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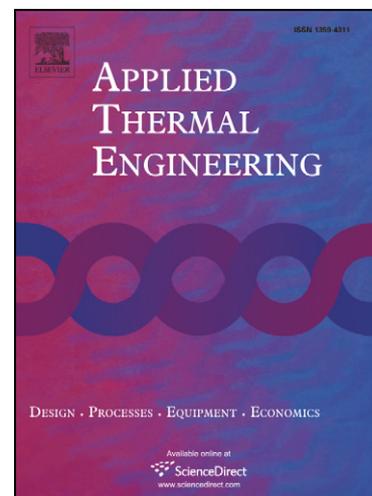
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# Energetic and economic investigation of Organic Rankine Cycle applications

A. Schuster<sup>a\*</sup>, S. Karellas<sup>b</sup>, E. Kakaras<sup>b</sup>, H. Spliethoff<sup>a</sup>

<sup>a</sup> *Institute for Energy Systems, Technische Universität München, Boltzmannstr. 15,  
85747 Garching, Germany*

<sup>b</sup> *Laboratory of Steam boilers and thermal power plants, National Technical University of  
Athens, Heroon Polytechniou 9, 15780 Athens, Greece*

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## Abstract

The use of organic working fluids for realizing the so called Organic Rankine Cycle (ORC) has been proven to be a promising solution for decentralized combined heat and power production (CHP). The process allows the use of low temperature heat sources, offering an advantageous efficiency in small-scale applications. This is the reason why the number of geothermal and biomass fired power plants based on this technology have been increased within the last years. The favourable characteristics of ORC make them suitable for being integrated in applications like solar desalination with reverse osmosis system, waste heat recovery from biogas digestion plants or micro-CHP systems. In this paper the state of the art of ORC applications will be presented together with innovative systems which have been simulated in a process simulation environment using experimental data. The results of the simulation like efficiencies, water production rates or achievable electricity production cost will be presented and discussed.

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\* Corresponding author: Tel: +49 89-28916269 Fax: +49 89-28916271  
email address: schuster@es.mw.tum.de (A. Schuster)

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## Nomenclature

### *Symbols*

A	Area	[m <sup>2</sup> ]
a	Annuity factor	[-]
A*	Membrane specific water permeability coefficient	[m/sbar]
A <sub>0</sub>	Investment	[€]
A <sub>N,K</sub>	Annuity of payments linked to capital	[€/a]
B*	Membrane specific salt permeability coefficient	[m/s]
c <sub>p</sub>	Specific heat capacity	[kJ/kgK]
C <sub>eff</sub>	Effective heat capacity of collector	[kJ/m <sup>2</sup> K]
epc	Electricity Production Costs	[€-Ct/kWh <sub>el</sub> ]
f <sub>k</sub>	Factor for repairs, maintenance and insurance	[-]
G <sub>b</sub>	Beam irradiance in the plane of collector	[W/m <sup>2</sup> ]
G <sub>d</sub>	Diffuse irradiance in the plane of collector	[W/m <sup>2</sup> ]
h <sub>fullload</sub>	Full load hours	[h]
i	Interest rate	[-]
k <sub>1</sub>	Collector loss coefficient	[W/m <sup>2</sup> K]
k <sub>2</sub>	Collector loss coefficient	[W/m <sup>2</sup> K <sup>2</sup> ]
k <sub>S</sub>	Salt mass transfer coefficient	[m/s]
K	Incident Angle Modifier	[-]
M	Molar mass	[kg/kmol]
m	Mass flow	[kg/s]
p	Pressure	[bar]

$\dot{q}$	Area related heat flow	[W/m <sup>2</sup> ]
$\dot{Q}$	Heat flow	[W]
R	General gas constant	[kJ/kgK]
t	Economic Life Time	[a]
t	Time	[s]
T	Temperature	[K]
v	Specific volume	[m <sup>3</sup> /kg]
w	Salt concentration	[kg/kg]

*Greek letters*

$\eta_0$	Zero loss collector efficiency	[-]
$\eta$	Efficiency	[-]
$\Pi$	Pressure ratio	[-]

*Subscripts*

amb	ambient
b	beam
c	critical
coll	collector
d	diffuse
el	electrical
F	Feed
L	longitudinal
mech	mechanical
mem	Membrane
osm	Osmotic

P	Permeate
R	Retentate
s	Saturation
S	Salt, Solute
T	transversal
$\theta$	Incident angle

## 1. Introduction

The Organic Rankine cycle (ORC) is a Clausius Rankine cycle in which an organic working fluid is used instead of water-steam. In the last years it became quite popular in energy production processes, due to the fact that it gives the possibility to use exhaust heat of low energy and temperature level. Fig. 1 shows the main components of the ORC process. The organic working fluid is compressed with a pump, which forces the fluid through a regenerator. The regenerator allows the preheating of the liquid working fluid by desuperheating the expanded vapour. The preheated working fluid is then evaporated, superheated and expanded in a turbine, which drives a generator. The desuperheated vapour is condensed in a condenser. If the temperature level of the condensation is high enough, as for example in the case of biomass combustion, the waste heat can be used in a district heating net. If low temperature heat is used for driving the ORC, like geothermal or waste heat, the condenser is cooled back by means of cooling water.

Most of the efforts in the literature include ways of maximizing the efficiency of the thermodynamic cycle for best waste heat recovery [1].

The use of ORC process for waste heat recovery has been applied for many years. An overview of manufacturers together with the appropriate temperature range of the applications is presented in table 1. In this table, also the expansion machine which is used for the superheated organic fluid vapour is presented.

It can be seen, that in the cases of low power output the use of screw motors is very common for the expansion, because turbines have in the small power range high gap losses.

## 2. Working fluids

The selection of the working fluid plays a significant role for the use of ORC process and is determined by the application and the waste heat level [2].

Saleh et al. [3] has presented the screening of 31 pure components working fluids for ORC process. In table 2 some selected fluids and their characteristics are presented.

The fluids are given in the order of rising critical temperature  $T_c$  and normal boiling temperature  $T_{s, 1 \text{ bar}}$ , whereas the critical pressure  $p_c$  and the vapour pressure at 20 °C decline. This gives a first hint for the application of these working fluids. Fluids with higher critical temperature allow on the one hand higher boiling temperatures but on the other hand lower pressures and thereby lower pressure differences are reached.

The fluids presented in table 2 are selected in order to cover a wide range of temperatures. Their critical temperature ranges from 101,3 °C for R134a up to 280,5 °C for cyclohexane.

Fig. 2 shows a T-s-diagram of the properties of water steam together with Cyclohexane and R245fa with isobars of 1, 10 and 25 bar for all three fluids. The surface enclosed from liquid and vapour saturation line is a degree for the achievable enthalpy differences in the thermodynamic cycle. The enthalpy differences for

organic substances are significantly lower compared with water. This implies bigger mass flows for the same power output.

The ORC process can work with saturated vapour or with a constant superheating of a few Kelvin depending on the fluid. Higher superheating in order to avoid liquid in the exhaust vapour is not necessary, because the expansion ends for the most fluids in the area of superheated vapour in contrast to water.

Higher superheating of the vapour is favourable for higher efficiencies, but because of the low heat exchange coefficients this would lead to very large and expensive heat exchangers.

A further important point that has to be considered besides the thermodynamic properties are legislative measures and regulations concerning the working fluid. The protection of the ozone layer and the emission reduction of greenhouse gases caused various regulations on international, European and national level. On an international level, the Montreal Protocol on Substances [4] that deplete the Ozone layer is the most important agreement. It controls the production and utilization of ozone depleting substances like halogenated CFCs; a complete list of the controlled substances can be found in the annex of the protocol. For the European level the regulation 2037/2000 [5] is the legal binding implementation of the protocol. It prohibits the marketing of ozone depleting substances depending on their ozone depletion potential (ODP) directly or with transition periods. The favourable working fluid R141b with an ODP of 0,11 will be completely forbidden in the year 2010.

Another well known convention is the Kyoto Protocol to the United Nations Framework Convention on Climate Change [6]. The Kyoto Protocol mentions hydrofluorocarbons (HFCs) and perfluorocarbons (PCFs) as greenhouse gases, but it gives no specific aim of reduction for these chemicals. The regulation 842/2006 of

European Parliament and Council [7] regulates the marketing and handling of fluorinated gases. The reduction of the emission of fluorinated gases will not be reached with a reduction to their utilization but with a reduction of leakage, obligation of recycling and education of operators. A replacement of HFCs / PFCs with substances with low GWP is only required for mobile air-conditioning (see Directive 2006/40 [8])

### **3. State of the art applications**

#### **3.1. Biomass Combustion**

The use of the ORC process for CHP production from Biomass combustion has been discussed a lot the last decade [9], [10].

Nowadays a lot of biomass combustion plants that work with the ORC process are installed and produce heat and/or power (e.g. Stadtwärme Lienz Austria 1000 kW<sub>el</sub>, Sauerlach Bavaria 700 kW<sub>el</sub>, Toblach South Tyrol 1100 kW<sub>el</sub>, Fußach Austria 1500 kW<sub>el</sub>) [11] [12].

The main reason why the construction of new ORC plants increases is the fact that it is the only proven technology for decentralized applications for the production of power up to 1 MW<sub>el</sub> from solid fuels like biomass. The electrical efficiency of the ORC process lies between 6-17% [13]. This efficiency is linked with the maximum heat recovery and the thermal efficiency of the boiler.

In the case of biomass combustion, thermal oil transfers the heat from the combustion chamber into the organic medium. Therefore, when the efficiency of the system is calculated, also the boiler efficiency of the thermal oil boiler has to be taken into consideration.

However, even if the efficiency of the ORC is low, it has some advantages, like the fact that the system can work without maintenance, which leads to very low personnel costs. Furthermore the organic working fluid has, in comparison with water, a relatively low enthalpy difference between high pressure and expanded vapour. This leads to higher mass flows compared with water. The application of larger turbines due to the higher mass flow reduces the gap losses compared to a water-steam turbine with the same power. The efficiency of an Organic Rankine Cycle turbine is up to 85 % and it has an outstanding part load behavior [14].

For numerous organic fluids the expansion of the turbine ends in the region of superheated vapour. This avoids drop erosion and allows a reliable operation and a fast startup of the cycle.

### **3.2. Geothermal plants using ORC**

Another case where the ORC technology is applied is its combination with the heat from geothermal heat sources. Conventional power station technology is not suitable for heat sources with temperatures between 80 °C and 120 °C. According to the literature, there is an optimum evaporation temperature of the working fluid, which depends on the quantity of the recycled geothermal water, at which the capacity of the Clausius–Rankine is the highest [15]. The Kalina process [16] which is a process using a mixture of ammonia and water seems to be the only alternative to ORC. An example of a geothermal plant using the ORC process is the plant Neustadt-Glewe in Germany [17], [18], which was the first geothermal power plant in Germany.

The facility uses hot water of approximately 98°C located at a depth of 2.250 m and converts this heat to 210 kW electricity by means of an ORC turbine. The ORC heat generation system is integrated into the thermal water cycle directly behind the

extraction bore hole. Following the electricity generation, the water, which is still at 70°C – 84°C, is transferred to the district heating network. Depending on the heat demand the cooled-off water has to be mixed with hot water from the bore hole. This means, that electricity production and the heating system compete for the thermal water.

#### **4. Innovative applications**

Due to the great advantage of the ORC process to use the waste heat from a process, many innovative concepts coupling the ORC process are being developed and optimized nowadays. For the modeling of these concepts, the process simulation environment IPSEpro was used. IPSEpro is a high flexible software tool used for heat balance analysis of power plants, component design, acceptance test calculations and on-line optimization [19].

##### **4.1. Solar Desalination with an Organic Rankine Cycle**

The rising of the world population leads automatically to the rising of water demand. As a consequence the lack of drinking water increases. Since a large part of the world population is concentrated in coastal areas, the desalination of seawater seems to be a promising solution.

The most common way to desalinate water is by means of processes with phase change like Multi-Stage-Flash-Evaporation (MFS) [20], Multi-Effect-Distillation (MED) [21] and others.

An efficient process without phase change is the Reverse Osmosis (RO), which energy consumption per m<sup>3</sup> of desalinated water is 5-6 times lower than thermal technologies [22].

A stand-alone solar desalination system driven by an Organic Rankine Cycle can be used to produce drinking water from seawater. The advantage of such a system is that it combines an efficient desalination technology (Reverse Osmosis) with a renewable energy source (solar radiation). The presented system has been developed in the European research project COOP-CT-2003-507997 called RO-Solar-Rankine [23].

The setup of the system is shown in Fig. 3.

A field of evacuated tube collectors (1) heats the collector fluid, circulated by a circulation pump (2). The collector fluid preheats (3), evaporates (4) and superheats (5) the organic working fluid. The superheated vapour is expanded in a scroll expander (6) generating mechanical work for the cooling water (7) and the seawater feed pump (8). The expanded vapour is condensed in a condenser (9), and cooled with seawater (10). After this, the feed pump (11) is pumping back the working fluid.

The seawater feed pump (8) raises the pressure of the seawater (12) which will be desalinated. In the reverse-osmosis-membrane (13) where the separation from clean water (14) and briny water (15) takes place. The clean water leaves the membrane and is drained to a storage tank. An energy recovery device (16) re-extracts the hydrostatic energy of the fluid, after this the brine is drained back to the sea.

The biggest advantage of the system is that despite the use of solar thermal energy, the product water has approximately the temperature of the seawater. This is advantageous for the direct use of the water as drinking water as well as for the storage of the water. Some photos of the above described project are shown in Fig. 4.

In the left photo [24], the ORC part of the plant is shown when on the right photo the installation of the solar collectors can be seen.

One of the key components of the system is the expansion machine. Turbines are available mostly in a power range above  $50 \text{ kW}_{\text{mech}}$ . One aim of the project was to find a market available expansion machine in a range of performance smaller than  $10 \text{ kW}_{\text{mech}}$ . A very promising machine is the scroll expander, which is a modification of a compressor, commonly used for air conditioning technologies. Scroll expanders are positive displacement machines having favourable part load behaviour. As expansion machine for the project, a Sanden scroll compressor from the TRS-series was modified [25]. The applied machine is a TRS105 with a capacity of 115 ccm.

A comparison of different possible working fluids, also derived from air conditioning technologies, has been done. The efficiencies for different working fluids can be seen in Fig. 5. The calculation is performed for a isentropic efficiency of 75% for the expansion machine and 85% for the feed pump. The volume ratio of the expansion machine is 3, a typical value for a scroll expander. Positive displacement machines like the scroll expander have a built-in volume ratio. Depending on the working fluid and its properties at given temperature and pressure, the pressure ratio can be determined. The pressure ratio of live vapour to condenser pressure should be the same as the pressure ratio of the working machine in order to avoid over- or under expansion. If the superheating temperature difference is constant, higher live vapour temperatures lead to higher boiling as well as condensation pressures for optimum efficiency of the expansion. Lower live vapour pressure leads to lower condensation pressure, it has to be checked if the corresponding condenser temperatures can be obtained. Due to this fact, the cycle efficiency has a maximum and declines afterwards with rising live vapour temperature. The working fluid R134fa has its

optimum at moderate live steam temperatures, the optimum will not be reached due to too low condensation temperature of  $< 10$  °C. With condensation temperatures of 30 °C, live vapour temperatures of around 75 °C are reached.

The vapour saturation line of R134a has a positive inclination and as a consequence, the expansion ends near the saturation line, as it can be seen in Fig. 6. A recuperator has in this case a minor effect on the efficiency. Due to the pressure losses; it could even lead to lower efficiencies. Because of this fact, no recuperator is used in this system.

A dynamic simulation of the system was performed in order to quote the mean daily water production of the system taking into consideration the fluctuation of solar irradiation.

As described above, for the simulation the program IPSEpro was used [19]. This cycle calculation program contains the Model Development Kit (MDK), which allows changing the component model equations and gives the possibility to the user to create his own models. The modelling has been performed as presented in Fig. 3. The main components of the process have been simulated as follows:

a) Collector

Evacuated tube collectors with direct flow have been used. The heat flow of the collector is calculated using equation (1). In this equation  $\Delta T$  is the difference of the collector's mean temperature and the ambient temperature.

$$\dot{q}_{coll} = \eta_0 \cdot (K_{gb} \cdot G_b + K_{gd} \cdot G_d) - k_1 \cdot \Delta T - k_2 \cdot \Delta T^2 - C_{eff} \cdot \frac{dT_{coll}}{dt} \quad (1)$$

$\eta_0$  corresponds to the zero loss collector efficiency,  $G_b$  is the beam irradiance in the plane of collector (in W/m<sup>2</sup>) and  $G_d$  the diffuse irradiance in the plane of collector (in W/m<sup>2</sup>).  $k_1$  is the collector's loss efficiency in [W/m<sup>2</sup>K] and  $k_2$  the

collector's loss efficiency in  $[\text{W}/\text{m}^2\text{K}^2]$ . Finally,  $C_{eff}$  is the effective heat capacity of the collector (in  $\text{kJ}/\text{m}^2\text{K}$ ).

Due to the fact, that tilted irradiation causes lower absorption, the Incidence Angle Modifiers (IAM)  $K$  have to be considered. For optical asymmetric collectors like evacuated tube collectors, it is a product from the IAM for the longitudinal (L) and transversal (T) incident angle (beam irradiation (b)).

$$K_b(\theta) = \frac{\eta(\theta)}{\eta_0} = K_b(\theta_L) \cdot K_b(\theta_T) \quad (2)$$

For diffuse irradiation which has no incidence angle the IAM has a constant value  $K_d(\theta) = const$  [26].

The experimentally determined parameters of equation (1) are all related to the aperture area of the collector. Therefore overall heat flow is calculated as product from specific heat flow and aperture area. The aperture area ( $A_{Aperture}$ ) is defined as the maximum projected area through which unconcentrated solar radiation enters a collector [27].

$$\dot{Q}_{coll} = \dot{q}_{coll} \cdot A_{aperture} \quad (3)$$

The collector's temperature is possible to be calculated for a specific time step  $\Delta t$ , using equations (1) to (3):

$$T_{coll,t} = \frac{\dot{Q}_{coll} \cdot \Delta t}{C_{eff} \cdot A_{aperture}} + T_{coll,t-\Delta t} \quad (4)$$

For the operation of the collector after the warm up matched flow and if steady collector temperatures are assumed. The mass flow can be calculated using equation (5).

$$\dot{m}_{coll} = \frac{\dot{Q}_{coll}}{c_p \cdot (T_{coll,in} - T_{coll,out})} \quad (5)$$

b) Climate data

For the calculation of the monthly mean daily water production rates for different sites, the climate data of the site taken from the meteorological database Meteonorm [28] was used. This state of the art global database includes over 7.400 weather stations and provides long term yearly and monthly mean values.

c) Scroll expander

As presented in equation (6), the scroll expander is modelled as expansion machine including part load behaviour. Therefore the isentropic efficiency ( $\eta_s$ ) is calculated as a product of the isentropic efficiency in the design point ( $\eta_{s0}$ ) and a mass depending function which indicates the efficiency's dependence on the live vapour mass flow.

$$P_{mech} = \eta_s \cdot \dot{m} \cdot (h_5 - h_4) \quad (6)$$

$$\eta_s = \eta_{s0} \cdot f\left(\frac{\dot{m}}{\dot{m}_0}\right) \quad (7)$$

d) Osmosis process

Osmosis is the process of diffusion of a solvent through a semi permeable membrane whereas the solvent diffuses to the side with the lower concentration of solute. The diffusion of solvent continues till the osmotic pressure on both sides of the membrane is assimilated. Due to the water flow to the side with the formerly higher concentrated solution the liquid level and therefore the hydrostatic pressure (equal to the osmotic

pressure) is higher. If pressure higher than the osmotic pressure difference on the side with the saline solution is applied, pure water is forced through the membrane

The osmotic pressure of a solution is calculated with equation (8).

$$\Delta p_{osm} = \frac{1}{M_s \cdot v} \cdot (w_{rich} - w_{poor}) \cdot R \cdot T \cdot 10^{-2} \quad (8)$$

The model equations of a reverse osmosis membrane are as follows:

$$\dot{m}_p \cdot (1 - w_p) = \frac{A \cdot A^* \cdot (\Delta p_{mem} - \Delta p_{osm})}{v_p} \quad (9)$$

The permeate mass flow is calculated using equation (9). Although the membrane is ideally permeable only for pure water, in reality small amounts of salt are passing through the membrane. This dependency is described with equation (10).

$$\dot{m}_p \cdot w_p = B^* \cdot A \cdot (w_{mem} - w_p) \quad (10)$$

At the surface of the membrane there is a higher concentration of solute because the pure water is forced through the membrane and the solutes are hold back. The solutes diffuse away against the direction of feed water flow. The concentration at the membrane surface adjusting can be calculated using the following equation [29].

$$w_{mem} = w_p + \left( \frac{w_F + w_R}{2} - w_p \right) \cdot e^{\left( \frac{\dot{m}_p \cdot v_p}{k_s \cdot A} \right)} \quad (11)$$

IPSEpro provides only steady state solutions for the whole process. In order to obtain the dynamic system behaviour IPSEpro is linked with an external algorithm. Thus the transient processes, like the warm-up of the collector field and the alteration of climate data are managed with Microsoft® Excel. Excel provides the input values for the IPSEpro calculation (e.g. collector temperature). After solving the system the

values of interest are requested by a visual basic macro and saved in an Excel Sheet for later analysis.

Fig. 7 shows the influence of the inclination of the collector field on the mean daily water production. As it can be seen, the water production can be adjusted to the specific needs of the consumers. This allows the design of the desalination plant for specific requirements like high water production in summer time for hotels or more constant water production over the whole year for industries.

#### **4.2. Waste heat recovery from biogas digestion plants**

Energy from biomass and in particular the digestion of biomass, organic wastes or energy crops in biogas plants is one of the most attractive options in order to reach Europe's targets for the implementation of renewable energies. It offers a very big potential to substitute fossil fuels. Furthermore, with this technology it is possible to produce electrical base load energy. However, in most cases, this technology is presently only economically feasible with subsidies. Hindering for the efficiency of biomass digestion plants is the huge amount of waste heat that could commonly not be used for district heating systems due to the characteristics of heat demand and the location of the digestion plants.

One possibility to overcome this restriction is to use the waste heat for further electricity production by the application of an Organic Rankine Cycle.

Fig. 8 shows the schematic setup of such a system. The Organic Rankine Cycle, which is similar to the ORC presented in the Introduction chapter, is heated with the excess heat of the combustion engine.

The biomass digester has a specific thermal requirement because the digestion takes place at around 38 °C. The thermal power is needed for heating the digester and for heating up the substrate to the digester temperature. Depending on the size of the digestion plant and the standard of the insulation of the plant, the thermal need is between 20 ... 25 % of the waste heat of the motor [30]. According to the low temperature level, the digester can be heated with the cooling water of the motor and the turbocharger. For driving the ORC the heat of the exhaust gas can be used.

In the present work, the simulation of the performance of a biomass digestion plant with an electric power of 537 kW<sub>el</sub> is presented.

The assembly of the system is shown in Fig. 9. The combustion engine (1) is fed with biogas (2) from the digester. The air (3) is compressed with a turbocharger. The combustion engine with a power of around 555 kW<sub>mech</sub> drives the generator producing electrical power. The heat from cooling the compressed air (4) and the motor cooling water (5) can be used for heating the digester. The exhaust gas (7), which leaves the motor at around 490 °C, is used in a heat-exchanger (8) to heat up the thermo oil (9), which following preheats (10), evaporates (11) and superheats (12) the organic working fluid. In reality these three steps would take place in one single heat exchanger. The superheated working fluid is then expanded in an expansion machine, which could be a turbine or a screw type expander, which is coupled with a further generator. Due to the fact, that the used working fluid is after the expansion still in the area of superheated vapour, it is used in the regenerator (18) in order to preheat the liquid working fluid. After being desuperheated, the vapour is condensed in a condenser (15) which is cooled back with air or water from a cooling tower (16). The feed pump (17) rises the pressure of the working fluid and forces the fluid again through the heat exchangers.

As working fluid the hydrofluorocarbon R245fa was chosen. The fluid has a very broad application range. It is used as foaming agent, refrigerant and filling for thermosiphons as well as working fluid for Organic Rankine Cycle for heat recovery and bottoming cycles [31]. Due to the negative inclination of the vapour saturation line (see Fig. 10) between evaporation (2-3) and condensation (6-0), the sensible heat that rests in the expanded vapour (5) can be used for preheating the liquid working fluid.

With the current setup, an additional power production from around 35 kW<sub>el</sub> is possible. This leads to a significant gain of electric efficiency of 2.6 basis points. The use of the motor cooling water for preheating the working fluid does not lead to additional power production, because the higher temperature of the working fluid leads to higher return temperatures of the thermal oil. Thus, the temperature difference and the transferred heat in the exhaust gas heat exchanger decline.

The efficiency of a waste heat recovery utility is strongly dependent on the interest rate and the full load hours of the system. According to these two degrees of freedom the electricity production cost is shown in Fig. 11. The diagram is based on a cost effectiveness study including maintenance, insurance and operation cost.

More specifically, Table 3 presents the cost assumptions of the efficiency calculation.

The electricity production cost (epc) is equal to the yearly produced electricity (expressed as the product of the net power and the full load hours), divided with the sum of the annuity of the investment and the overheads linked with the maintenance and the insurances of the plant:

$$epc = \frac{P_{el} \cdot h_{fullload}}{A_{N,K} + \frac{f_K}{100} \cdot A_0} \quad (12)$$

The annuity of the investment is calculated as product of the investment and the annuity factor, which is determined by the interest rate and the economic life time.

$$A_{N,K} = A_0 \cdot a = A_0 \cdot \frac{(1+i)^t \cdot i}{(1+i)^t - 1} \quad (13)$$

The calculations have been performed using the German guideline for economic calculation VDI 2067 [32].

In Fig. 11, it can be clearly seen, that a reliable and long operation is necessary for an economic operation of such a waste-heat recovery unit. The point in the figure indicates an operation range with 7500 hours of full load operation and an interest rate of 5 %. With this basic conditions, electricity production cost of around 5.65 €-Ct can be achieved. This is significantly less than the actual refunding of electricity from biomass, which is 13.16 €-Ct/kWh.

#### 4.3. Micro-CHP

A further application for ORC is the combined heat and power production in a power range from a few kW<sub>el</sub>, often called Micro-CHP. In this application range the system will be driven in a way that the heat from the condenser will be completely used for heating purposes.

Fig. 12 shows the setup of such a system. A biomass fired boiler (1) heats a thermal oil loop (2). The thermal oil then preheats (3), evaporates (4) and superheats (5) the organic working fluid. As expansion machine a scroll type expander (6) can be used.

The electricity produced (7), can be used in the housing complex or can be distributed to the grid. Important is, that the excess heat from the condenser (8) is completely used for heating purposes in the house or in a small district heating network (9).

This implies that the condensation temperature of the organic working fluid is in a temperature range that is suitable for the use of the condenser's waste heat for space or water heating. This would be around 90 °C. The working fluid has to have a critical temperature at a sufficient level. One fluid that fits well for this application is R141b but it will be forbidden in 2010. Thermodynamic efficiencies of around 8% can be reached with live vapor temperatures of around 150°C. By using a scroll type expander with a pressure ratio  $\Pi=3$ , the condensation temperature will be around 91°C, which is sufficient for heating purposes.

Fig. 13 shows the dependence of the electricity production cost of the achievable heat revenues and the full load hours for a 5 kW<sub>el</sub>/50 kW<sub>th</sub>-system. It can be clearly seen, that it is absolutely necessary to use such a system for the production of heat base-load, which makes a second boiler necessary. Due to the high investment cost for small systems the electricity production cost is in the range of the cost the user has to pay for the supply of electricity from the grid. Heat revenues of 90 €/MWh and 5500 hours of full load operation would lead to electricity production cost of 10.7 c€/kWh.

Finally, it should be mentioned that a very interesting Micro-CHP application is the coupling of a micro-turbine with an ORC as presented by Invernizzi et al. [33]. In this case, with reference to a micro-gas turbine with a size of about 100 kW<sub>el</sub>, a combined configuration could increase the net electric power by about 1/3, yielding an increase of the electrical efficiency of up to 40%.

## 5. Conclusions

Nowadays, the Organic Rankine Cycle can be characterised as the only proved technology that is commonly used in ranges of a few kW up to 1 MW. Despite the fact that it is linked with low efficiencies, new applications of this technology are commonly discussed due to its possibility to utilise the low-level waste heat from other processes. Therefore it's coupling with waste heat recovery from biogas digestion plants or micro-CHP systems provide very promising solutions for low cost, decentralised applications. Additionally, the lack of fuel costs makes these applications even more attractive. Finally, the introduction of the ORC process in a Reverse-Osmosis desalination system using solar energy provides a solution for areas with drinking water problems.

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### List of Figures

Fig. 1 Main components of the ORC

Fig. 2 T-S diagram of water steam, R245fa and Cyclohexane

Fig. 3 Schematic representation of solar driven RO-System

Fig. 4 Photos of a solar driven RO-System

Fig. 5 Comparison of different working fluids

Fig. 6 T-s-Diagram R134a (0-1 compression in feed pump 1-2 preheating 2-3 evaporation 3-4 superheating 4-5 expansion 5-6 desuperheating 6-0 condensation)

Fig. 7 Influence of different inclination of collector field on mean daily water production

Fig. 8 Biomass digestion [13] including waste heat recovery with low-temperature ORC

Fig. 9 Schematic representation of waste heat recovery for combustion engines

Fig. 10 T-s-Diagram R245fa (0-1 compression in feed pump 1-2 preheating 2-3 evaporation 3-4 superheating 4-5 expansion 5-6 desuperheating 6-0 condensation)

Fig. 11 Influence of the interest rate and the full load hours on the electricity production cost

Fig. 12 Schematic Representation of Micro-CHP System

Fig. 13 Dependence of electricity production cost of heat revenues and full load hours

### **List of Tables**

Table 1: List of ORC-plants manufacturers

Table 2: List of working fluids

Table 3: Cost assumptions for efficiency calculations

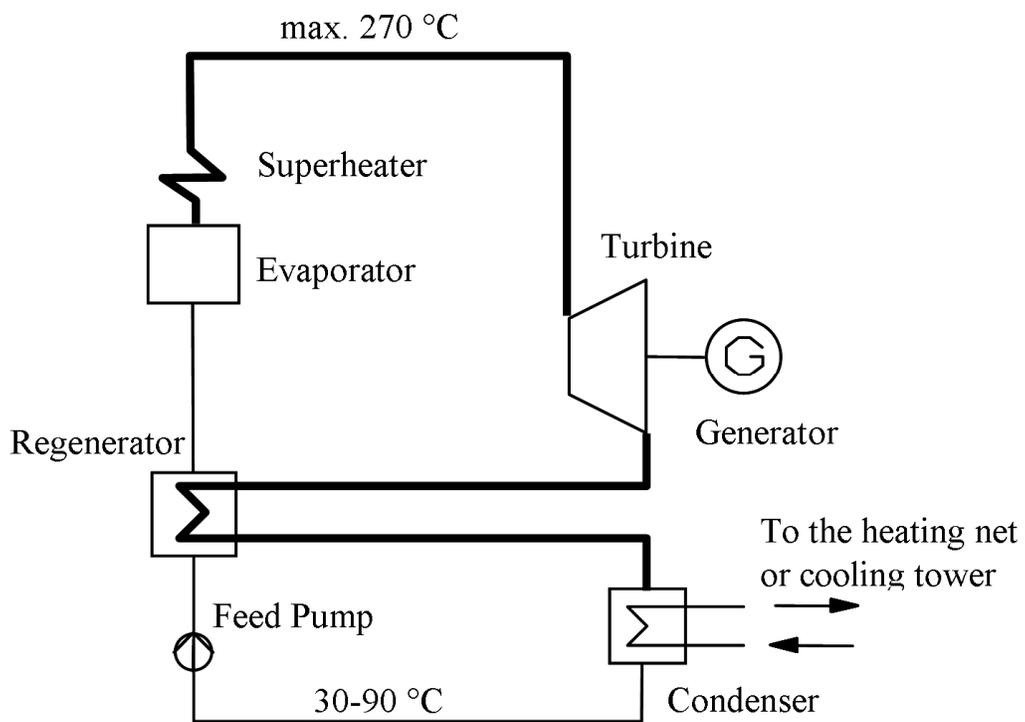


Fig.1

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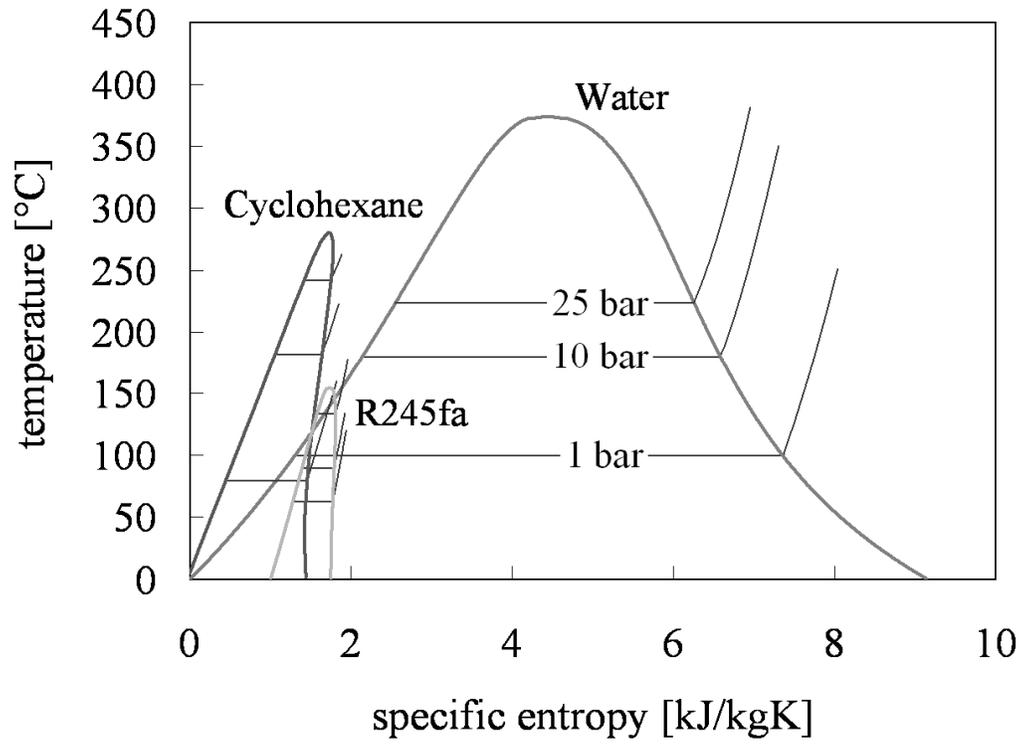


Fig.2

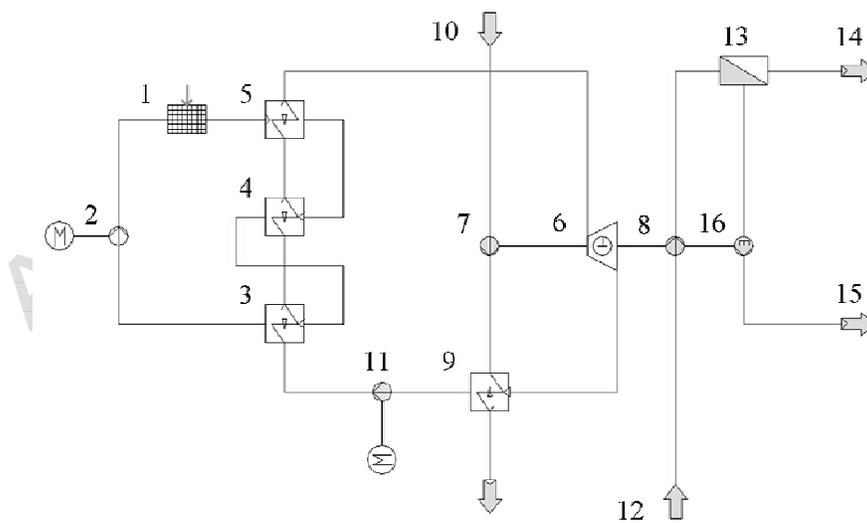


Fig.3



Fig.4

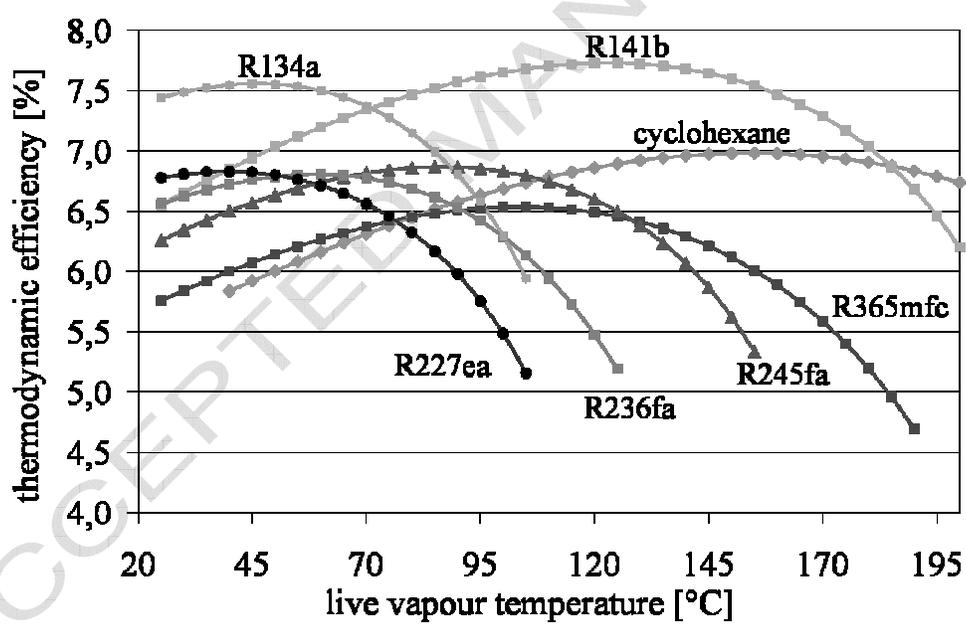


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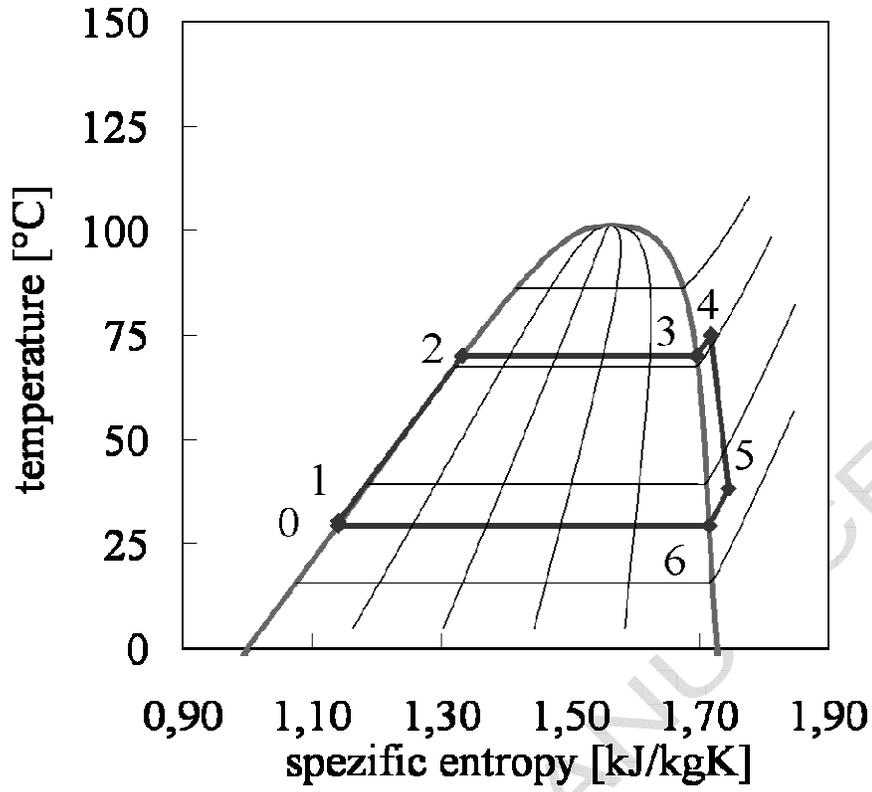


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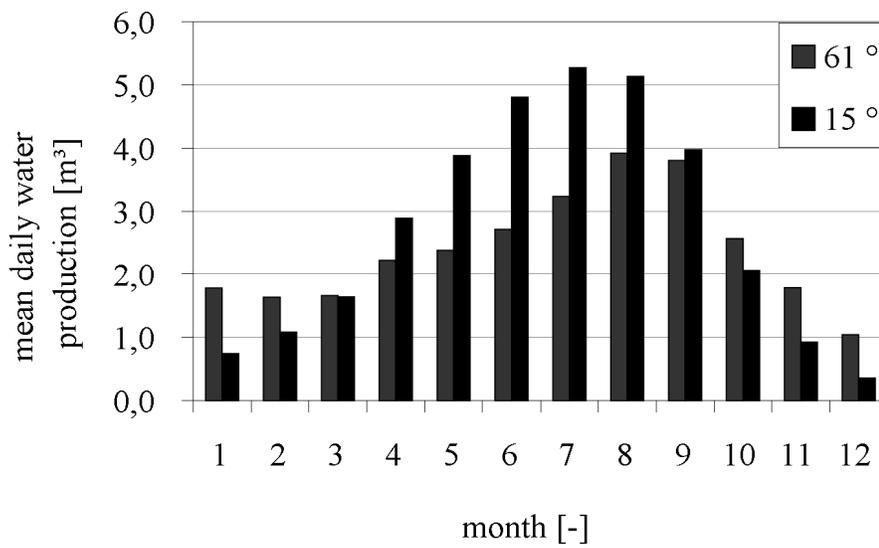


Fig.7

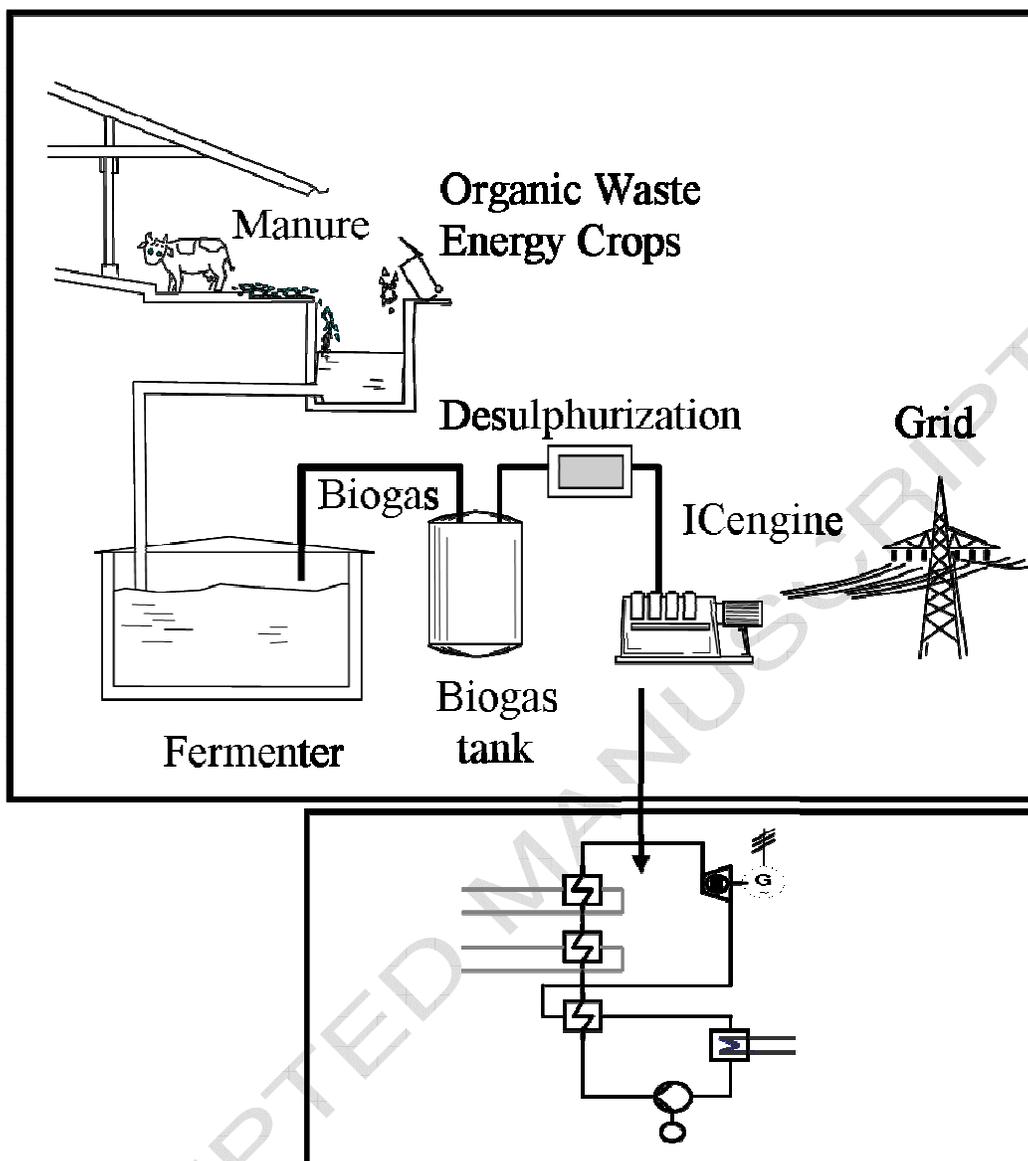


Fig.8

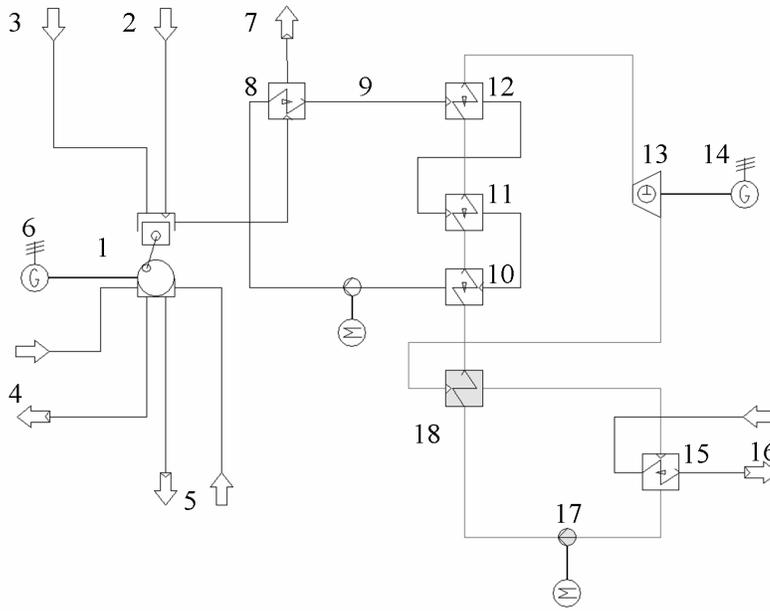


Fig.9

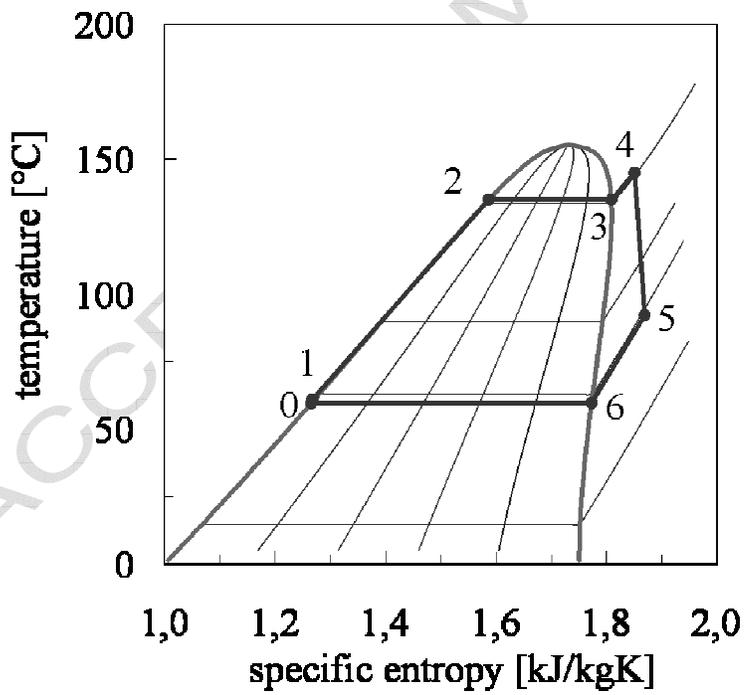


Fig.10

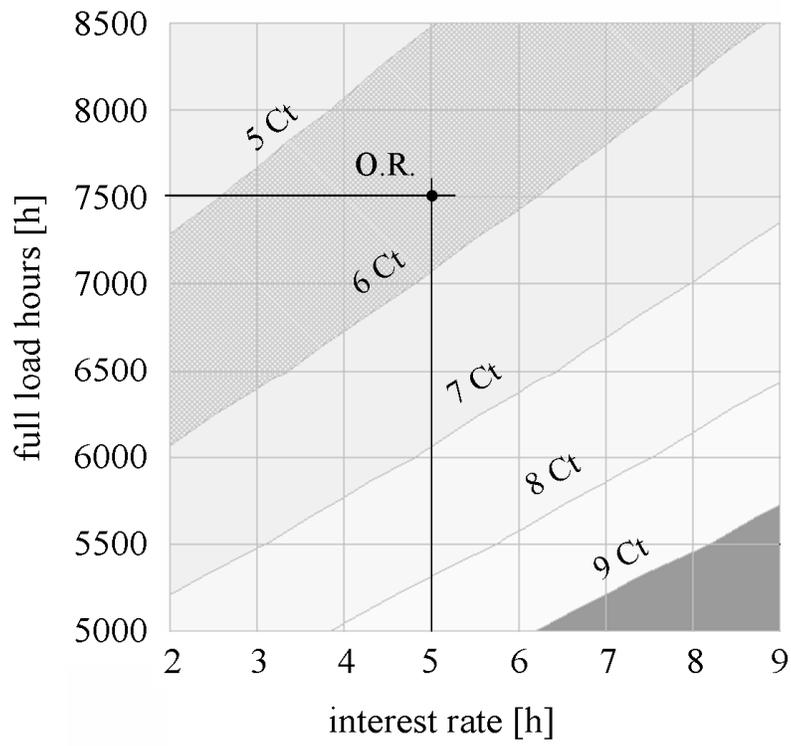


Fig.11

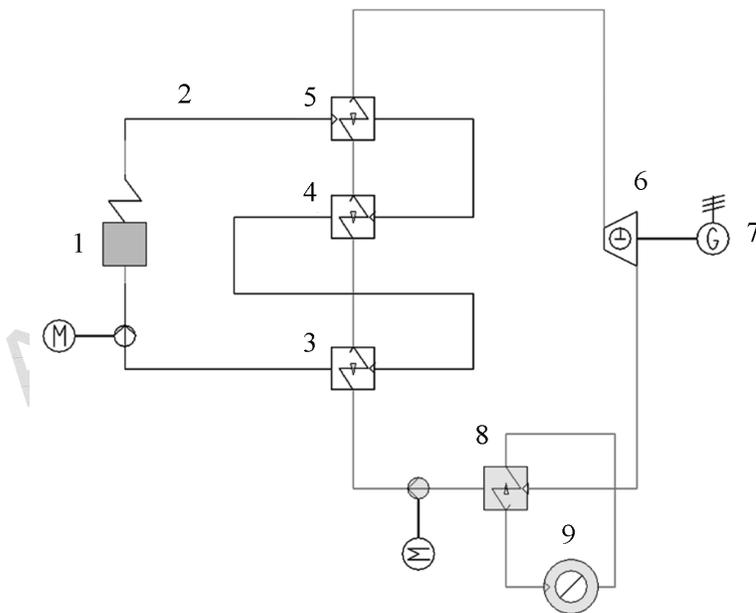


Fig.12

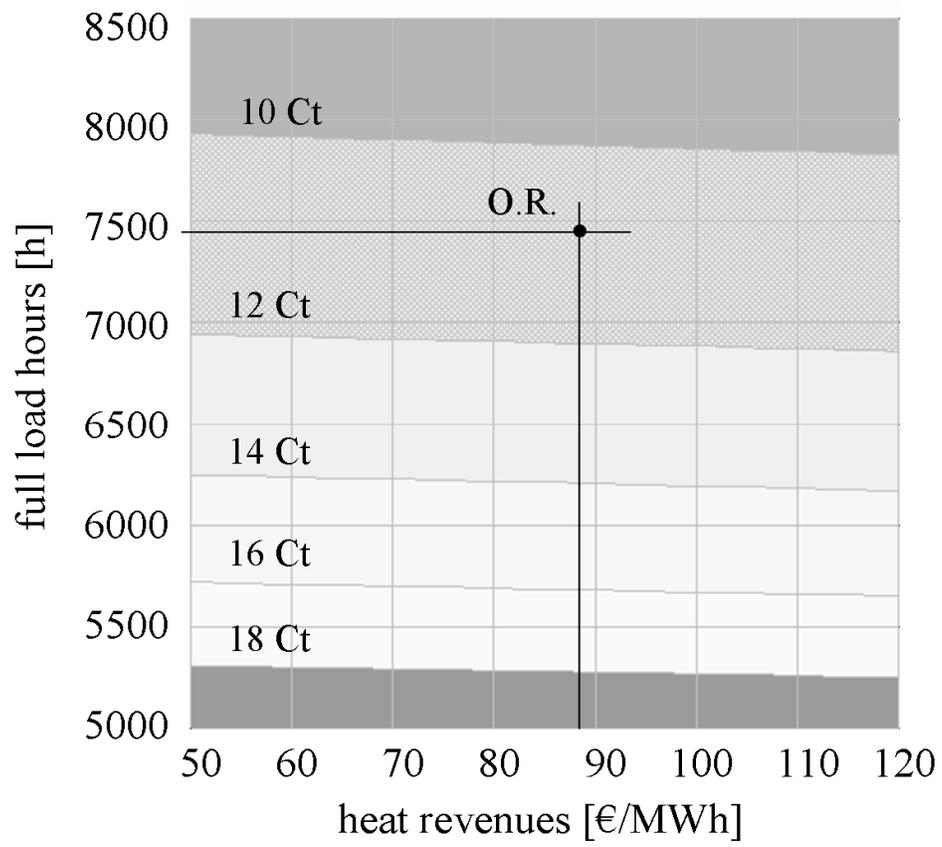


Fig.13

ACCEPTED

Manufacturer	Power Output	Electrical efficiency	Useful temperature level	Fluid	Technology	Web-site
ORMAT 980 Greg Street Sparks, Nevada 89431-6039 USA	200kW- 7,5MW	no data	no data	no data	no data	<a href="http://www.ormat.com">http://www.ormat.com</a>
Turboden s.r.l. Viale Stazione, 23 25122 Brescia-Italy	300kW - 2MW	16%-18%	about 280°C	Silicon oil	2-staged radial turbine	<a href="http://www.turboden.com">http://www.turboden.com</a>
GMK GmbH Reuterstraße 5 18211 Bargeshagen	50kW- 2MW	9%-21% depending on the heat source	>120°C when screw motor is used, >350°C for turbine	no data	up to 500kW screw motor, from 500kW turbine of KK&K	<a href="http://www.gmk.info">www.gmk.info</a>
Köhler und Ziegler Anlagentechnik GmbH Auweg 10c 35457 Lollar	70kW- 200kW	11%	100°C	Hydro- carbons	Screw motor	<a href="http://www.koehler-ziegler.de">www.koehler-ziegler.de</a>
Ergion GmbH Bruchhäuser Str. 19 68723 Schwetzingen	4kW- 300kW	13%-16%	120°C-300°C	water based with additives	Modified screw motor	<a href="http://www.ergion.de">www.ergion.de</a>
ADORATEC GmbH Rotterdammer Str. 20-30 68219 Mannheim	300kW- 1750kW	15%-18%	>100	no data	Turbine	<a href="http://www.adoratec.com">www.adoratec.com</a>
WSK Energie und Umwelttechnik GmbH Linxer Straße 26 77694 Kehl-Leutesheim	52kWel to 65kWel depending on cooling water temperatu re	Average of 16,3%	Cooling of exhaust gas from 490°C to 175°C	Fluoric Hydro- carbons	Screw motor	<a href="http://www.wsk-group.com">www.wsk-group.com</a>

Table 1: List of ORC-plants manufacturers

Table 2: List of working fluids

Working fluid	$T_c$ [°C]	$p_c$ [bar]	$T_{s, 1 \text{ bar}}$ [°C]	$p_{s, 20^\circ \text{C}}$ [bar]
R134a	101,1	40,6	-26,4	5,7
R227ea	101,8	29,3	-16,6	3,9
R236fa	124,9	32,0	-1,8	2,3
R245fa	154,0	36,5	14,8	1,2
R141b	204,4	42,1	31,7	0,7
R365mfc	186,9	32,7	39,8	0,5
Cyclohexane	280,5	40,8	80,3	0,1

Table 3: Cost assumptions for efficiency calculations

Additional net Power	35	$\text{kW}_{el}$
Specific Investment Cost	3755	$\text{€}/\text{kW}_{el}$
Investment Cost	131	k€
Economic Life Time	15	a
Maintenance cost	1969	€/a
	(1,5 % of investment cost)	
Insurance cost	197	€/a
	(0,15 % of investment cost)	