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Innovative CHP concept for ORC and its benefit compared to conventional concepts

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HIGHLIGHTS

- Two-stage ORC with turbine bleeding is investigated for combined heat and power.
- Two-stage concept shows high flexibility for representative isentropic fluids.
- Performance with siloxane is investigated for different ORC-CHP concepts.
- Annual produced electricity is used to compare revenues for different concepts.
- Two-stage concept produces more electricity for high cover ratios of heat demand.

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ABSTRACT

Combined heat and power is highly favorable in order to prevent harmful CO_2-emissions. Besides that, economic benefits go along due to higher full load operation hours. The present work investigates the flexibility and suitability of different Organic Rankine Cycles for combined heat and power concepts, since it can play a significant role to achieve climate goals.

An integrated concept for heat decoupling based on a two-stage Organic Rankine Cycle with regenerative preheating from turbine bleeding is introduced. The heat extraction to the district heating system is directly in line with the receiver tank for the preheating. The flexibility of this integrated concept is determined for different isentropic fluids and siloxanes. Under general circumstances it is more favorable to apply a recuperator for dry fluids such as siloxanes, due to the high amount of sensible heat after the turbine outlet. It has been shown in this work, that the proposed regenerative preheating concept is more beneficial for combined heat and power. While other concepts are only applicable for base load heat demand or for peak load heat demand, this concept is suitable for the entire range of cover ratios of a district heating system. Thus, this concept offers highest flexibility in application and a good capacity utilisation.

1. Introduction

The Organic Rankine Cycle (ORC) is an established technique for waste heat recovery, as well as for the utilization of biomass, geothermal energy and solar thermal energy [1]. Several reviews on technological aspects and applications can be found in literature [1–3]. Its application enables the electricity generation from low temperature heat sources and is especially applied in the small to medium size range (few kW_e up to several MW_e).

Besides the working fluid selection, several technological approaches to increase either the system efficiency or the economic performance have been investigated in literature. Current research focusses on the following topics:

1. Optimized cycle configuration to increase efficiency (e.g. [4–9]).
2. Co-generation systems to increase flexibility (e.g. [10–16]).
3. Super- and trans-critical cycle operation to increase efficiency (e.g. [1,17–19,6]).
4. Working fluid mixtures to reduce exergy losses, GWP or costs (e.g. [20–23]).
5. Control strategies to increases operational stability and efficiency (e.g. [24–27]).

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Nomenclature

Roman symbols

$\dot{Q}$	heat flow rate, W
c	specific costs, €/MWh
P	power, W
p	pressure, bar
Q	heat, J
T	temperature, K
t	temperature, °C
X	cover ratio
z	hours per year, h/a

Greek symbols

$\bar{\beta}$	average load factor
β_{min}	minimal load factor
η	efficiency
Π	pressure ratio
σ	CHP-coefficient
τ	dimensionless time

Subscripts

a	annual
c	cooling
cond	condensation
crit	critical
e	electric
flh	full load operational hours
gen	generator
HP	high pressure
is	isentropic
LP	low pressure
m	mechanical
max	maximum
min	minimum
p	pump
pp	pinch point
r	return temperature
s	supply temperature
t	thermal
t	turbine
u	utilisation

Abbreviations

CCGT	Combined Cycle Gas Turbine
CHP	Combined Heat and Power
DHS	district heating system
HFC	Hydrofluorocarbons
HFO	Hydrofluoroolefins
HS	heat source
HTM	heat transfer medium
MDM	Octamethyltrisiloxane
MM	Hexamethyldisiloxane
OMTS	Octamethylcyclotetrasiloxane
ORC	Organic Rankine Cycle

Concerning cycle concepts, the majority of studies focus on simple ORC processes (e.g. [28,22,18,17,6]) or concepts with internal heat recovery (recuperator) [23,20,6,1]. Almost all of the organic working fluids have an isentropic (vertical slope of the saturated vapor curve) or dry expansion behavior (positive slope of the saturated vapor curve). These characteristic thermophysical property leads to superheated vapor after the expansion, which enables heat recovery in a so-called recuperator. Superheating of the live vapor should be avoided [14], because it does not increase the thermodynamic efficiency [4]. Besides this, recently two-stage ORC with turbine bleeding and regenerative preheating are focused especially for isentropic fluids. For pure power generation the relevant literature is summarized in the next paragraph.

Mago et al. [4] compared a two-stage concept with a simple ORC process for several working fluids (R113, R245fa, R123 and isobutane). They concluded, that the two-stage concept is more efficient in terms of thermodynamic and exergetic efficiency and that this advantage increases with increasing system pressure. Similar results have been obtained by Yari [29], who compared recuperation against two-stage regenerative preheating for geothermal applications with n-pentane, R113 and R123 as working fluids. The author concluded, that a combination of regenerative preheating and recuperator can reach even higher efficiencies. However, geothermal applications do not necessarily favor such efficiency measures, since the system efficiency remains constant. These efficiency measures go along with a higher re-injection temperature of the geothermal brine. Also Branchini et al. [5] compared different cycle designs and several working fluids for high and low heat source temperatures based on a performance index. Their results indicate, that recuperation performs better for MDM than two-stage regenerative preheating for pure power generation. Also Meinel et al. [7] concluded, that two-stage regenerative preheating is not favorable for working fluids with dry expansion behavior. Results showed, that regenerative preheating is more beneficial than recuperation for wet and isentropic fluids and offers residual waste heat at significant higher temperature levels [7]. This residual waste heat can be used for further purposes, such as CHP. Economic competitiveness of such two-stage concepts has been also demonstrated by Meinel et al. [30].

For CHP applications three types of heat utilisation purposes have to be distinguished [31]:

1. Heat for district heating
2. Heat for commercial buildings
3. Heat for industry (process heat)

Note, that the necessary temperature levels are specific for each application. Heat for residential applications can be provided at fairly low temperatures 30–60 °C [31]. While the lower limit accounts for panel heating (e.g. floor heating), the upper bound is a necessary requirement for hot tap water. Due to heat transfer pinch points the temperature levels of a corresponding district heating system (DHS) needs to be at least 70 °C. For industry heating and drying purposes the necessary temperature levels easily exceed 100 °C and can be as high as 550 °C [32].

The combined production of heat and power (CHP) with ORC is a highly favorable option in order to prevent harmful CO_2-emissions. The production of heat and power can enhance economic performance by higher full load operation hours [33]. The share of CHP in the German power sector accounts currently for about 12% and is expected to increase up to 25% in 2020 [34]. ORC-CHP systems feature great potentials in primary energy savings and environmental conservation and thus, it is an important pillar of the energy sector. Scientific literature on ORC-CHP focuses mainly on low temperature applications, such as geothermal or solar energy and small scale applications for the residential sector. Quoilin et al. [2] show in their survey, that biomass CHP can be considered as state-of-the-art and highlight the CHP potential for the residential sector. Mago et al. [35] investigate an ORC-CHP as a bottoming cycle for an internal combustion engine in the commercial building sector. Freeman et al. [36] consider ORC-CHP applications for the residential sector with a solar heat source in their work. Pernecker and Uhlig [37] report their experiences from a geothermal ORC-CHP project and give some economic figures on heat generation costs. Khennich et al. [38] compare two different ORC-CHP concepts for low temperature geothermal heat sources and R134a as the working fluid. Peris et al. [39] test a small scale commercial ORC-CHP for low grade heat sources on a laboratory test bench and Gewald et al. [40] introduce a commercial small scale ORC-CHP system in the 20 kW_e range. Lecompte et al. [14] optimized the performance of an ORC-CHP concept with a thermo-economic approach and for different working fluids (R152a, R1234yf and R245fa). The authors concluded, that R245fa offers highest efficiencies, but R152a is most economic.

The afore mentioned two-stage preheating concept has not been investigated for CHP-applications so far. Concepts for the integration of heat decoupling to a district heating system have not been developed. Based on this, it is the aim of this publication to extend the state-of-the-art in ORC-CHP concepts by introducing an ORC-CHP concept based on a two-stage ORC with regenerative preheating. Moreover, the concept is investigated for hydrofluoroolefins and compared to similar working fluids.

After an evaluation and comparison of state-of-the-art ORC-CHP concepts (Section 2), the simulation approach is introduced in Section 3. The results are presented in Section 4, where a separate discussion of isentropic (Section 4.1) and dry fluids (Siloxanes, Section 4.2) is carried out in terms of P_e-$\dot{Q}_{DHS}$-diagrams. For OMTS as a representative of these dry fluids a detailed comparison with other concepts is performed and an energetic comparison for district heating systems is presented based on annual load duration curves (Section 4.3).

This paper addresses to scientists, researchers and ORC manufacturers to develop new and more efficient concepts to end up in commercial products and to improve energy utilisation.

2. Combined Heat and Power with Organic Rankine Cycle (ORC-CHP)

After an introduction to ORC-technology this section provides an overview on state-of-the-art CHP concepts, the flexibility of these concepts is assessed in terms of CHP coefficient, applicable heat source and temperature level. The section concludes with an overview on different operational strategies for CHP plants.

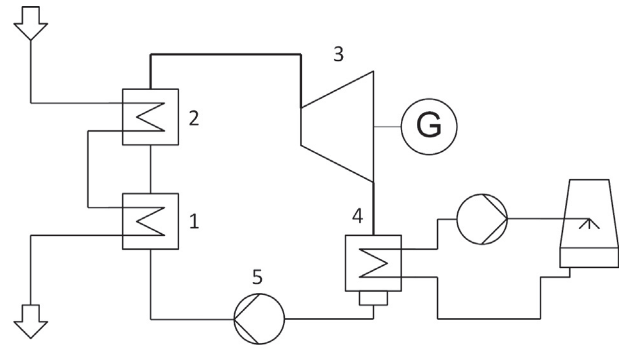
2.1. Organic Rankine Cycle

The standard or simple ORC concept as shown in Fig. 1(a) is described more thoroughly in literature (e.g. in [41,42,2]). This concept contains all the basic components for a thermodynamic cycle: Preheater (1), evaporator (2), turbine (3), condenser (4) and pump (5). No additional measures to increase the efficiency are applied.

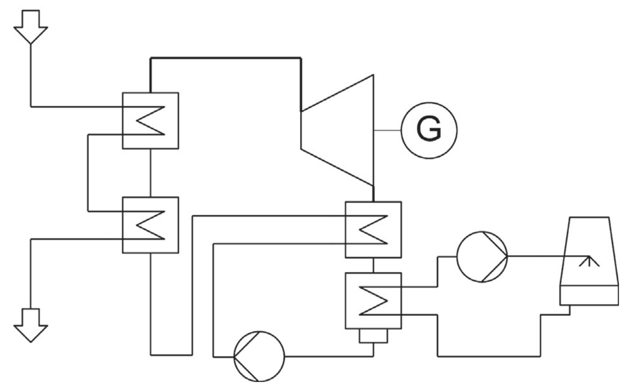
For high temperature applications (e.g. biomass) siloxanes can be considered as state-of-the-art fluids. Due to their dry expansion behavior, the integration of a recuperator is favorable (Fig. 1(b)). The sensible heat after the turbine outlet is recuperated and used to preheat the working fluid on the high pressure side before entering the preheater and evaporator. Further details can be found in literature (e.g. [1,2,43]).

2.2. Assessment criteria for CHP concepts

The efficiency of electricity production η_e is expressed as



(a) Standard ORC Cycle (for isentropic fluids)



(b) Recuperator ORC Cycle (for dry fluids)

Fig. 1. State-of-the-art ORC concepts.

$$\eta_e = \frac{P_e}{\dot{Q}_{HS}}, \tag{1}$$

with the generated electrical power P_e and the heat flow from the heat source $\dot{Q}_{HS}$. Instead of the heat input, the available heat source is taken into account, therefore the electric efficiency is equivalent to a system efficiency. The combined production of heat and power is commonly described by a utilisation efficiency η_u in dependence of heat input [44,31]. This utilisation efficiency can be considered also as a CHP system efficiency and can be expressed by the equation

$$\eta_u = \frac{P_e + \dot{Q}_{DHS}}{\dot{Q}_{HS}}, \tag{2}$$

with $\dot{Q}_{DHS}$ being the heat flow extracted to the district heating network. In order to evaluate the flexibility of CHP concepts, another parameter is necessary. The CHP coefficient σ is therefore introduced, which is defined as the ratio between generated electrical power P_e and the heat flow extracted to the district heating network $\dot{Q}_{DHS}$

$$\sigma = \frac{P_e}{\dot{Q}_{DHS}}. \tag{3}$$

This CHP coefficient can be either constant or variable depending on the individual concept.

2.3. State-of-the-art CHP concepts

Several configurations for CHP with an Organic Rankine Cycle can be found in literature [2,44,35,14,36]. Commercial products for these ORC-CHP concepts are also available in different size ranges (e.g. [45,40]). These concepts are shown in Fig. 2, where

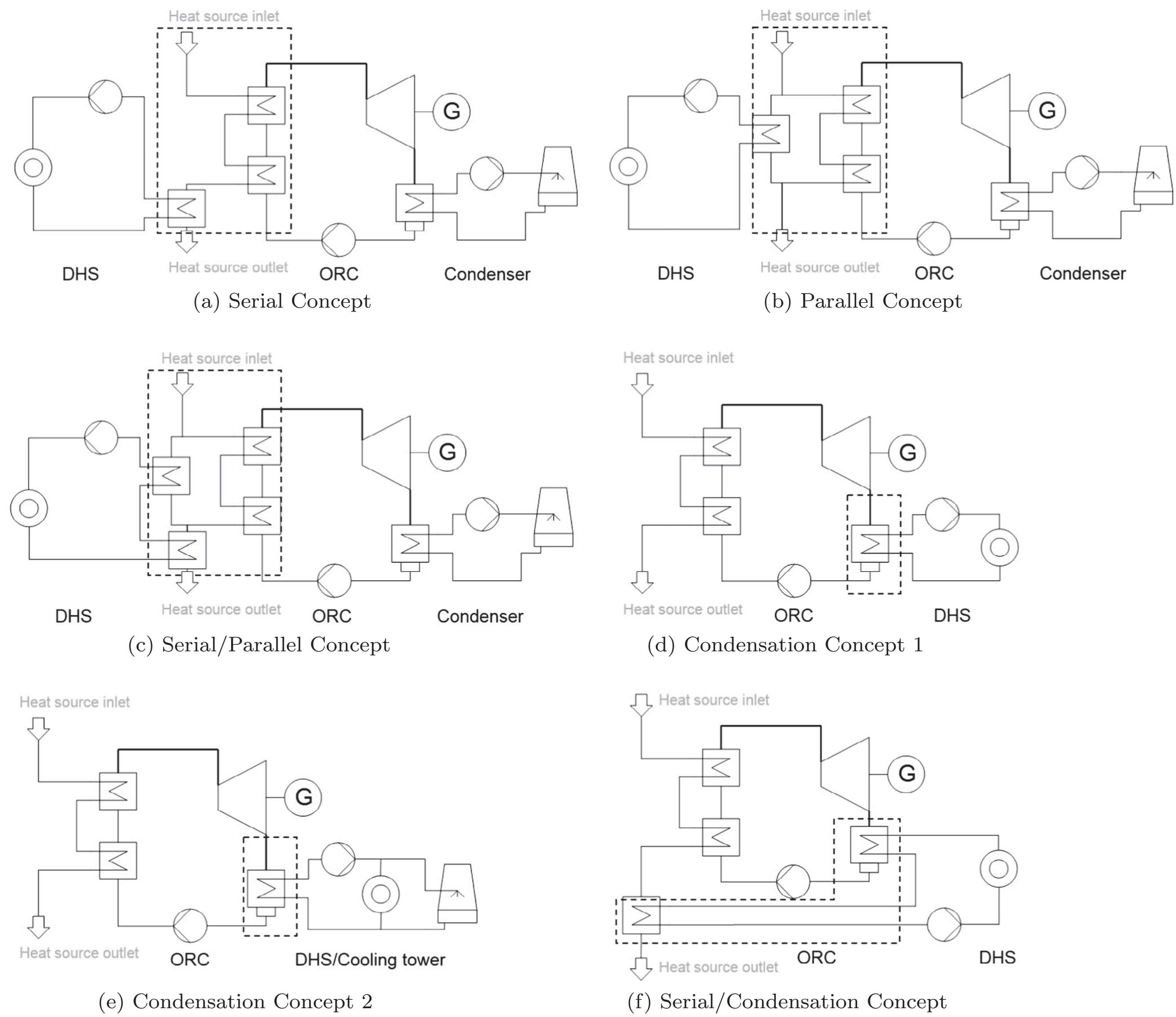


Fig. 2. Different state-of-the-art CHP concepts.

the heat exchanger arrangement and the extraction to the district heating system (DHS) is highlighted with the dashed line. They are described in more detail in the next paragraphs.

Serial Concept. This concept is investigated e.g. by Khennich et al. [38] and is shown in Fig. 2(a). Heat to DHS is extracted directly from the heat source. It uses the residual heat after the ORC, which limits in turn the supply temperature level of the DHS. Note, that the heat amount to the DHS can be increased when the ORC is operated in part load, which results in a variable CHP coefficient (variable σ).

Parallel Concept. This concept is investigated e.g. by Hueffed and Mago [33] and shown in Fig. 2(b). Heat to DHS is extracted directly from the heat source. It is connected in parallel with the ORC to the heat source and thus the supply temperature is only limited by the temperature level of the heat source. Note, that the heat source can be used flexibly either in the ORC or the DHS, which results in a flexible CHP coefficient σ. The concept can be applied to heat sources with low and high temperatures.

Serial/Parallel Concept. This concept is reported e.g. by Pernecker and Uhlig [37] for the geothermal plant in Altheim and is shown in Fig. 2(c). It combines the advantages of the afore mentioned concepts. While the Parallel Concept offers high temperature levels for the district heating system, the Serial Concept allows a high utilisation of the heat source. This utilisation is only limited by the return temperature of the DHS.

Condensation Concept 1. This concept is investigated e.g. by Khennich et al. [38] and Clemente et al. [46] and is shown in Fig. 2(d). The heat to the DHS is extracted from the condenser of the ORC. The condensation pressure is limited by the necessary supply temperature of the DHS and due to the direct connection of DHS with ORC, the CHP coefficient σ is constant. Note, that this concept is only suitable for fairly high heat source temperatures and comparatively low temperature levels of the DHS.

Condensation Concept 2. This concept is shown in Fig. 2(e). Commercial products can be found by Orcan Energy AG for small scale applications [40]. Apart from an additional cooling device, this concept is similar to the Condensation Concept 1. Note, that the additional cooling tower allows a variable CHP coefficient σ and a power production with the ORC also in periods with less heat demand.

Serial/Condensation Concept. This concept is shown in Fig. 2(f) and it combines the advantages of the Serial Concept with the Condensation Concept. In case of high temperature applications, it allows a higher utilisation of the waste heat source.

Note, that the CHP concepts need a controllable heat source to cover base load heat demand, which limits their applicability for dynamic waste heat sources. This drawback can be overcome by the application of additional heat storage systems to buffer the dynamic behavior.

2.4. Flexibility of CHP concepts

A qualitative characterization of the flexibility in terms of a P_e-$\dot{Q}_{DHS}$-diagram is shown in Fig. 3. Note, that the above considerations neglect any heat losses resulting from heat transfer and within the district heating system and part load behavior, apart from assuming a minimal part load for the ORC of $P_{min} = 0.4 \cdot P_{max}$. Condensation Concept 2 (e) is highlighted with the grey shaded area. It shows a higher flexibility of heat provision than the Condensation concepts (d) and (f). However, this flexibility is associated with a lower overall utilization efficiency of the heat source. For the Serial and Parallel Concepts (a)-(c) the diagonal curves denote full load operation with the highest heat source utilization. Since all of these concepts have a variable CHP coefficient, operational points can be located in the hatched area, when lower utilisation efficiencies are accepted. The horizontal lines of concepts (a)-(c) at the bottom of the diagram show the heat-only mode, which is able to provide higher heat amounts to the district heating system when the ORC is switched off.

A qualitative comparison of the concepts is also given in Table 1. It can be concluded that the applicability and suitability of the state-of-the-art CHP concepts are strongly depending on the temperature level of the (waste) heat source T_{HS} and the temperature level of the DHS T_{DHS}. While the condensation concepts (Fig. 2 (d)-(f)) are only suitable for medium to high temperature heat source application and for fairly low DHS supply temperatures, the parallel concepts (Fig. 2(b) and (c)) are suitable for high and low temperature heat sources, where the supply temperature to the DHS is only limited by the temperature level of the heat source. Note, that the Serial Concept (2 (a)) is only suitable for medium to high temperature applications and the achievable temperature level is strongly influenced by the ORC and limited by the heat source.

Based on this characterization, the concepts can be allocated to cover specific heat loads. For this purpose, an annual load duration curve is exemplary shown in Fig. 4. In this figure base load, mid load and peak load are shown qualitatively by the highlighted

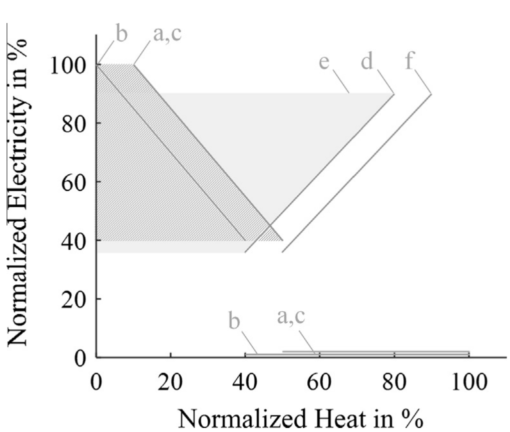


Fig. 3. Qualitative P_e-$\dot{Q}_{DHS}$-diagram for different CHP concepts.

areas. While base load is supposed to lead to high full load operation hours (>6000 h/year), peak load has fairly small operational hours (<2000 h/year) [47]. Everything in between is considered as mid load. The more flexible a CHP system is, the more suitable it is to achieve a high coverage of the heat demand. In most cases peak load is covered by a fossil fuel fired peak load boiler.

2.5. Operational strategies of CHP systems

Andrews et al. [49] distinguished four major operation modes for CHP plants: (1) Matching the electrical base load. Additional power is purchased from the grid and heat is covered either by CHP or additional boilers; (2) Matching the thermal base load in combination with a peak load boiler; (3) Matching the electrical load. For heat provision an additional boiler is used; (4) Matching the thermal load. Additional power is purchased from the grid. Operation modes (3) and (4) can be considered as the classical electricity-driven and heat-driven operation.

A least cost strategy has been investigated by Hawkes and Leach [50], which can be a mix of the above mentioned strategies. This operation mode is depending on the season, the overall coverage of the heat demand and market prices. Except for highly efficient systems such as fuel cells, the heat-driven operation leads always to minimal CO_2-emissions because it reduces unused waste heat [50,33], which is directly linked to primary energy consumption. In order to obtain a high coverage of the heat demand and to reduce the back-up capacity of fossil fuel fired peak load boilers, flexible CHP systems are favorable.

3. Simulation approach

This section provides an overview on the simulations and the working fluids applied. Furthermore, an innovative two-stage ORC-CHP concept is described, which will be compared against state-of-the-art concepts.

3.1. Simulation and assumptions

All cycle simulations are carried out in Aspen V8.8, which is a well-known simulation tool for process engineering and offers different component libraries, modules as well as fluid databases. Note, that the Peng-Robinson equation of state is used in Aspen to calculate fluid properties for these simulations.Simulation models have been developed for the Parallel Concept, Condensation Concept 1 & 2 as well as the proposed two-stage ORC concept for CHP applications. An overview of the applied boundary conditions, parameters and constraints is given in Table 2 and will be described in this section.

The heat source is assumed to be a hot gas stream coming from an internal combustion engine (ICE) or direct combustion of fuels and has a temperature level of t_{HS} = 490 °C at ambient pressure [51]. For simplification reasons the heat source fluid is assumed to be air. For reasons of comparability the energy content of $\dot{Q}_{HS}$ = 5 MW_t, with a reference temperature of 490 °C. A thermal oil loop transfers this heat to the ORC. In the present study, thermal oil temperatures of 240 °C and 340 °C are used, which are typical for waste heat from an ICE [51] and biomass [43], respectively. Tetradecamethylhexasiloxane (MD4M) is used as heat transfer fluid.

A pinch point of 10 K has been assumed for the design case at maximum duty of all heat exchangers. This pinch point is determined by a minimal temperature approach in Aspen, where the heat exchanger is discretized into 25 finite segments. The modelling of the part load behaviour for heat exchangers is based on the assumption of a constant UA-value, which is determined from

Table 1
Qualitative comparison of state-of-the-art CHP concepts.

	T of heat source	T limitation of DHS	σ	DHS heat extraction
(a) Serial Concept	High	$T_{HS} > T_{DHS}$	Variable	Heat source
(b) Parallel Concept	Low/high	$T_{HS} \geqslant T_{DHS}$	Variable	Heat source
(c) Serial/Parallel Concept	Low/high	$T_{HS} \geqslant T_{DHS}$	Variable	Heat source
(d) Condensation Concept 1	High	$T_{HS} \gg T_{DHS}$	Constant	Condenser
(e) Condensation Concept 2	High	$T_{HS} \gg T_{DHS}$	Variable	Condenser
(f) Serial/Condensation Concept	High	$T_{HS} \gg T_{DHS}$	Constant	Condenser + heat source

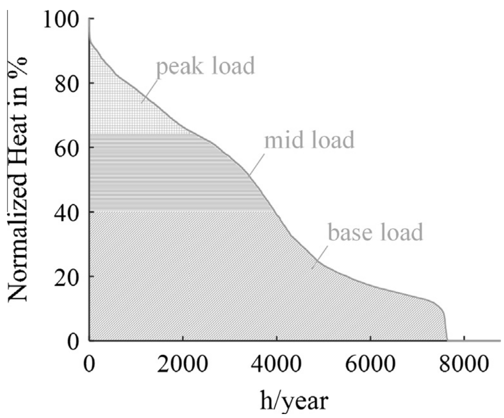


Fig. 4. Annual load duration curve for a district heating network in Germany [48] with an indication of different load bands.

the design case. Rovira et al. [52] investigate supercritical combined cycle gas turbine (CCGT) systems and mention, that this approach is appropriate for a first approximation of the part load behaviour. Moreover, they state, that the actual part load performance of a heat exchanger is always better due to the actual oversized heat exchangers [52], which in turn reduce pinch point limitations. Furthermore, this approach is also used in Ebsilon® Professional, a commercial software tool for cycle simulations. Note, that the part load behaviour for the expansion machine is not further specified, but a minimum load of $P_{min} = 0.4 \cdot P_{max}$ is defined as a boundary condition.

The live vapor pressure of the cycle has been determined for maximum power output. The assumed temperatures of the district heating with 80 °C are in line with literature values found e.g. in [53,54].

3.2. Working fluid selection

In this study the focus is on isentropic working fluids and siloxanes, where the latter are dry in nature and state-of-the-art for biomass applications. This section describes the working fluid selection in more detail.

Among others, fluorinated alkanes such as R245fa and the related R245ca are considered, which are so called Hydrofluorocarbons (HFC). In particular R245fa is an often applied and investigated fluid, which is also used in experimental test rigs [55,39] and operating plants nowadays.

In order to circumvent environmental-related issues, like high global warming potential, new molecules with lower impact on the climate are available. Thus, representatives of Hydrofluoroolefins (HFO) also called fourth generation fluids are included in the analysis, which are promising replacements for currently applied fluids [55]. Namely R1234zeZ and R1233zd-E are considered.

Cyclobutane, Cyclopentene, Furan and 2,5-Dihydrofuran are selected from the group of cyclic molecules due to suitable critical parameters in the range of the chosen heat source temperatures. For readability, Furans are distinguished from the other two cyclic working fluids in respective figures.

Commercially available systems for high temperature biomass and industrial waste heat applications are mainly operated by siloxanes. Siloxanes feature high critical temperatures in conjunction with good safety and environment related properties. Thus, in the present study OMTS is considered in the analysis as a representative siloxane. To conclude, representatives of HFC, HFO, cyclic compounds (cyclic) and Siloxanes are investigated. Table 3 lists the considered working fluids and corresponding critical parameters as well as fluid type and the condensation pressure at 30 °C.

3.3. Two-stage ORC concept for CHP applications

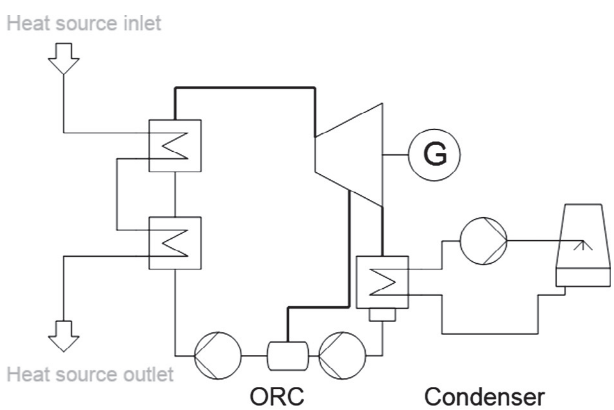
In the literature a two-stage ORC with turbine bleeding has been investigated (e.g. [4,29,5,7,30]). This concept is shown schematically in Fig. 5(a), where vapor from the turbine is extracted in order to saturate the liquid working fluid in a direct contact heat exchanger at an intermediate pressure level. This measure increases the thermal efficiency of the ORC. Meinel et al. [7] have shown, that this concept is especially suitable for wet and isentropic organic fluids and that the performance increase is significantly better compared to a recuperator concept operated with the same fluid. They concluded from simulations, that the highest thermal efficiency increase was obtained at extraction pressure levels which correspond to a saturation temperature of about 80 °C for R245fa. These findings support the idea of integrating the DHS heat extraction directly between the two turbine stages and the direct contact heat exchanger. The turbine bleeding pressure level and mass flow rates are adjusted in order to meet the boundary conditions from the DHS. The heat exchanger is rated for maximum heat decoupling and a pinch point of 10 K. In part load the determined UA value is kept constant. This concept is shown in Fig. 5(b).

Table 2
Global specifications of the simulations.

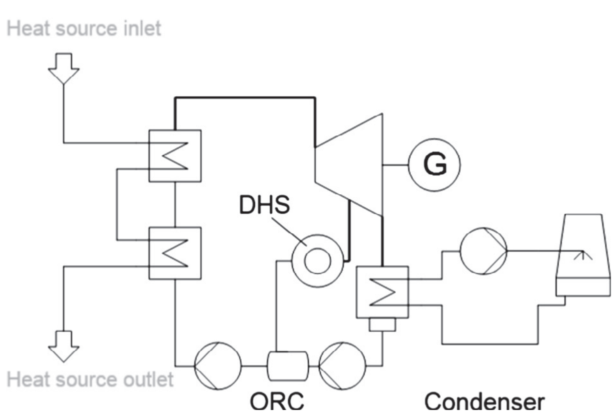
Parameter	Value	Parameter	Value
Mechanical efficiency	$\eta_m = 0.98$	Supply temperature in heating network	$T_s = 80$ °C
Generator efficiency	$\eta_{gen} = 0.95$	Return temperature in heating network	$T_r = 50$ °C
Pump isentropic efficiency	$\eta_{is,p} = 0.9$	Condensation temperature	$T_{cond} = 30$ °C
Turbine isentropic efficiency	$\eta_{is,t} = 0.8$	Temperature of heat transfer medium	$T_{HTM} = 240$ °C/340 °C
Design pinch point	$\Delta T_{pp} = 10$ K	Heat source temperature	$T_{HS} = 490$ °C
Heat source scale	$\dot{Q}_{HS} = 5$ MW$_t$		

Table 3
Investigated working fluids including fluid type and critical parameters.

Working fluid		Fluid type	t_{crit} [°C]	p_{crit} [bar]	p_{cond}(30 °C) [bar]
1,1,1,3,3-Pentafluoropropane	R245fa	Isentropic	154.29	36.50	1.77
1,1,2,2,3-Pentafluoropropane	R245ca	Isentropic/dry	174.58	39.30	1.21
cis-1,3,3,3-Tetrafluoropropene	R1234zeZ	Isentropic	150.08	35.30	2.10
trans-1-Chloro-3,3,3-trifluoro-1-propene	R1233zd-E	Isentropic	165.56	35.70	1.55
Cyclobutane		Isentropic	187.05	49.88	1.83
Cyclopentene		Isentropic/dry	234.11	48.05	0.61
2,5-Dihydrofuran		Isentropic/dry	269.1	55.13	0.29
Furan		Isentropic/dry	217.26	55.13	1.00
Hexamethyldisiloxan	MM	Dry	245.68	19.13	0.070
Octamethyltrisiloxane	MDM	Dry	290.94	14.15	0.008
Octamethylcyclotetrasiloxan	OMTS	Dry	313.49	13.38	0.003



(a) Two-stage ORC concept with turbine bleeding



(b) Two-stage ORC-CHP concept with turbine bleeding

Fig. 5. Two-stage ORC concepts for regenerative preheating.

The advantages of this proposed Two-stage Concept is the thermal efficiency increase of the ORC and the possibility to integrate the CHP heat extraction into the direct contact heat exchanger, which only slightly increases the system complexity. In case of continuous base load heat demand, the size of other components can be reduced as well (e.g. low-pressure turbine and condenser). However, the necessity of additional components results in higher investment costs compared to simple concepts. These higher investments can be compared by economic assessment.

4. Results

Since Meinel et al. [7] have already shown that the efficiency of a two-stage concept with turbine bleeding performs best with isentropic working fluids, the first section focuses specifically on these isentropic working fluids and the Two-stage CHP concept as shown in Fig. 5(b). Due to the dry expansion behaviour of siloxanes the recuperation of the sensible heat after the turbine outlet becomes more and more favorable. Therefore, a second section deals with a comparison of four different concepts for siloxanes as a working fluid. A final section presents a method to estimate annual electricity generation based on annual load duration curves.

4.1. Comparison of isentropic working fluids

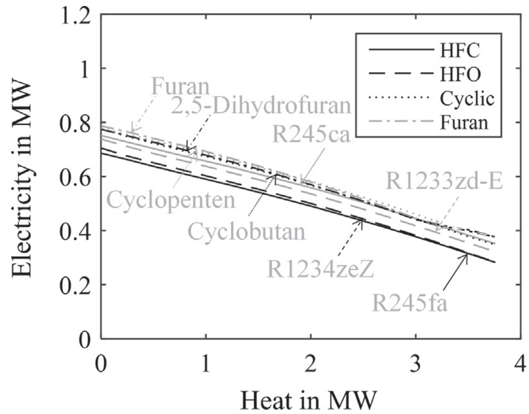
In Fig. 6, results of the CHP simulations with the Two-stage Concept for isentropic working fluids are shown. Two different temperature levels of the thermal oil are investigated, which represent waste heat from internal combustion engines (240 °C) and biomass application (340 °C). For the 240 °C case, Fig. 6(a) shows, that all the fluids perform similar and only minor differences are observed. Among the fluids, Furan performs best generating up to 787.6 kW_e. The next ranked fluids in terms of electric power output are Cyclobutane (−1.0%) and Cyclopentene (−1.0%) with slightly less power output compared to Furan. Note, that at a condensing temperature of 30 °C Cyclopentene, Furan and 2,5-dihydrofuran expand into vacuum (Table 3), which causes high sealing efforts for turbine and condenser to avoid non-condensable gases in the cycle. Noteworthy alternatives are Cyclobutane (−3.0%) and R1233zd-E (−8.5%). Especially R1233zd-E represents a good trade-off between environmental-related (low global warming potential, no ozone depletion potential), thermodynamic (relative high power output) and economic aspects (no vacuum expansion).

For a thermal oil temperature of 340 °C only 2,5-dihydrofuran and cyclopentene as working fluids offer higher power outputs compared to the 240 °C case. This increased power output is based on the fact, that the pinch point location changes from preheater outlet (240 °C) to the preheater inlet (340 °C) which increases efficiency. For the other fluids no significant increase in the power output is observed. With 2,5-dihydrofuran as the working fluid 1031.66 kW_e are generated followed by Cyclopentene with 939 kW_e (−11.6%) and Furan (−17.3%). The power output of Cyclobutene with 787 kW_e is almost identical with the calculated power output from the 240 °C case. The HFC and HFO fluids haven't been investigated due to significant thermal degradation above temperatures of 250 °C.

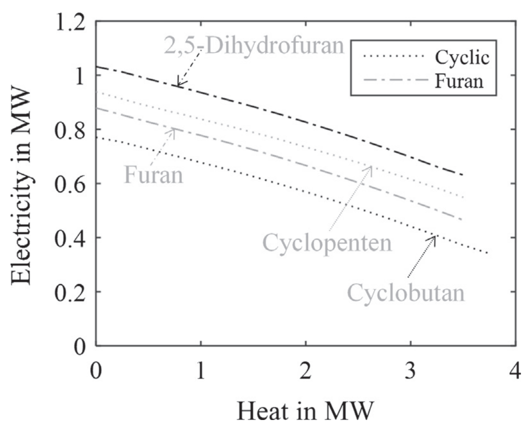
4.2. Comparison of siloxanes

Since siloxanes are suitable for high temperatures and mainly used for biomass applications, the comparison of these working fluids focuses only on high temperature thermal oil (340 °C).

Recuperation is a favored option for efficiency increase with these working fluids. Since commercial products for ORC-CHP with



(a) Thermal oil temperature 240 °C



(b) Thermal oil temperature 340 °C

Fig. 6. Performance of the two-stage concept for isentropic fluids and two different thermal oil temperatures in a P_e-$\dot{Q}_{DHS}$-diagram.

biomass and waste heat are available, the following concepts are considered as state-of-the-art [45]:

1. Parallel Concept with a recuperator to increase the thermodynamic efficiency (Figs. 1(b) combined with 2(b)).
2. Condensation Concept 1 without a recuperator (Fig. 2(d)). This concept is state-of-the-art for biomass fired power plants and residential, low temperature heating purposes.
3. Condensation Concept 2 without a recuperator (Fig. 2(e)). This concept is state-of-the-art for small scale CHP and enables maximum electricity generation for a heat-driven operation from a continuous heat source due to the flexible CHP coefficient.
   These above concepts will be compared against the proposed two-stage ORC concept for CHP applications.
4. Two-stage ORC-CHP Concept as shown in Fig. 5(b). This concept is beyond state-of-the-art CHP applications for ORC.

Fig. 7 shows the results of the four concepts for OMTS as reference working fluid, since it is found in commercial products [2]. It can be seen, that the Two-stage Concept offers a significantly higher power output compared to the Parallel Concept, if the decoupled heat to the DHS exceeds 1.45 MW_t indicated by the vertical line in Fig. 7.

The Condensation Concept 1 shows the lowest electricity output for all operational conditions and equals the Two-stage Concept with maximum heat decoupling. The Condensation Concept 2 is able to produce constant electricity output over the entire range of heat demand, but produces less electricity compared to

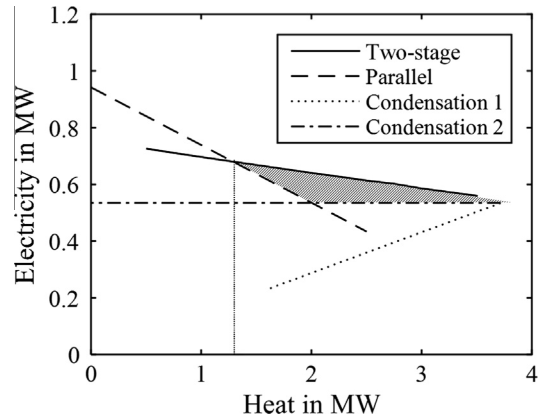


Fig. 7. Performance of dry fluids with different CHP concepts in a P_e-$\dot{Q}_{DHS}$-diagram.

the Two-stage Concept. Note, that the CHP coefficients of the Parallel Concept, the Two-stage Concept and the Condensation Concept 2 are variable, as was already described in Section 2 and Table 1. Operational points for these concepts can be found also below the respective performance curve, if lower utilisation efficiency is accepted. The shaded triangle in Fig. 7 denotes operational points for OMTS, where the proposed Two-stage Concept is more beneficial compared to the other investigated concepts.

Note, that the recuperator cycle needs in general higher investments for the additional heat exchanger, which is an expensive component due to its size. This large size of the recuperator comes from the low heat transfer coefficient for the vapor phase in conjunction with the desired low pressure losses after the turbine outlet and the condenser. On the other hand side, the Two-stage Concept needs additional components, such as an additional pump, a two-stage or multi-stage expansion machine and a receiver tank. This causes higher costs as well. It is not the scope of this work to quantify these costs due to high uncertainties associated with available cost functions [56]. Before assessing additional benefits from electricity sales, the Two-stage Concept is analyzed in terms of operating behavior.

The major difference of the Two-stage Concept compared to the other ORC-CHP concepts is the expansion with fluid extraction after the first stage. In order to analyze this concept in more depth, the extraction pressure is depicted in Fig. 8 for the whole operating range in terms of heat demand. As stated in Section 3.1, the extraction pressure is adjusted such, that the rated UA-value of the DHS heat exchanger is kept constant in part load condition. However, for a small amount of heat decoupling, the extraction pressure has been fixed to 0.012 bar, which corresponds to a pressure ratio

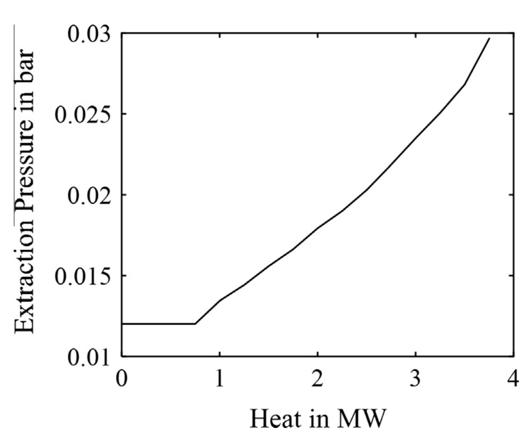


Fig. 8. Extraction pressure level for the turbine bleeding as a function of heat extraction.

of 4.4 bar for the low pressure stage. This is shown in Fig. 8, with the horizontal line at low heat demands.

It can be seen, that the extraction pressure varies by a factor of 2.25 from 0.012 bar for no heat demand up to 0.027 bar for the rated heat demand. This necessary variation of the extraction pressure is caused by the oversized heat exchanger area in part load condition leading to a decreasing pinch point.

This effect is shown in Fig. 9, where the T-Q diagram of the DHS heat exchanger is depicted for the rated heat demand (solid line) with a specified pinch point of 10 K as well as the part load condition with a heat demand of 2 MW (dashed line) and a pinch point of only 3.1 K. The decreasing pinch point in part load conditions leads to decreasing condensation temperatures and thus, to decreasing pressure.

However, the variation of extraction pressure in part load condition has huge influence on the expansion machine, since the pressure ratio of both, the high pressure stage and the low pressure stage is changing significantly as it is depicted in Fig. 10. The pressure ratio of the high pressure stage changes from 1000 at low heat demand to 400 at rated heat demand. However, the pressure ratio and its variation in part load is very high, it is possible with radial turbines to guarantee this performance. For example, Uusitalo et al. [57] analyzed a high-speed turbogenerator with a radial turbine using different siloxanes as working fluids. They investigated a recuperative cycle design with two different condensation temperatures concluding with turbine pressure ratios between 200 and 1700 for OMTS.

In the low pressure stage, not only the pressure ratio varies in part load but also the mass flow rate, because of changing extraction mass flow. At maximum heat demand almost no mass flow is directed to the low pressure stage, while with no heat demand, only a little part is extracted after the high pressure stage. This causes huge variation of the volume flow rate and the flow velocity at the expander inlet. With these operation condition a turbomachine might not be suitable. Instead, a volumetric expander, such as a scroll or a screw expander, can be used as low pressure stage. In such machines the volume flow rate is directly linked to the rotational speed of the machine due to a fixed built-in volume ratio. In case of a constant mass-flow rate, a reduction of the rotational speed leads to an increase in live vapor pressure ratio over the low pressure expander. In that way, the variation in the pressure ratio as well as the changing flow rates can be dealt with a speed control. Furthermore, the necessity of the fixed extraction pressure at low heat demands becomes clear due to a maximum rotational speed of the expander.

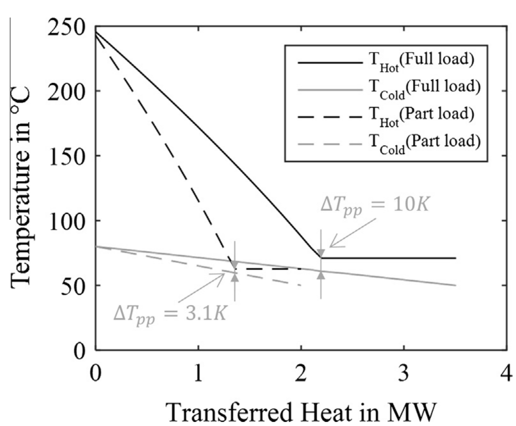


Fig. 9. T-Q diagram for full load and part load heat extraction.

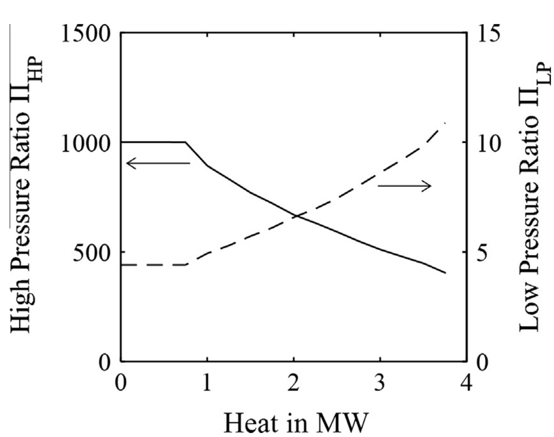


Fig. 10. Pressure ratios in the high pressure and low pressure part as a function of heat extraction.

4.3. Estimation of energetic benefits

This calculation procedure is based on annual load duration curves. If no measured data is available, models are found in literature to approximate such curves. One of these models is the Sochinsky model (e.g. [58]), which determines the time dependent heat demand $\dot{Q}_{DHS}(\tau)$ in dependence of specific parameters

$$\dot{Q}_{DHS}(\tau) = \dot{Q}_{DHS,max}\left(1 - (1 - \beta_{min}) \cdot \tau^{\frac{\bar{\beta}-\beta_{min}}{1-\bar{\beta}}}\right) \tag{4}$$

with the maximum heat demand of the district heating system $\dot{Q}_{DHS,max}$ and the dimensionless hour of the year τ, the minimal load factor of the DHS β_{min} and the mean load factor of the DHS $\bar{\beta}$ throughout the year. This mean load factor can be considered as equivalent to the ratio of full load operational hours per year z_{flh} and the total heating hours per year $z_a \leqslant 8760$ h/a

$$\bar{\beta} = \frac{z_{flh}}{z_a}. \tag{5}$$

The minimal load factor β_{min} can be calculated as the ratio between $\dot{Q}_{DHS,min}$ and $\dot{Q}_{DHS,max}$ being the minimum and maximum heat demands in the DHS, respectively:

$$\beta_{min} = \frac{\dot{Q}_{DHS,min}}{\dot{Q}_{DHS,max}}. \tag{6}$$

The resulting normalized annual load duration curve assuming $z_a = 8760$ h/a (continuous heat demand), $z_{flh} = 2500$ h/a and $\dot{Q}_{DHS,min} = 0.1 \cdot \dot{Q}_{DHS,max}$ is shown in Fig. 11. In addition to that, the grey curve represents data from a real district heating network in southern Germany. It can be seen, that both load profiles differ significantly. Note, that annual load duration curves for a real

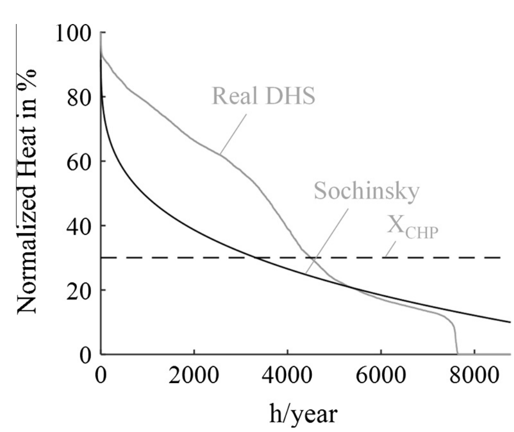


Fig. 11. Annual load duration curve according to the Sochinsky model and real operational data for a district heating network.

DHS differ, because they strongly depend on the type and number of consumers and are individual for each DHS. In order to take these variations into account, both curves will be used for further investigations.

The integration of a CHP system into the DHS is described by a cover ratio X_{CHP} which denotes the share of the maximum heat demand in the district heating system $\dot{Q}_{DHS,max}$ that can be covered by the CHP system at full load operation $\dot{Q}_{CHP,max}$

$$X_{CHP} = \frac{\dot{Q}_{CHP,max}}{\dot{Q}_{DHS,max}}. \tag{7}$$

A cover ratio of 0.3 is shown by the dashed line in Fig. 11. This cover ratio is varied for the investigated CHP concepts and the cumulated generated electricity from a heat-driven operation strategy is compared. Note that the part load of the ORC is limited by $P_{min} = 0.4 \cdot P_{max}$. Fig. 12 shows the cumulated electricity as a function of the cover ratio for the different concepts and the different annual load duration curves. For high cover ratios the Parallel Concept exceeds the Two-stage Concept, while the Two-stage Concept is similar to the Condensation Concept for low cover ratios. In between the generated electricity from the Two-stage Concept is significantly higher than for the other concepts. These findings are valid for the Sochinsky model as well as the real DHS. Therefore, it can be concluded that the Two-stage Concept is suitable for the whole range of cover ratios, while the condenser concept and the Parallel Concept are either limited in their application or in the produced electricity.

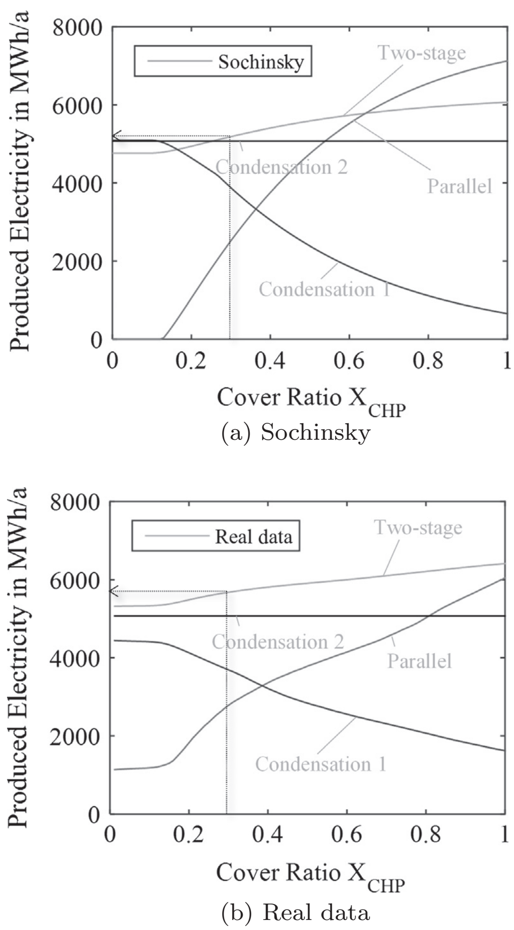


Fig. 12. Annual cumulated electricity production for CHP depending on the cover ratio.

According to [59] cover ratios of $0.1 \leqslant X_{CHP} \leqslant 0.4$ are common for real CHP plants and offer best profitability due to high full load operation hours. For $X_{CHP} = 0.3$ - highlighted by the additional shaded arrow line in Fig. 12 - the data are shown in Table 4. For the annual load duration curve of Sochinsky (Fig. 12(a)), the Two-stage Concept is able to generate almost 5164 MWh_e/a, while the Condenser Concept 1 (3972 MWh_e/a), the Condenser Concept 2 (5073 MWh_e/a) and the Parallel Concept (2383 MWh_e/a) produce less electricity. For the DHS based on real data (Fig. 12(b)), the results are even more beneficial: While the Two-stage Concept produces 5675 MWh_e/a, the Condenser Concept 1 produces 35% less electricity output. The Condenser Concept 2 (−10.6%) and the Parallel Concept produce 51% less electricity output with an efficiency of 10.6%.

For estimating the economical benefit, revenues are calculated by taking electricity prices into account, which range around $c_{e,EEX} \approx 40$ €/MWh_e at the European stock exchange and $c_{e,EEG} = 105.50$ €/MWh_e for biomass applications with CHP according to the German renewable energy act [60]. These prices lead to annual revenues of $R_e(EEX)$ = 206,556 €/a and $R_e(EEG)$ = 544,791 €/a for the Two-stage Concept and the Sochinsky-like DHS. For the real data DHS the respective values are $R_e(EEX)$ = 227,005 €/a and $R_e(EEG)$ = 598,725 €/a. Compared to the Parallel Concept the revenues of the Two-stage Concept are $R_e(EEG)$ = 293,431 €/a higher.

5. Discussion

Two-stage ORC concepts have been already addressed in recent literature. There it has been shown, that the regenerative preheating is more beneficial than recuperation for isentropic fluids [5,7]. Therefore, the proposed Two-stage ORC-CHP cycle has been investigated for different fluid families which show an almost isentropic behavior of the saturation curve. The selection of the fluids is considered to be representative and of relevance. The P_e-$\dot{Q}_{DHS}$-diagram for these fluids reveals, that the best power output can be achieved for heat source temperatures of 240 °C with Furan and for a heat source of 340 °C with 2,5-Dihydrofuran, however both fluids are flammable. The HFO fluid family are promising alternative working fluids to replace conventional refrigerants of the HFC family, since they combine environmental and safety aspects [55]. However,

Table 4
Concept comparison for $X_{CHP} = 0.3$ for electricity output and two different gratuity cases.

MWh	Integrated	Parallel	Condenser 1	Condenser 2
Real data	5675.12	2782.51	3689.92	5072.82
Sochinsky	5163.90	2382.56	3971.56	5072.82
Real data (40 €/MWh)	227004.76 €	111300.52 €	147596.79 €	202912.73 €
Sochinsky (40 €/MWh)	206556.18 €	95302.54 €	158862.47 €	202912.73 €
Real data (105.50 €/MWh)	598725.04 €	293555.13 €	389286.53 €	535182.33 €
Sochinsky (105.50 €/MWh)	544791.93 €	251360.45 €	418999.76 €	535182.33 €
Real data (Δ)	Reference	−50.97%	−34.98%	−10.61%
Sochinsky (Δ)	Reference	−53.86%	−23.09%	−1.76%

Table 5
Comparison of CHP-flexibility for isentropic fluids.

240 °C	District heat demand: 0 MW_t			District heat demand: 3 MW_t		
Working fluid	η_e	η_u	σ	η_e	η_u	σ
R245fa	13.7%	13.7%	∞	7.6%	67.6%	0.13
R245ca	15.0%	15.0%	∞	8.9%	68.9%	0.15
R1234zeZ	14.1%	14.1%	∞	7.6%	67.7%	0.13
R1233zd-E	14.8%	14.8%	∞	8.4%	68.4%	0.14
Cyclobutane	15.6%	15.6%	∞	8.9%	68.9%	0.15
Cyclopentene	15.6%	15.6%	∞	9.2%	69.2%	0.15
2.5-Dihydrofuran	15.5%	15.5%	∞	8.8%	68.8%	0.15
Furan	15.7%	15.7%	∞	8.9%	68.9%	0.15
340 °C	District heat demand: 0 MW_t			District heat demand: 3 MW_t		
Working fluid	η_e	η_u	σ	η_e	η_u	σ
Cyclobutane	15.41%	15.41%	∞	8.84%	68.86%	0.15
Cyclopentene	18.78%	18.78%	∞	12.31%	72.33%	0.21
2.5-Dihydrofuran	20.63%	20.63%	∞	13.94%	73.96%	0.23
Furan	17.57%	17.57%	∞	10.72%	70.74%	0.15

Table 6
Concept comparison of CHP-flexibility for OMTS.

340 °C	District heat demand: 0 MW			District heat demand: max			
Working fluid	η_e	η_u	σ	η_e	η_u	σ	
Two-stage Concept	16.1%	16.1%	∞	11.6%	86.6%	0.15	3.7 MW_t
Parallel Concept	20.2%	20.2%	∞	10.3%	60.0%	0.21	2.5 MW_t
Condensation Concept 1	5.0%	35.3%	0.16	11.6%	86.6%	0.16	3.7 MW_t
Condensation Concept 2	11.6%	11.6%	∞	11.6%	85.9%	0.16	3.7 MW_t

they do not offer the highest power outputs as compared to cyclic molecules. Nevertheless, their system efficiencies performs slightly better than with R245fa.

The applied part load behavior in the simulations is based on the assumption of a constant UA-value. This results in a higher power output due to the overrated heat exchangers when operated in part load, which in turn reduces pinch-point limitations. However, a detailed heat exchanger rating is advised for future work in order to take pressure losses and the exact heat transfer coefficient into account, as it has been applied in [14].

The part load behaviour of the expansion machine should be included as well in future modelling work in order to assess possible implementation. So far only a limitation of $P_{min} = 0.4 \cdot P_{max}$ is applied. Stodola law and additional loss mechanisms are expected to influence the results.

Since recuperation is supposed to be most beneficial for siloxanes, a detailed concept comparison is performed for different ORC-CHP concepts with OMTS as the working fluid. The bench mark has been chosen in accordance with available commercial products [45]. For low heat demands (e.g. <1.4 MW_e) the Parallel Concept with recuperator is still the preferred option. But for higher heat demands the Two-stage Concept becomes significantly more efficient and performs for high heat loads similar to a condensation concept.

Additional revenues caused by the higher efficiency are significantly larger than for other concepts. Note, that additional components, such as pump, two-stage turbine and receiver tank cause higher costs of the proposed system. Therefore the energetic benefits due to the increased efficiency are estimated. This approach is based on representative annual load duration curves and a heat-driven operation strategy. These calculated benefits depend strongly on heat and electricity prices. Thus, they can be considered as market dependent. The advantage of this method is, that it does not suffer the drawback of high uncertainties associated with cost functions for specific components. However, in order to have a complete economic assessment, the component costs need to be taken into account in future work.

Note, that the utilisation of the heat source is fluid dependent. Therefore the maximum heat to the district heating system depends also on the fluid and is not the same for all the fluids. For this reason, the characteristic parameters from Section 2.2 are referred to a common heat demand of 3 MW. Tables 5 and 6 show the system efficiencies of the 5 MW heat source, which takes the recuperable heat into account. Based on the electric efficiency and the utilisation efficiency the CHP-coefficient is determined. Table 5 shows the fluid comparison of isentropic fluids. If there is no heat demand of the district heating system, the utilisation efficiency is equivalent to the electric efficiency. At a heat demand of 3 MW, the utilisation efficiency is in the range of 67–69% (240 °C) and 68–74% (340 °C). The electric efficiencies increase significantly from 240 °C to 340 °C. Table 6 shows the concept comparison for siloxanes. It can be seen, that the Parallel Concept offers highest electric efficiencies, but the available district heat is limited to 2.5 MW for co-generation. In contrast to this, the condensation concepts have low electric efficiencies, but a significantly higher available heat. Based on the parameters, the proposed Two-stage Concept is a good trade-off between Parallel Concept and Condensation Concept with excellent performance.

6. Conclusion

A Two-stage ORC-CHP system has been proposed, where heat extraction to the DHS is integrated into the regenerative preheating. The following conclusions can be drawn from the presented results:

1. For different isentropic working fluids the performance and the flexibility of the Two-stage ORC-CHP system was investigated and shows a flexible CHP coefficient combined with high

electric efficiencies. A comparison of isentropic working fluids showed, that Furan and 2,5-Dihydrofuran perform best for temperatures of 240 °C and 340 °C of the heat transfer fluid.

2. It was shown, that the excess heat transfer area during part load condition has a positive effect on the electricity production, since pinch point limitations are reduced during part load operation.
3. Moreover, advantages for dry fluids were identified as well. Since the proposed concept shows a high and continuous electricity production during heat decoupling to the DHS, the system becomes beneficial for high heat demands. However, for small heat demands, a Parallel ORC-CHP concept with recuperator is still the best option.
4. Large cover ratios of the DHS heat demand, lead to a significant part load operation of the ORC. Therefore, the Two-stage Concept becomes more beneficial for such large cover ratios.
5. Finally, the benefits of such a Two-stage ORC have been supported by annual cumulated electricity revenues. However, these additional revenues depend strongly on the shape of the annual load duration curve as well as heat and electricity prices.

References

[1] Schuster A, Karellas S, Kakaras E, Spliethoff H. Energetic and economic investigation of organic Rankine cycle applications. Appl Therm Eng 2009;29(8–9):1809–17. http://dx.doi.org/10.1016/j.applthermaleng.2008.08.016.

[2] Quoilin S, van den Broek M, Declaye S, Dewallef P, Lemort V. Techno-economic survey of organic Rankine cycle (ORC) systems. Renew Sustain Energy Rev 2013;22:168–86. http://dx.doi.org/10.1016/j.rser.2013.01.028.

[3] Tchanche BF, Lambrinos G, Frangoudakis A, Papadakis G. Low-grade heat conversion into power using organic Rankine cycles – a review of various applications. Renew Sustain Energy Rev 2011;15(8):3963–79. http://dx.doi.org/10.1016/j.rser.2011.07.024.

[4] Mago PJ, Chamra LM, Srinivasan K, Somayaji C. An examination of regenerative organic Rankine cycles using dry fluids. Appl Therm Eng 2008;28(8–9):998–1007. http://dx.doi.org/10.1016/j.applthermaleng.2007.06.025.

[5] Branchini L, Pascale A, Peretto A. Systematic comparison of ORC configurations by means of comprehensive performance indexes. Appl Therm Eng 2013;61(2):129–40. http://dx.doi.org/10.1016/j.applthermaleng.2013.07.039.

[6] Tchanche BF, Pétrissans M, Papadakis G. Heat resources and organic Rankine cycle machines. Renew Sustain Energy Rev 2014;39:1185–99. http://dx.doi.org/10.1016/j.rser.2014.07.139.

[7] Meinel D, Wieland C, Spliethoff H. Effect and comparison of different working fluids on a two-stage organic Rankine cycle (ORC) concept. Appl Therm Eng 2014;63(1):246–53. http://dx.doi.org/10.1016/j.applthermaleng.2013.11.016.

[8] Zare V. A comparative exergoeconomic analysis of different ORC configurations for binary geothermal power plants. Energy Convers Manage 2015;105:127–38. http://dx.doi.org/10.1016/j.enconman.2015.07.073.

[9] Gleinser M, Wieland C. The misselhorn cycle: batch-evaporation process for efficient low-temperature waste heat recovery. Energies 2016;9(5):337. http://dx.doi.org/10.3390/en9050337.

[10] Wang J, Dai Y, Gao L. Exergy analyses and parametric optimizations for different cogeneration power plants in cement industry. Appl Energy 2009;86(6):941–8. http://dx.doi.org/10.1016/j.apenergy.2008.09.001.

[11] Heberle F, Brüggemann D. Exergy based fluid selection for a geothermal organic Rankine cycle for combined heat and power generation. Appl Therm Eng 2010;30(11–12):1326–32. http://dx.doi.org/10.1016/j.applthermaleng.2010.02.012.

[12] Wang J, Jing Y, Zhang C, Zhai Z. Performance comparison of combined cooling heating and power system in different operation modes. Appl Energy 2011;88(12):4621–31. http://dx.doi.org/10.1016/j.apenergy.2011.06.007.

[13] Tempesti D, Manfrida G, Fiaschi D. Thermodynamic analysis of two micro CHP systems operating with geothermal and solar energy. Appl Energy 2012;97:609–17. http://dx.doi.org/10.1016/j.apenergy.2012.02.012.

[14] Lecompte S, Huisseune H, van den Broek M, Schampheleire Sd, Paepe Md. Part load based thermo-economic optimization of the organic Rankine cycle (ORC) applied to a combined heat and power (CHP) system. Appl Energy 2013;111:871–81. http://dx.doi.org/10.1016/j.apenergy.2013.06.043.

[15] Maraver D, Quoilin S, Royo J. Optimization of biomass-fuelled combined cooling, heating and power (CCHP) systems integrated with subcritical or transcritical organic Rankine cycles (ORCs). Entropy 2014;16(5):2433–53. http://dx.doi.org/10.3390/e16052433.

[16] Uris M, Linares JI, Arenas E. Size optimization of a biomass-fired cogeneration plant CHP/CCHP (combined heat and power/combined heat, cooling and power) based on organic Rankine cycle for a district network in Spain. Energy 2015;88:935–45. http://dx.doi.org/10.1016/j.energy.2015.07.054.

[17] Chen H, Goswami DY, Stefanakos EK. A review of thermodynamic cycles and working fluids for the conversion of low-grade heat. Renew Sustain Energy Rev 2010;14(9):3059–67. http://dx.doi.org/10.1016/j.rser.2010.07.006.

[18] Karellas S, Leontaritis A-D, Panousis G, Bellos E, Kakaras E. Energetic and exergetic analysis of waste heat recovery systems in the cement industry. Energy 2013;58:147–56. http://dx.doi.org/10.1016/j.energy.2013.03.097.

[19] Beckmann M, Hurtado A. Kraftwerkstechnik: Sichere und nachhaltige Energieversorgung. Neuruppin: TK-Verl.; 2013.

[20] Angelino G, Colonna P. Multicomponent working fluids for organic Rankine cycles (ORCs). Energy 1998;23(6):449–63. http://dx.doi.org/10.1016/S0360-5442(98)00009-7.

[21] Liu Q, Duan Y, Yang Z. Effect of condensation temperature glide on the performance of organic Rankine cycles with zeotropic mixture working fluids. Appl Energy 2014;115:394–404. http://dx.doi.org/10.1016/j.apenergy.2013.11.036.

[22] Liu Q, Shen A, Duan Y. Parametric optimization and performance analyses of geothermal organic Rankine cycles using R600a/R601a mixtures as working fluids. Appl Energy 2015;148:410–20. http://dx.doi.org/10.1016/j.apenergy.2015.03.093.

[23] Heberle F, Brüggemann D. Thermo-economic evaluation of organic Rankine cycles for geothermal power generation using zeotropic mixtures. Energies 2015;8(3):2097–124. http://dx.doi.org/10.3390/en8032097.

[24] Quoilin S, Aumann R, Grill A, Schuster A, Lemort V, Spliethoff H. Dynamic modeling and optimal control strategy of waste heat recovery organic Rankine cycles. Appl Energy 2011;88(6):2183–90. http://dx.doi.org/10.1016/j.apenergy.2011.01.015.

[25] Manente G, Toffolo A, Lazzaretto A, Paci M. An organic Rankine cycle off-design model for the search of the optimal control strategy. Energy 2013;58:97–106. http://dx.doi.org/10.1016/j.energy.2012.12.035.

[26] Hu D, Zheng Y, Wu Y, Li S, Dai Y. Off-design performance comparison of an organic Rankine cycle under different control strategies. Appl Energy 2015;156:268–79. http://dx.doi.org/10.1016/j.apenergy.2015.07.029.

[27] Hernandez A, Desideri A, Ionescu C, Quoilin S, Lemort V, Keyser R d, editors. Experimental study of predictive control strategies for optimal operation of organic Rankine cycle systems, July 15–17, 2015.

[28] Roy J, Mishra M, Misra A. Performance analysis of an organic Rankine cycle with superheating under different heat source temperature conditions. Appl Energy 2011;88(9):2995–3004. http://dx.doi.org/10.1016/j.apenergy.2011.02.042.

[29] Yari M. Exergetic analysis of various types of geothermal power plants. Renew Energy 2010;35(1):112–21. http://dx.doi.org/10.1016/j.renene.2009.07.023.

[30] Meinel D, Wieland C, Spliethoff H. Economic comparison of ORC (organic Rankine cycle) processes at different scales. Energy 2014;74:694–706. http://dx.doi.org/10.1016/j.energy.2014.07.036.

[31] Beith R. Small and micro combined heat and power (CHP) systems: advanced design, performance, materials and applications. Woodhead Publishing series in energy. Oxford: Woodhead Publishing; 2011.

[32] Thekdi A. Waste heat to power: economic tradeoffs and considerations; September 25–26, 2007. <http://northwestchptap.org/NwChpDocs/Thekdi%20presentation.pdf>.

[33] Hueffed AK, Mago PJ. Energy, economic, and environmental analysis of combined heating and power-organic Rankine cycle and combined cooling, heating, and power-organic Rankine cycle systems. Proc Inst Mech Eng, Part A: J Power Energy 2011;225(1):24–32. http://dx.doi.org/10.1177/09576509JPE1063.

[34] Deutscher Bundestag. Gesetz für die Erhaltung, die Modernisierung und den Ausbau der Kraft-Wärme-Kopplung: KWKG; 2002.

[35] Mago PJ, Hueffed A, Chamra LM. Analysis and optimization of the use of CHP–ORC systems for small commercial buildings. Energy Build 2010;42(9):1491–8. http://dx.doi.org/10.1016/j.enbuild.2010.03.019.

[36] Freeman J, Hellgardt K, Markides CN. An assessment of solar-powered organic Rankine cycle systems for combined heating and power in uk domestic applications. Appl Energy 2015;138:605–20. http://dx.doi.org/10.1016/j.apenergy.2014.10.035.

[37] Pernecker G, Uhlig S. Low-enthalpy power generation with ORC-turbogenerator: the Altheim project, Upper Austria. GHC Bull 2002:26–30.

[38] Khennich M, Galanis N, Sorin M. Comparison of combined heat and power systems using an organic Rankine cycle and a low-temperature heat source. Int J Low-Carbon Technol 2013;8(Suppl. 1):i42–6. http://dx.doi.org/10.1093/ijlct/ctt028.

[39] Peris B, Navarro-Esbrí J, Molés F, González M, Mota-Babiloni A. Experimental characterization of an ORC (organic Rankine cycle) for power and CHP (combined heat and power) applications from low grade heat sources. Energy 2015;82:269–76. http://dx.doi.org/10.1016/j.energy.2015.01.037.

[40] Gewald D, Rostek K, Schuster A, Aumann R, editors. From technology development to (pre-series) product: the EPack hybrid; 2015.

[41] Hung T, Shai T, Wang S. A review of organic Rankine cycles (ORCs) for the recovery of low-grade waste heat. Energy 1997;22(7):661–7. http://dx.doi.org/10.1016/S0360-5442(96)00165-X.

[42] Wei D, Lu X, Lu Z, Gu J. Performance analysis and optimization of organic Rankine cycle (ORC) for waste heat recovery. Energy Convers Manage 2007;48(4):1113–9. http://dx.doi.org/10.1016/j.enconman.2006.10.020.

[43] Drescher U, Brüggemann D. Fluid selection for the organic Rankine cycle (ORC) in biomass power and heat plants. Appl Therm Eng 2007;27(1):223–8. http://dx.doi.org/10.1016/j.applthermaleng.2006.04.024.

[44] Karl J. Dezentrale Energiesysteme: Neue Technologien im liberalisierten Energiemarkt. 1st ed. München: Oldenbourg; 2004.
[45] Turboden. Turboden biomass solutions; 2015. <URL http://www.turboden.eu/en/public/downloads/12-COM.P-18-rev.17.pdf>.
[46] Clemente S, Micheli D, Reini M, Taccani R. Energy efficiency analysis of organic Rankine cycles with scroll expanders for cogenerative applications. Appl Energy 2012;97:792–801. http://dx.doi.org/10.1016/j.apenergy.2012.01.029.
[47] Spliethoff H. Power generation from solid fuels, power systems. Heidelberg and New York: Springer; 2010.
[48] W. Geisinger, Operational data of the geothermal power plant in Unterhaching; 2014.
[49] Andrews D, Vatopoulos K, Carlsson J, Papaioannou I, Zubi G. Study on the state of play of energy efficiency of heat and electricity production technologies. EUR (Luxembourg), vol. 25406. Luxembourg: Publications Office; 2012 [Online].
[50] Hawkes A, Leach M. Cost-effective operating strategy for residential micro-combined heat and power. Energy 2007;32(5):711–23. http://dx.doi.org/10.1016/j.energy.2006.06.001.
[51] Karellas S, Schuster A. Supercritical fluid parameter in organic Rankine cycle applications. Int J Thermodyn 2008;11(3):101–8.
[52] Rovira A, Valdés M, Durán M. A model to predict the behaviour at part load operation of once-through heat recovery steam generators working with water at supercritical pressure. Appl Therm Eng 2010;30(13):1652–8. http://dx.doi.org/10.1016/j.applthermaleng.2010.03.023.
[53] Paar A. Transformationsstrategien von fossiler zentraler Fernwärmeversorgung zu Netzen mit höheren Anteilen erneuerbarer Energien; 2013.
[54] Duminil E, Tereci A, Kesten D. Economical aspect and environmental impact of renewable trigeneration in urban areas: Scharnhauser park case study; 2009.
[55] Eyerer S, Wieland C, Vandersickel A, Spliethoff H. Experimental study of an ORC (organic Rankine cycle) and analysis of R1233zd-E as a drop-in replacement for R245fa for low temperature heat utilization. Energy 2016;103:660–71. http://dx.doi.org/10.1016/j.energy.2016.03.034.
[56] Lemmens S. A perspective on costs and cost estimation techniques for organic rankine cycle systems; 2015.
[57] Uusitalo A, Turunen-Saaresti T, Honkatukia J, Colonna P, Larjola J. Siloxanes as working fluids for mini-ORC systems based on high-speed turbogenerator technology. J Eng Gas Turb Power 2013;135(4):042305. http://dx.doi.org/10.1115/1.4023115.
[58] Carr A, Kellett P, Dubuisson X, Happold B. Procurement guidelines for wood biomass heating; 2005. <http://www.seai.ie/Renewables/Bioenergy/Wood_heating_procurement_guidelines.pdf>.
[59] Dötsch C, Taschenberger J, Schönberg I. Leitfaden Nahwärme; 1998.
[60] Deutscher Bundestag. Erneuerbare Energien Gesetz: EEG; 2014.