree (Eicker, U., 2011). Aleppo with geographic latitude of 36.2 degree has an optimal collector slope of 26.2 degree. With a space factor of 2.2, four collectors take place on one container in the operation. With an installed rail system 10 collectors take place on one container. Two containers get enough space for a collector field of a 2 kW solar-ORC-power-system. The boiler can be free scaled, because there is enough space in the container. The ORC and boiler take place in one container and the other container can use for service quantities and other technical systems to compensate the hours were no electricity is needed. A potential technology is a water treatment system.

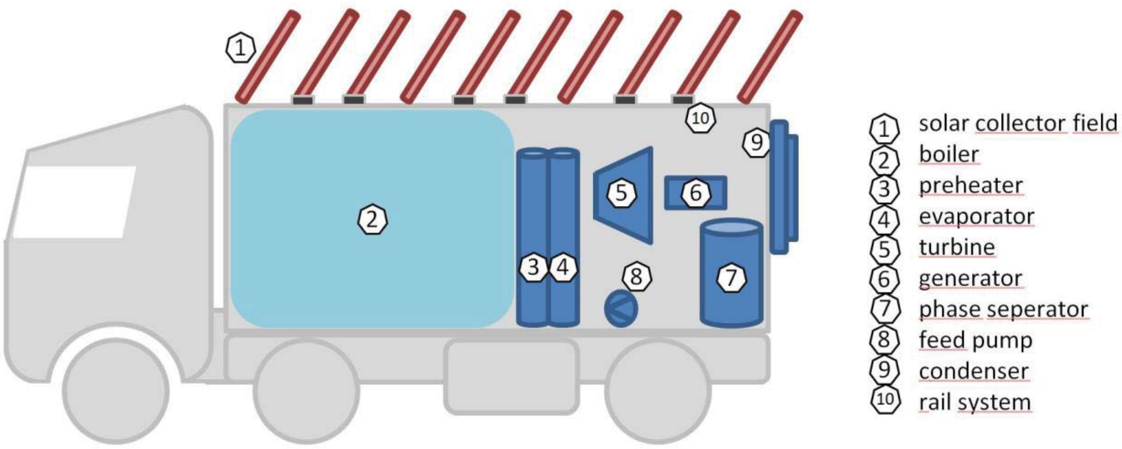


Fig. 10: schematic drawing of the solar driven ORC as container system

5. Conclusions

An ORC-process driven by solar heat is simulated, analyzed and dimensioned in this work for reliable electricity generation in remote areas, especially for refugee camps. ORC-processes in connection with solar heating and a heat reservoir plus a high value of sun radiation in remote areas in the sunbelt have the potential for safe and environmental friendly electricity-generation. The whole system as a container concept is simple, robust, low-maintenance, mobile and can power for example a medicate centrum and public light with 2kW. Evacuated tube collectors with the heat pipe system in low flow are chosen as the best collector type for theses application because of the easy replaceable tubes, higher performance at high temperatures (better isolation). 60 m² ETC collector area is needed for the generation of 2 kW electrical power. This value increases with lower temperatures. The system efficiency is increasing with higher source temperatures. Therefore the increasing thermal efficiency of the ORC with higher source temperatures has more influence than the decreasing collector efficiency. The most common fluid for ORC-systems is R245fa. In 2022 this fluid will not be available because of the f-gas decree from 2015. Four different fluids (n-butane, n-pentane, R1233zd(e), R1234ze(z)) with low GWP for the temperatures of a designed solar system are simulated in the ORC-process to analyze the performance and potential to replace R245fa. All fluids produce similar results in terms of evaporation pressure and thermal respectively exergetic efficiency. Only n-pentane has lower efficiencies and evaporation pressures in comparison to the other fluids. A local maximum point for the exergetic efficiency is found. After this point, the increasing pump power with higher temperatures has more influence as the raising exergetic part of solar heat. R1234ze(z) offer a high mass flow because of the low enthalpy difference between high and low pressure in the turbine. From the in performance results R1233zd(E), R1234ze(z) and n-butane have the potential to replace R245fa in organic Rankine cycles with low temperatures. Other analysis concerning thermal stability, security and economical aspect should be made in future. Also, the pressure losses should be analyzed. In conclusion, the solar driven organic Rankine cycle is a potential alternative off-grid power technology to generate reliable electricity in remote areas in the sunbelt.

6. References

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7. Nomenclature and symbols

a	heat loss coefficient [W/m²K]
d	diameter [m]
G	sun radiation [W/m²]
h	enthalpy [kJ/kg]
k	overall heat transfer coefficient [W/m²K]
$\dot{m}$	mass flow [kg/s]
P	power [W]
$\dot{Q}$	heat transfer rate [W]
T	temperature [K]
η	efficiency [-]

Subscripts

a	ambient
C	condenser
Co	copper
Coll	collector
E	evaporator
ex	exergetic
I	inner
o	outer
op	optical
P	feed pump
prim	primary
s	isentropic
sec	secondary
T	turbine
th	thermal