

Article

High-Efficiency Small-Scale Combined Heat and Power Organic Binary Rankine Cycles

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Abstract: Small-CHP (Combined Heat and Power) systems are generally considered a valuable technological option to the conventional boilers, in a technology developed context. If small-CHP systems are associated with the use of renewable energies (biomass, for example) they could play an important role in distributed generation even in developing countries or, in any case, where there are no extensive electricity networks. Traditionally the considered heat engines for micro- or mini-CHP are: the gas engine, the gas turbine (with internal combustion), the steam engine, engine working according to the Stirling and to the Rankine cycles, the last with organic fluids. In principle, also fuel cells could be used. In this paper, we focus on small size Rankine cycles (10–15 kW) with organic working fluids. The assumed heat source is hot combustion gases at high temperature (900–950 °C) and we assume to use only single stages axial turbines. The need to work at high temperatures, limits the choice of the right organic working fluids. The calculation results show the limitation in the performances of simple cycles and suggest the opportunity to resort to complex (binary) cycle configurations to achieve high net conversion efficiencies (15–16%).

Keywords: Organic Rankine Cycle (ORC); Rankine cycles; organic working fluids; small power size; dynamic expanders

1. Introduction

A sharp rise in global energy demand and ambitions to curtail global greenhouse gas emissions have led to various innovative alternatives in energy sector. Utilisation of low quality heat source, for example, is one of the key ingredient of the solution to improve efficiencies of current energy conversion and electricity generation processes. Among these ideas and technologies, the Organic Rankine Cycle (ORC) has gained much attention over almost a century. Unlike the conventional Steam Rankine Cycle, ORC uses unconventional fluids, low boiling and stable hydrocarbons mostly, to extract work from the medium to low grade sources with a broad power sizes interval.

Rankine cycles with organic fluids can also be used, in principle, to convert high temperature heat from any source in useful work, but applications in this context comes up against the thermochemical stability of the working fluid and the very high volumetric expansion ratios of the thermodynamic cycles, which involve many turbine stages, technologic difficulties and high costs, in particular for small-size engines. On the other hand, the low electric efficiency of the small-scale power generation systems, justifies and makes convenient to recover heat from the engine, to realise the so called CHP (Combined Heat and Power) systems. Nevertheless, choosing appropriate working fluid for the operation of ORC is essential for its thermo-economic viability, [1–3].

CHP-based generation systems have ramped up over the last decade especially in Europe. For instance, in UK only the share of CHP-based systems is around 5.69 GW by 2015, and about 10% of CHP systems used renewable fuel sources, [4]. However, the size of CHP plants and their connectivity to non-connected zones or off-grid areas is an issue.

With increasing global pressure for the reduction of the carbon footprint, mechanical utilisation of natural resources such as biomass, for small CHP engines, in an efficient and environment friendly manner is highly recommended. Indeed, for lesser grade biomass and derived fuels, several plants using organic fluids are already operational for low to medium power rating range (1–10 MW). Unfortunately, small scale ORC supplied by the external combustion of wood are affected by intrinsic rather low electrical efficiencies.

As it is well known among the main operational modes to convert solid and dry biomass in electrical energy there are the direct combustion and the gasification, [5]. Usually, the syn-gas from small scale gasifiers is utilised as fuel in internal combustion engines (ICE) or micro-gas turbines, [5–7].

With regard to the ICE, small CHP engines (nearly 10 kW) fuelled with natural gas, have electrical efficiencies approximately of 25%, [8]. Micro-gas turbines have turbo-machines rotating at very high speeds (50,000–250,000 rpm), with efficiencies of about 25% (in the power range about ten kW), decreasing to 10% when the power is some kW, [9]. External combustion micro-gas turbines, working with lower maximum temperatures, have also lower electric efficiencies. Potentially, the syn-gas can be used in fuel cells, also, [10].

Among the external combustion engines, Stirling engines, [11–13], and the small steam engines and steam turbines must also be mentioned.

Nevertheless, in this paper, we concentrate only on small size ORC (Organic Rankine Cycle) engines, without to have pretensions to compare each other all the available and usable different heat engines. The aim is, considering the limits of design and utilisation of single stage axial turbines, to discuss and to identify some technically possible configurations of the thermodynamic cycles. Exploring so the potential of small ORC engines at high temperatures.

As the specific application here considered is the heat recovery from a high temperature source, one has to expect very large volumetric expansion ratios (of the order of many tens). As a consequence, the natural expander to choose is a turbine. Volumetric expander combined with organic Rankine cycles are considered in literature, and, in accordance with their peculiar characteristics, the expansion ratios are limited to 2–10, [14,15]. Furthermore, (for some kW) the isentropic efficiency of the commercially available volumetric expanders, basically, decreases to fifty per cent for pressure ratios greater than about 10, [15].

As the engines working according to the Rankine cycles convert external heat into mechanical work, we assumed, as a reference example, the availability of a mass flow rate of syn-gas of about 60 kg h^{-1} (as available from some marketable small gasifiers), with a lower heating value of about 6.5 MJ m^{-3} (a value roughly equal to the lower heating values of wood chips with high moisture content –40 to 60%). The combustion of the syn-gas produces gases at high temperatures (900–950 °C), making it particularly difficult to achieve a full exploitation of the heat source by a simple organic Rankine cycle. These severe conditions, associated with the small size power, worsen the design problems, and to carry that thermodynamic engines out proves to be really demanding.

The topic is not entirely new. For example, in [16,17] some small engines from 3 kW to 100 kW (with solar, geothermal or exhaust gases as heat sources) are cited and described. In [18] the design of a small solar engine of 3 kW using scroll expanders is discussed. A supersonic turbine (at a design rotational speed of 100,000 rpm, with a measured efficiency of about fifty per cent) for a unit of 10 kW is presented in [19]. In [20] a 330 kW modular, hermetically sealed turbine-generator unit is discussed. The working fluid, a refrigerant, has an inlet design temperature of 200–220 °C and the expansion ratio is about 10–30. The turbine rotor and the high-speed generator are hermetically sealed and the shafts are magnetically levitated. The design turbine rotational speed is 20,000 rpm.

A 170 kW biomass Rankine engine is described in [21]. The considered working fluid is toluene, with turbine, electrical generator and pump running at 25,000 rpm.

Technical and economic reviews about small scale Rankine cycles with organic fluids are in [22,23].

Usually, the selected working fluids are common refrigerants, the maximum temperatures are rather low (though the maximum temperature of the gaseous heat source is high) or, on the contrary, the considered Rankine cycles have a substantial superheating, often with volumetric expanders, [24]. In this work we assume as calculation hypotheses the use of only one stage (axial) turbine, to minimise the mechanical complexity of the expander, and to resort only to saturated thermodynamic cycles that, fixed the maximum operative temperature, have potentially the better thermodynamic efficiency. The superheating could be useful but, it adds a further design variable and increases the thermal power of the recuperator. For simplicity it was here neglected.

These assumptions, together with the small power levels and the high temperature of the heat source, to a large degree influence the working fluids selection and the choice of the evaporation and condensation temperatures. Given the specific application, not many are the working fluids available on the market and the choice is reduced. A discussion of the properties of some available organic working fluid for high temperature applications is in Section 2.

A great current disadvantage of small power Rankine cycles is the considerable specific cost of the engine in comparison with the reciprocating internal combustion engines and, basically, in the small power sizes range, the lower efficiencies. Nevertheless, economic considerations (discussed, for example, in references [18,22,23]) are not here considered. In the following analysis, the efficiency of a hypothetical gasifier (or wood combustor) is not taken into account because we focused on the ORC engine.

2. The Working Fluids

In our case, the selection of the right working fluid was primarily conditioned by the low level of the useful mechanical power. In the cases here discussed the turbine power will result in about 10–15 kW; so, strictly from the thermodynamic point of view, the ideal working fluid should have (i) a large molecular mass, (ii) an high critical temperature and an high boiling point and (iii) a low molecular complexity, [25–28], and ([29], Chapter 3, p. 125). In addition, the fluid has to be thermally stable at the operating temperatures and, if possible, non corrosive and with a good environmental acceptability.

Unfortunately, frequently, high boiler temperatures and high molecular mass involve a great number of atoms and, as a consequence, a great molecular complexity and the need of recuperators after the expansion in turbine. The molecular complexity of a working fluid is a function of the number of atoms in the molecule. As the molar specific heat at constant pressure C_p of the vapour augments essentially with the number of atoms in the molecule, when this latter increases the vapour slope ($\partial S/\partial T$) in the Temperature-Entropy thermodynamic plane increases too. Fixed at such a temperature, when the (positive) local contribution of the heat capacity C_p to the slope, exceeds the (negative) contribution of the vapour pressure, the saturation dome inclines to the right. In this cases, the expansion ends in the superheating phase. In addition, fixed a pressure expansion ratio, the temperature drop during the expansion decreases and a recuperator after the expansion results really necessary. In Table 1 there are some properties of working fluids considered in this work. In Figure 1 are reported the vapour pressures of the considered fluids; in Figure 2 there are the dew and the bubble lines in the Temperature-Entropy plane. The three considered working fluids are all suitable for high temperature applications.

Decamethyl-tetra-siloxane (MD₂M) is a member of the siloxane family. Poly-dimethyl-siloxanes are largely used and are characterised by a relatively high thermal stability (used up to 250–300 °C). They are physiological inert and they have good lubricating properties. Siloxanes are used as working fluids in commercial biomass Rankine cycles, [30]. In [31] the design of a small radial turbine (of about

10 kW) with a siloxane is discussed. The evaluated turbine efficiency is said to be about 0.76, at a rotational speed of 31,000 rpm.

The eutectic mixture of diphenyl ((C₆H₅)₂, 26.5% in mass) and diphenyl oxide ((C₆H₅)₂O, 73.5% in mass), is a well-known synthetic heat transfer fluid. The mixture, according to the manufacturers, is utilizable up to 400 °C, [32] The p-terphenyl, the most common isomer among the terphenyls, has an high critical temperature (about 640 °C) and it is probably as thermally stable as the eutectic mixture of diphenyl and diphenyl oxide. Terphenyls, for their high boiling point and thermal stability, were considered and thoroughly studied as cooling and moderating fluids in some particular nuclear reactors, ([33], Chapter 17, p. 411). Among the organic fluids, the eutectic mixture of biphenyl oxide and biphenyl and the p-terphenyl are considered the most thermally stable. The eutectic mixture is proposed as heat transfer fluid up to 400 °C.

Given their boiling point (see Table 1 and Figure 1), the eutectic mixture of biphenyl oxide and biphenyl and the p-terphenyl can be used only at high condensation temperature and, in our case, they will be considered as working fluids in topping high-temperature cycles in binary systems.

Table 1. Some properties of the considered working fluids.

Fluid Property	MD ₂ M ^a	Biphenyl Oxide + Biphenyl ^b	p-Terphenyl
Critical Temperature (°C)	326.25	490	643.85
Critical Pressure (bar)	12.27	33.1	29.9
Boiling Temperature ^c (°C)	194.35	257	382
Melting Temperature ^c (°C)	−68	12 ^e	213
Molecular Weight	310.69	166.31	230.31
Parameter of Molecular Complexity ^d	64.19	33.48	64.36

^a Decamethyl-tetra-siloxane; ^b Eutectic Mixture; ^c at ambient pressure; ^d $\sigma = T_{cr}/R(\partial S/\partial T)_{so,T_r=0.7}$; ^e crystallizing point.

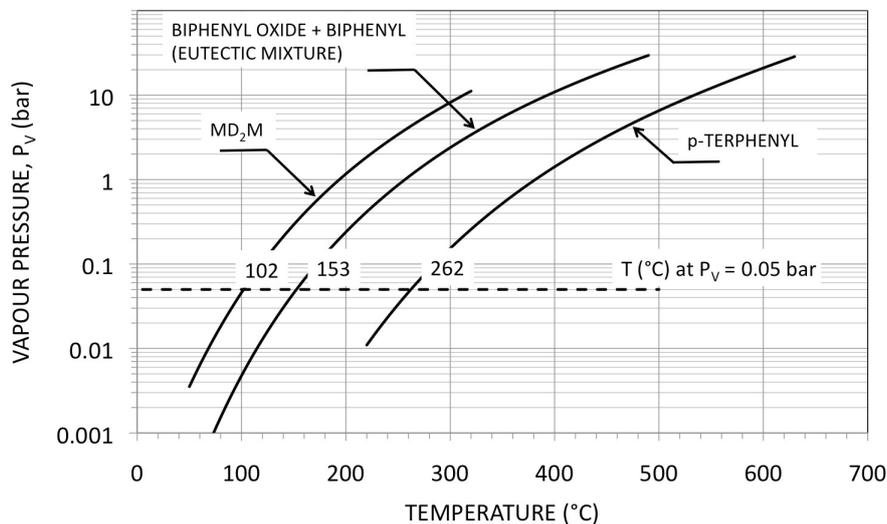


Figure 1. Vapour pressure of the considered working fluids as a function of the temperature.

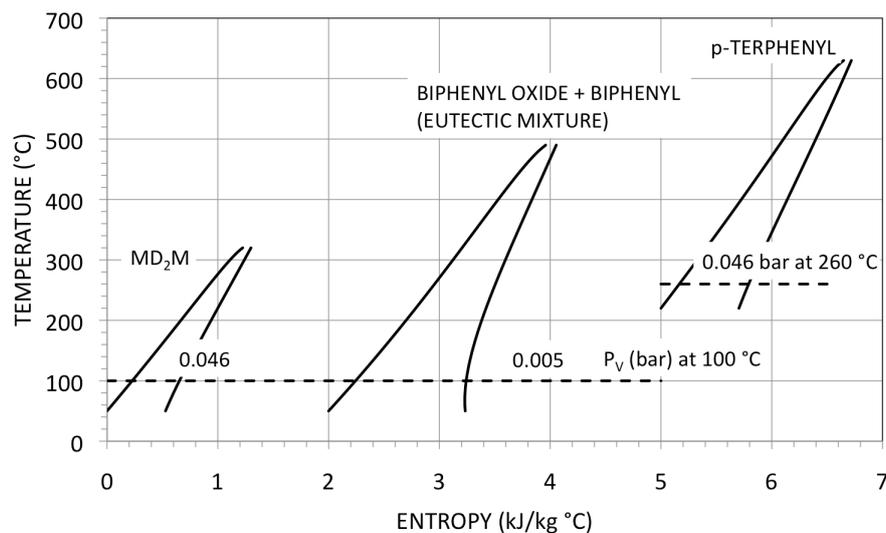


Figure 2. Dew and bubble lines of the considered working fluids in the Temperature-Entropy plane.

3. The Cycle Performances

The assumed parameters for the calculations are in Table 2. The product $\eta_M \times \eta_G$, or the product between the mechanical efficiency η_M and the generator efficiency η_G , is assumed unitary in the following calculations. An exception are the results in Tables 3 and 4, where the product $\eta_M \times \eta_G$ was assumed equal to 0.9.

Table 2. The main assumptions for the power cycles calculations.

Assumed Design Parameter	Value
Syn-gas mass flow rate (kg h^{-1})	60
Gas energy density (MJ m^{-3})	6.5
Pump and fans efficiencies ^a , η_P	0.5
Mechanical and electric generator efficiencies ^b , $\eta_M \times \eta_G$	1.0 or 0.9
Minimum temperature difference in the Recuperators, ΔT_R (°C)	20.0
Minimum temperature difference in the Primary Heat Exchanger, ΔT_{PHE} (°C)	25.0
Minimum temperature difference in the “Hot” Condenser ^c , $\Delta T_{C,HT}$ (°C)	20.0
No pressure drops (in the main thermodynamic cycle components)	-
No chimney radiation losses	-
Radiation losses in the combustion chamber	1%
Exhaust gaseous temperature (°C)	120

^a comprehensive of the mechanical and of the electrical efficiencies; ^b according to the considered conditions;

^c in a binary system, the condenser of the “High Temperature” cycles is, at the same time, the evaporator of the “Low Temperature” cycle.

To assign an exact value to the product $\eta_M \times \eta_G$ is not really simple. Usually, the friction power loss is proportional to the rotational speed and, for an alternator, in addition to electrical losses, the windage power loss increases with the cube of the rotational speed. Furthermore, the magnetic losses are also connected with the rotational speed. In reference [34], for example, for the design of a turbo-alternator of 10 kW, the assumed alternator and mechanical efficiencies were about eighty per cent and ninety seven per cent respectively. Where necessary and just as a reference, here, optimistically, we assumed for their product a value of ninety per cent.

In Figure 3 there is the considered plant scheme for the syn-gas boiler with a recovery Organic Rankine cycle. For the boiler section, for simplicity and just as a reference, we assumed a simplifying configuration of the biomass boiler described in [35]. The combustion chamber was simulated (using AspenPlus V7.3[®]), by a Gibbs Reactor, with a radiation loss equals to one per cent of the thermal

chemical reaction power. A circulation of an appropriate mass flow rate of gases assures a temperature in the stream A of 950 °C. A direct primary heat exchanger provides the heat recovery from the hot combustion gases to the working fluid of the Rankine cycle. The temperature in the stream B was assumed equal to 180 °C and an air pre-heater cools the gases to 120 °C, the temperature in stream C, before their discharge in the atmosphere. The fans C₁ and C₂ provide the circulation and the discharge of the exhausted gases, respectively. Radiation losses in the chimney stack were neglected. After the primary heat exchanger, before the point B, we supposed a possible heat recovery with water (heated from 25 °C to 90 °C); a second heat recovery is on the condenser of the ORC.

For the analysis of the performances of the different single Rankine cycle we supposed the following design constraints:

- a single stage axial turbine, to minimise the mechanical complexity of the expander;
- only saturated cycles. Firstly for simplicity, avoiding so a further design variable, and then to avoid to increase the thermal power of the recuperator. In addition, saturated cycles have, fixed the maximum and minimum temperatures, the better thermodynamic efficiency.

In the following, firstly, we will discuss the results with regard to the scheme configuration in Figure 3.

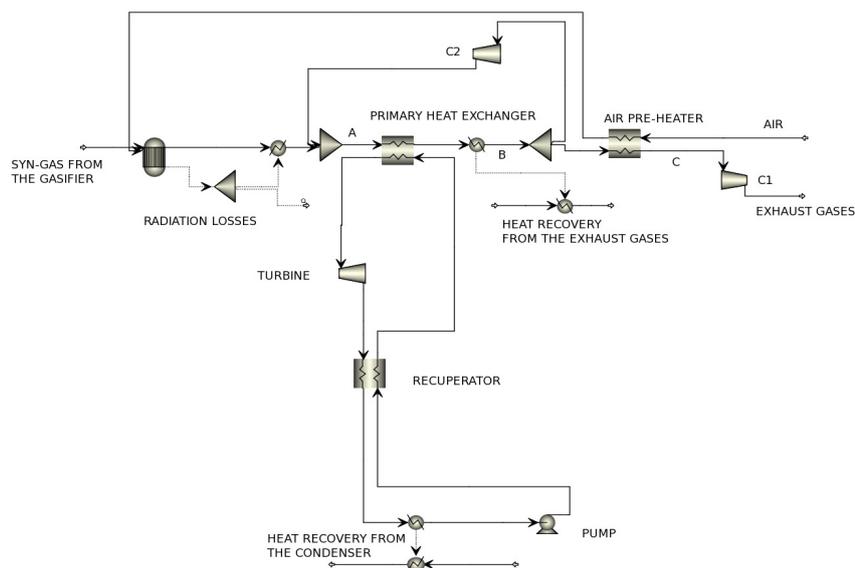


Figure 3. Plant scheme for the considered high temperature source. A single cycle ORC engine.

Case (a)—single heat recovery cycle

As the turbine efficiency depends on the size parameter $SP = \sqrt{\dot{V}_{out} / \Delta H_{is}^{0.25}}$ and on the isentropic volumetric expansion ratio $\dot{V}_{out} / \dot{V}_{in}$, [36], Figure 4 gives the parameter SP and the ratio $\dot{V}_{out} / \dot{V}_{in}$ for cycles with MD₂M as working fluid. In Figure 4 the condensation pressure P_C was assumed equal to 0.03 bar, 0.05 bar and 0.1 bar. The ratio $\dot{V}_{out} / \dot{V}_{in}$, fixed a condensation pressure P_C , results a function of the evaporation pressure and increases with it.

In the case of a condensation pressure equal to 0.03 bar, for example, (see Figure 4) it seems very difficult to design an efficient single-stage turbine; if $P_C = 0.1$ bar, the size parameter SP almost halves, but, as also the isentropic volumetric expansion ratio decreases substantially, it is possible to realise single stage turbines with a very good efficiency ($\approx 80\%$). At least for isentropic expansion ratios between 20 and 30.

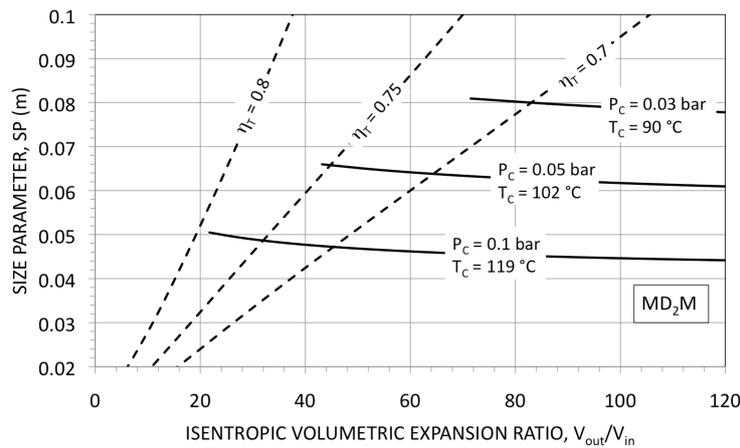


Figure 4. Turbine size parameter SP and isentropic volumetric expansion ratio for MD_2M . The parameter η_T represents the efficiency of a one-stage axial turbine. The relationship between turbine efficiency, size parameter and isentropic volumetric expansion ratio was taken from ref [36]. The solid lines are at different fixed condensation pressures P_C , increasing the ratio $\dot{V}_{out}/\dot{V}_{in}$ increases the evaporation pressure.

Assuming, as a reference, a condensation pressure P_C of 0.05 bar, in Figure 5 are the net cycle efficiency and the isentropic turbine power for different evaporation pressures P_E and for three values of turbine efficiency η_T (0.7, 0.75 and 0.8).

Points A, B, C and D represent different cycle configurations. For example, fixing $P_E = 2.5$ bar (evaporation temperature $T_E = 234$ °C), cycles B and C have the same isentropic expansion ratio ($\dot{V}_{out}/\dot{V}_{in} = 55$) and the same size parameter ($SP = 0.065$). Then, in principle, according to Figure 4, the turbine efficiency could take an intermediate value between 0.7 and 0.75. If $\eta_T = 0.7$ (cycle C) the cycle efficiency results about 0.116; if $\eta_T = 0.75$ (cycle B) the cycle efficiency increases to about 0.125.

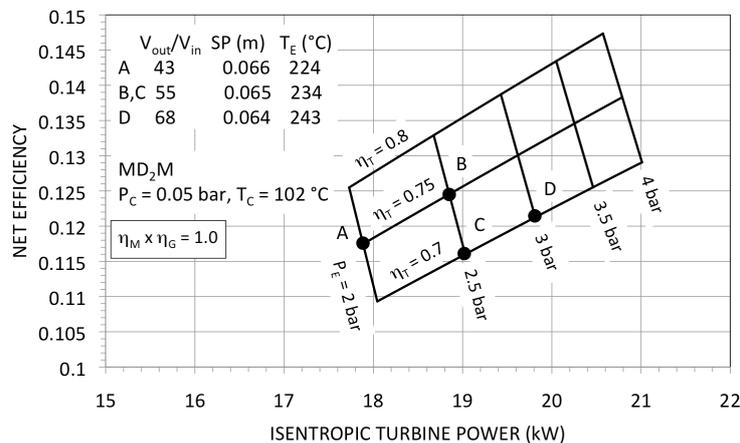


Figure 5. Net cycle efficiency and isentropic turbine power for MD_2M . The condensation pressure is fixed at 0.05 bar. The parameter η_T represents the efficiency of a one-stage axial turbine.

As the condensation is at about 102 °C ($P_C = 0.05$ bar) it is possible to recover heat from the condenser of the thermodynamic cycle and, consistently with the plant scheme assumed (see Figure 3), it is possible to recover heat also from the pre-cooling of the exhaust gases. Considering the recovered thermal power as a further useful effects, the a “First Law” efficiency (the CHP efficiency) for cycle B, for example, increases to about 0.9.

As you can derive from Figure 5, the evaporation temperature T_E , in the allowable cases A, B, C and D, is rather low in comparison with the maximum temperature of the available hot gases (950 °C),

so it is worth considering binary cycles too: a topping cycle at 350–400 °C and a bottoming cycle at 200–250 °C. As a matter of fact (with the considered constraints), it is impossible, using a single fluid, to realise the full expansion in a one stage turbine.

Case (b)—binary heat recovery cycle

Assuming as a basic constrain a one stage axial turbine, the need to limit the volumetric expansion ratio has as a result (fixed a condensation pressure P_E of 0.05 bar) an evaporation temperature T_E rather low: see Figures 4 and 5.

As the heat source is, according to the assumed model of the boiler, at a temperature of 950 °C, a possible option to attempt a better exploitation of its thermodynamic potential could be to resort to a binary system, see the scheme in Figure 6. The condenser of the “High Temperature” module provides the heat for the evaporation of the “Low Temperature” thermodynamic cycle.

Assuming MD₂M as working fluid in the “Low Temperature” module, Figures 7 and 8 give the size parameter SP and the isentropic volumetric expansion ratio $\dot{V}_{out}/\dot{V}_{in}$ for the HT turbine. In Figures 7 and 8 an eutectic mixture of biphenyl oxide and biphenyl and pure p-terphenyl as working fluids in the HT cycle are considered, respectively.

Against $\dot{V}_{out}/\dot{V}_{in}$ ratios of the same magnitude, the turbine with p-terphenyl has size parameters four times greater than the corresponding values for the eutectic mixtures, allowing so the design of better turbines.

The results of net cycle efficiency and of isentropic power for binary cycles with the eutectic mixture and with p-terphenyl as working fluids in the HT cycles are in Figures 9 and 10, respectively. Resorting to binary systems, the net efficiency increases from about 12–13% to about 18–19% (with $\eta_M \times \eta_G = 1.0$). In all considered cases the CHP efficiencies result about 90%.

In Table 3 there are some results for the cycle corresponding to the point B in Figure 5. Assuming $\eta_M \times \eta_G = 0.9$, the cycle efficiency results 12.5%, while the net efficiency is 11%. The recovered thermal power, supposed to be available in hot water at 90 °C, is about 6 times the electrical turbine power.

In Table 4 there are the results for two binary configurations. In the first case, the working fluid for the HT section is terphenyl (case C of Figure 10), in the second one the working fluid is the eutectic mixture of biphenyl oxide and biphenyl (case D of Figure 9). The gain in electric efficiency is of 0.051 points in the first case and 0.041 in the second case with respect to the single cycle, respectively. The electrical power, in the case of the binary cycles, increases of about 4.7 kW.

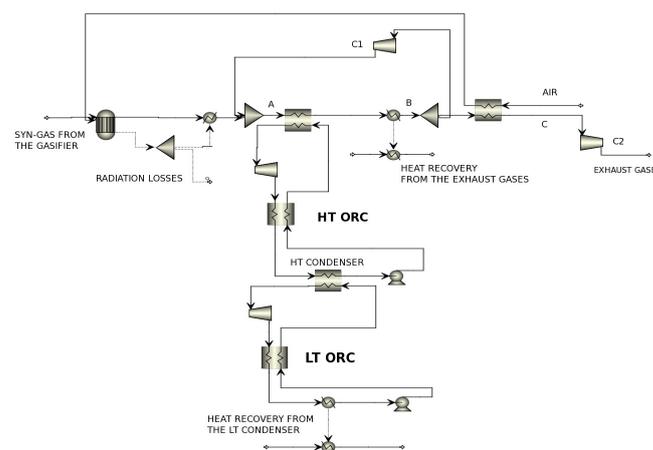


Figure 6. Plant scheme with a binary cycle configuration.

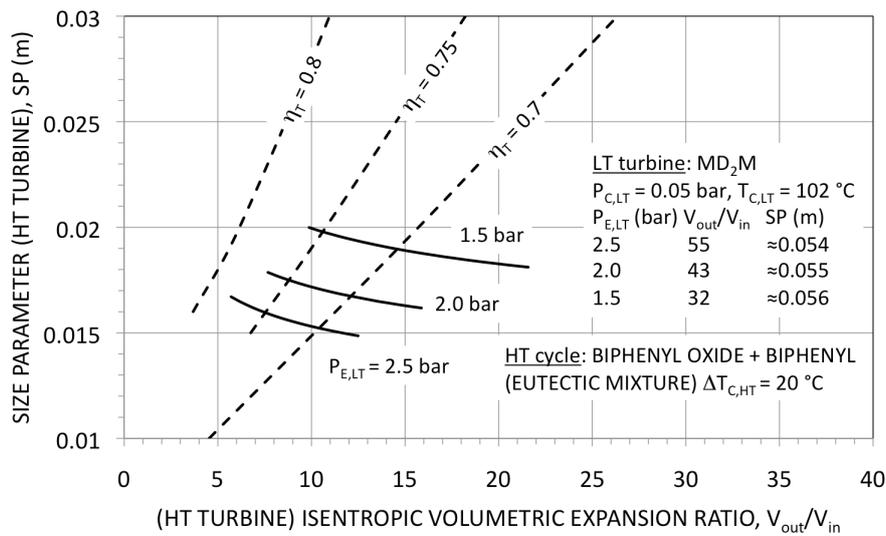


Figure 7. Size parameter and isentropic volumetric expansion ratio for the “High Temperature” turbine at different evaporation pressures $P_{E,LT}$. The working fluid for the HT cycle is the eutectic mixture of biphenyl oxide and biphenyl. The solid lines are at different fixed evaporation pressures $P_{E,LT} = 1.5$ bar, 2 bar and 2.5 bar. Increasing the ratio $\dot{V}_{out}/\dot{V}_{in}$ increases then the evaporation pressure of the HT cycle.

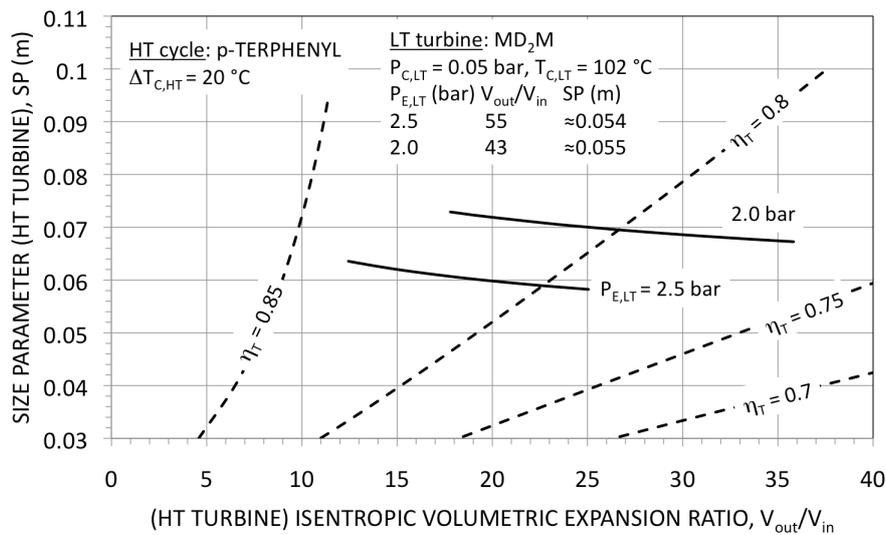


Figure 8. Size parameter and isentropic volumetric expansion ratio for the “High Temperature” turbine at two different evaporation pressures $P_{E,LT}$. The working fluid for the HT cycle is p-terphenyl. The solid lines are at two different fixed evaporation pressures $P_{E,LT} = 2$ bar and 2.5 bar. Increasing the ratio $\dot{V}_{out}/\dot{V}_{in}$ increases then the evaporation pressure of the HT cycle.

Although with the assumed calculation hypotheses, the global performances of the two considered binary cycles are quite similar, the turbines of the two HT modules results are very different. For its limited Size Parameter SP (see Figure 7) the turbine with the eutectic mixture requires a partial admission and a very small admission degree (see Table 5). Then its efficiency result could be considerably lower than the values here assumed.

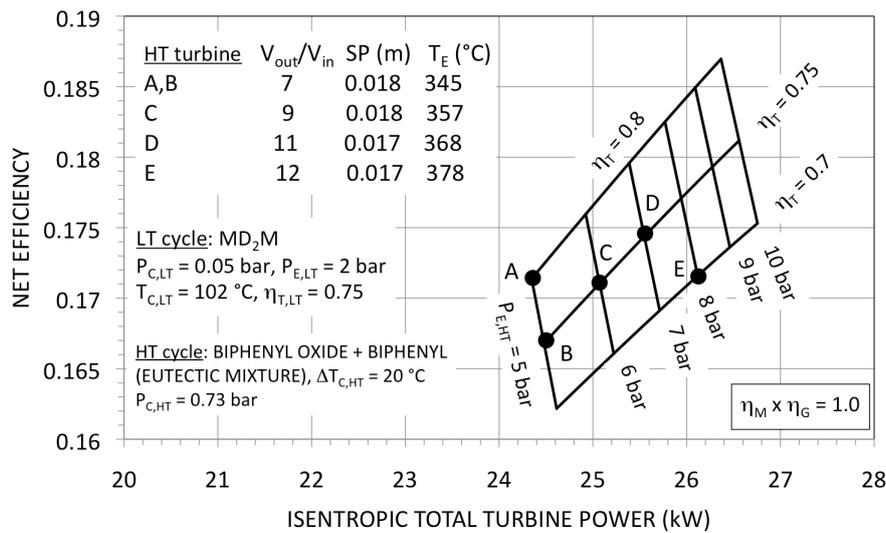


Figure 9. Net cycle efficiency and isentropic turbine power for binary cycles. Working fluids: MD₂M for the bottoming (LT) cycle; a eutectic mixture for the topping one.

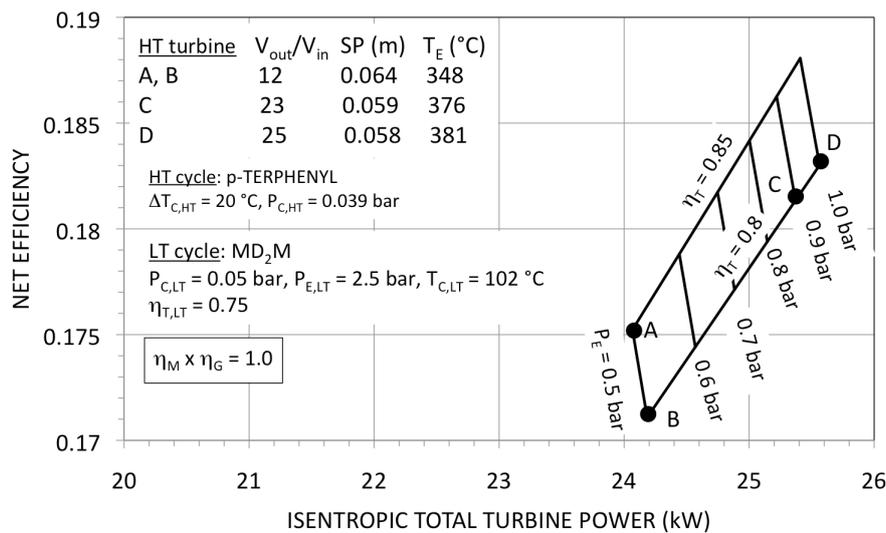


Figure 10. Net cycle efficiency and isentropic turbine power for binary cycles. Working fluids: MD₂M for the bottoming (LT) cycle; p-terphenyl for the topping (HT) one.

Table 3. Some results for a simple cycle. Working fluid MD₂M.

MD ₂ M - Single Cycle ^a	Result Value
Net efficiency	0.110
CHP efficiency	0.780
Electrical turbine power (kW)	12.724
Thermal power (kW)	75.028
Evaporation pressure (bar)	2.5
Condensation pressure (bar)	0.05
Cycle efficiencies	0.136
Mass flow rate in the turbine (kg s ⁻¹)	0.38229109
Turbine isentropic enthalpy drop (kJ kg ⁻¹)	49.310

^a η_T = 0.75, η_M × η_G = 0.90.

Table 4. Some results for binary cycles.

MD₂M (LT)/Terphenyl (HT) Cycle ^a	LT	HT
Net efficiency (LT + HT)	0.161	-
CHP efficiency (LT + HT)	0.884	-
Electrical turbine power (kW)	17.657	-
Thermal power (kW)	69.538	-
Evaporation pressure (bar)	2.5	0.9
Condensation pressure (bar)	0.05	0.039
Cycle efficiencies	0.144	0.119
Mass flow rate in the turbine (kg s ⁻¹)	0.27467604	0.16751581
Turbine isentropic enthalpy drop (kJ kg ⁻¹)	49.310	70.589
Electrical turbine power (kW)	9.143	8.514
MD₂M (LT)/Eutectic Mixture Biphenyl Oxide+Biphenyl (HT) Cycle ^b	LT	HT
Net efficiency (LT + HT)	0.154	-
CHP efficiency (LT + HT)	0.884	-
Electrical turbine power (kW)	17.245	-
Thermal power (kW)	70.207673	-
Evaporation pressure (bar)	2.0	7.0
Condensation pressure (bar)	0.05	0.73
Cycle efficiencies	0.136	0.111
Mass flow rate in the turbine (kg s ⁻¹)	0.28603896	0.19022992
Turbine isentropic enthalpy drop (kJ kg ⁻¹)	45.817	65.408
Electrical turbine power (kW)	8.846	8.399

^a $\eta_{T,LT} = 0.75$, $\eta_{T,HT} = 0.80$, $\eta_M \times \eta_G = 0.90$; ^b $\eta_{T,LT} = 0.75$, $\eta_{T,HT} = 0.75$, $\eta_M \times \eta_G = 0.90$.

Table 5. Some calculated approximate geometrical turbine features for the binary cycles.

MD₂M (LT)/Terphenyl (HT) Cycle	LT	HT
Rotational speed (rpm)	20000	20000
Nozzle exit flow angle (°)	15	15
Reaction degree at mean radius	0.2	0.15
Mean radius (m)	0.0839	0.0975
Mean rotor blade height (m)	0.0129	0.0135
Blade height on mean diameter ratio	0.0767	0.0694
Mean peripheral velocity (m s ⁻¹)	176	204
Mach number at the nozzle discharge (mean radius)	2.56	2.39
Stage inlet volume flow rate (m ³ s ⁻¹)	0.0126	0.0424
Rotor inlet volume flow rate (m ³ s ⁻¹)	0.324	0.594
Rotor outlet volume flow rate (m ³ s ⁻¹)	0.698	0.951
Stage Specific Speed	0.506	0.449
Stage Specific Diameter	3.037	3.313
MD₂M (LT)/Eutectic Mixture Biphenyl Oxide+Biphenyl (HT) Cycle	LT	HT
Rotational speed (rpm)	20000	20000
Nozzle exit flow angle (°)	15	18
Reaction degree at mean radius	0.2	0.0
Admission degree ^a	1.0	0.1
Mean radius (m)	0.0808	0.0866
Mean rotor blade height (m)	0.0143	0.0120
Blade height on mean diameter ratio	0.0882	0.0693
Mean peripheral velocity (m s ⁻¹)	169	181
Mach number at the nozzle discharge (mean radius)	2.49	2.20
Stage inlet volume flow rate (m ³ s ⁻¹)	0.0164	0.00725
Rotor inlet volume flow rate (m ³ s ⁻¹)	0.343	0.0771
Rotor outlet volume flow rate (m ³ s ⁻¹)	0.708	0.0771
Stage Specific Speed	0.539	0.131
Stage Specific Diameter	2.85	10.25

^a the ratio between the admitted inlet arc length and the total inlet arc length.

4. Conclusions

From the analysis and from the results presented and discussed in the previous sections, it follows that the design of truly efficient small high temperature Rankine engines is really very difficult, as one has to reconcile high efficiencies with low cost and demanding technological constraints.

In conclusion:

- Rankine cycles at high temperatures and small useful power require in general very high volumetric expansion ratios that, combined with small Sizes Parameters, usually demands: (i) axial multi-stages turbines with (ii) a high rotating speed, that is, very expensive turbomachines with rather low internal efficiencies. Assuming one-stage turbines the worked out expansion ratios are limited and the resulting thermodynamic cycle efficiencies are modest. A possible way to solve the problem is to resort to “standard” HT and LT Rankine modules and to combine them properly, achieving binary cycles.
- A significant contribution to the final cost of the engine is certainly due to the heat exchangers. The surface of all the necessary heat exchangers can be considered proportional to the sum $\Sigma(UA)_i$: the sum of the total heat transfer coefficients times the heat transfer area, in which the index i represents the i -th heat exchanger. For the single cycle in Table 3, $\Sigma(UA)_i$ results 4.12 kW K^{-1} , while for the binary cycle with terphenyl (in Table 4), $\Sigma(UA)_i$ is 6.38 kW K^{-1} : 55% more, but against an efficiency and an electrical power 46% and 39% higher, respectively. In the case of the binary MD₂M (LT)/Terphenyl (HT) cycle, the (UA) of the HT condenser represents the 35 per cent of the $\Sigma(UA)_i$ and it could results so a demanding component to design.

Here we investigated the potential gains in efficiency and power resorting to binary modules each one with single axial turbines, in comparison with simple cycles, for high temperature and small size power Rankine cycles. An in-depth economic analysis - being very difficult without detailed calculations and design and off-design evaluation - is outside the aims of this work. However, the complexity and (probably) an higher maintenance needs for the binary systems must be cited and should be taken in account.

In principle, the design of axial turbines could be easily adapted to any expansion ratio and to any power size. The machine does not require lubrication (and this is an advantage: the oil mixed with the working fluid can produce thermal degradation at high temperature) and even single stages, as discussed in the previous Sections, for the power sizes here considered, can have good efficiencies. Therefore, in the opinion of the authors, small dynamic expanders are generally preferable to volumetric expanders.

The considered reference syn-gas flow rate, from a small commercial gasifier, was selected with the intent of exploring the lower power limits. Increasing the available thermal power, the “scale effects” offer advantage to costs and to efficiencies. The coupling of ORC modules and small biomass stoves could be, strictly from the thermodynamic point of view, certainly less problematic, but the efficiencies will result lower.

Lastly, the problem of the choice of the right working fluids should be mentioned. Steam for small power sizes cannot be used. However, if one wants to make the most of the high temperature sources, the thermal stability limits the choice of the available organic fluids.

In the case of external combustion of syn-gas, the obtainable CHP efficiencies (of about 80–90%) are in agreement with those of the conventional ICE of similar mechanical power. However the electrical efficiencies result lower (about 10–15% against 15–25%).

On the credit side, the small Rankine engines with organic fluids can convert heat from any source and the development of small HT and LT modules could be interesting in many different applications. From this point of view, the choice of single-stage turbines could favour the design of small ORC modules as simple as possible, efficient and best used in well-defined temperature ranges. Small modules are potentially combinable with each other or used separately, according to the requirements.

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Abbreviations

The following abbreviations, symbols and subscripts are used in this manuscript:

C_p	molar heat capacity at constant pressure, $\text{J mol}^{-1} \text{K}$
P	pressure, bar
P_C	condensation pressure, bar
P_E	evaporation pressure, bar
R	gas constant, $\text{J kg}^{-1} \text{K}^{-1}$
S	entropy, $\text{J kg}^{-1} \text{K}^{-1}$
T	temperature, $^{\circ}\text{C}$ or K
ΔT_{PHE}	minimum temperature difference in the “Primary Heat Exchanger”, $^{\circ}\text{C}$
ΔT_C	minimum temperature difference in the condenser, $^{\circ}\text{C}$
UA	total heat transfer coefficient times heat transfer area, W K^{-1}
η	cycle efficiency
η_P	pump efficiency
η_T	isentropic turbine efficiency
$\eta_M \times \eta_E$	mechanical and electrical efficiency (pump and turbine)
\dot{V}_{in}	isentropic volume flow at turbine inlet
\dot{V}_{out}	isentropic volume flow at turbine outlet
ω_S	specific speed ($2\pi N/60 \times \sqrt{\dot{V}_{out}/\Delta H^{3/4}}$)
D_S	specific diameter ($D \times \Delta H^{1/4}/\sqrt{\dot{V}_{out}}$)
HT	High Temperature cycle
LT	Low Temperature cycle
ORC	Organic Rankine Cycle
<i>in</i>	inlet
<i>out</i>	outlet
<i>cr</i>	property in critical conditions

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