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COMPARISON OF RANKINE CYCLES FOR MICRO-CHP GENERATION BASED ON INWARD FLOW RADIAL TURBINE OR SCROLL EXPANDER

Luca Casarsa
Dip. di Energetica Macchine,
University of Udine
Udine , Italy

Claudio Mucignat
Dip. di Energetica e Macchine,
University of Udine
Udine , Italy

Mauro Reini
Dip. di Ingegneria Meccanica,
Università di Trieste
Polo di Pordenone, Italy

ABSTRACT

This contribution aims to analyze micro-CHP units based on Rankine cycles. Two types of expander are considered: a small scale inward flow radial turbine and a volumetric scroll type expander. This latter, should allow to overcome the limitation imposed by a standard steam-turbine that arise when the required shaft-power is very low. Moreover, the scroll expander will also allow to easily treat wet steams, which must be avoided when considering a turbo-expander.

The final aim is to deduce which one of the two types of expander is more suitable, with a specified target performance and the availability of a certain hot source. In order to define the thermodynamic expansion process, the analysis uses a one-dimensional model of the radial turbine, previously developed by the authors, and of an estimation of the scroll expander efficiency. Also, the analysis is carried out for different working fluids, such as water, and two organic fluids, cyclohexane and toluene.

Through the discussion of the results, for a specified set of constraints (e.g. expander inlet temperature, temperature of condensation, expander geometrical parameters) it is possible to deduce important indications on the most suitable expander for a given cycle layout.

INTRODUCTION

The liberalization of the electric market, together with environmental policies, leads to a new perspective for power generation. Indeed, the diffused energy generation and cogeneration can yield to effective energy savings, thus reducing pollution and global warming. In this sense, micro Combined Heat and Power (micro-CHP) is believed to play an important role, because of its capacity of satisfying both thermal and electric demand at the same time.

The required power sizes are very small, therefore a new generation of devices is currently being developed and many different technologies may be suitable: internal and external combustion engines, fuel cells, micro gas turbines and Organic Rankine Cycles (O.R.C.). [1-4]. However, one of the most interesting option is to recover a certain amount of heat at a certain temperature otherwise dissipated, employing a bottoming cycle.

One classical example is to recover the sensible heat of the exhaust gasses from a micro gas turbine. In fact micro gas turbine energy conversion efficiency can be improved by raising the Turbine Inlet Temperature (TIT) or realizing a bottoming cycle using the turbine exhaust gases. While an higher TIT involves challenging technical problems, a bottoming Rankine cycle employs relatively simple and high reliable components. If, at the same time, a thermal demand has to be satisfied, energy losses are strongly reduced.

In the authors' opinion, the Rankine cycle with water steam or organic working fluids, will have fairly good efficiencies (with respect to the low thermal levels) with high reliability of the components employed. However, an issue about which expander may be more suitable arises if the power level is very low, as in micro-CHP devices. Indeed below a certain power level, the required mass flow may be too small for a dynamic machine, whereas the volumetric expansion ratio may be too high. Due to these limitations a volumetric expander should be employed. One of the last and more interesting options is surely the adoption of a scroll-type expander, which should guarantee fairly good efficiencies and the capacity to realize also full wet expansions.

The feasibility of the scroll configuration is currently under the attention of the scientific community. Kim et al. [5] developed a prototype and a model of a 15 kW water steam Rankine unity equipped with scroll expander, which isentropic efficiency was measured to be 34%. Peterson et al. [6] realized a small scale regenerative unity with a system efficiency of 7.2%, with the expander's isentropic efficiency varying in the range from 45 to 50%. Best results were obtained by Lemort et al. [7], whose expander isentropic efficiency was 68%, with HCFC-123 as working fluid, and by Kane et al. [8]. The tests performed on their Hybrid Solar power system showed a maximum isentropic efficiency of the two scroll expanders of 68%. At the present, anyway, data regarding the delivered power and efficiency with variable vapor conditions are unavailable.

The aim of the present work is to obtain important indications on the performances delivered by a small scale cogenerative unit based on a Rankine cycle that operates with water or organic fluids and that employs two types of expander: an inward flow radial turbine or a scroll volumetric expander. The peculiarities and the technical limitations associated with both expanders are highlighted and the differences in the working conditions, depending on the thermal input, are put in evidence.

NOMENCLATURE

b	channel width [m]
C	absolute velocity [m/s]
DT_{pp}	temperature difference at the pitch-point [°C]
L	rotor axial length [m]
P	expander power output [kW]
$\dot{Q}$	heat source power [kW]
r	radius [m]
R_{id}	isentropic degree of reaction
S_4	expander inlet entropy [kJ/kgK]
T_{eva}	evaporation temperature [°C]
T_c	condensation temperature [°C]
T_{HS}	temperature of the hot source [°C]
T_{CR}	working fluid critical temperature [°C]
$T_{g,out}$	exhaust gas temperature [°C]
U	blade speed [m/s]
V	charge volume of the scroll expander [cm^3]
W	relative velocity [m/s]
zzl	axial length and inlet rotor radius ratio
Δh	enthalpy drop [kJ/kg]
Δq	loss dimensionless term
Δr	radial distance [m]

greek letters

α	nozzle inclination [°]
θ	blade trailing edge angle [°]
η_{exp}	isentropic expansion efficiency
η_{id}	total to static isentropic efficiency
η_r	recovery efficiency
λ	mean exit and inlet rotor radius ratio
ν	hub and shroud exit radius ratio
ς_n	nozzle loss coefficient
ρ_v	volumetric expansion ratio
ω	rotational speed [rad/s]

Subscripts

0	nozzle inlet
1	rotor inlet
2	rotor outlet
cl	clearance
h	hub
is	isentropic
s	shroud
sf	skin friction

THERMODYNAMIC ANALYSIS

The present work considers the possibility of recovering the thermal power associated with two different hot sources such as:

- a *600°* hot stream of gases *(T_{HS}=600°C)*, which may represent the exhaust of a micro gas turbine (air as a perfect gas has been used in the calculations);
- a *300°C* flux of diathermic oil, as a result of a certain energy recovery.

The hot source thermal power was fixed at *30 kW* and, only for the case of the radial turbine, a value of *60 kW* was also considered.

Regarding the working fluid, the authors chose to evaluate the performances of the bottoming Rankine cycle employing three different working fluids:

- Water;
- Cylclohexane;
- Toluene.

The choice of using water as working fluid is justified by the effort of keeping the plant layout as simple as possible. Furthermore, choosing to operate the vapor expansion also through a volumetric machine which should tolerate high rates of liquid phase [5,7], wet expansion can be considered as an option.

The two selected organic fluids should guarantee fully dry expansions and satisfying mass flow rates associated to relatively small values of ρ_v and V, compared to water, so that small components feasibility is expected to improve.

Since this analysis refers to small-size CHP units, where cost and simplicity of the plant lay-out is one of the main concerns, only recovery cycles at single pressure level have been considered. The analysis of the performances is carried out assuming that the components are adiabatic and neglecting the heat-exchangers pressure drops and the work required by the pump. These assumptions simplify the thermodynamic cycle analysis and would affect only in a minor way the predicted performances, although they must be removed for an accurate design of the plant components. This last issue has not been considered in the present paper.

The performances of the recovery unit are evaluated for a specified set of parameters:

- Operational Parameters (working fluid, temperature of evaporation, temperature of condensation, expander's inlet entropy, heat source and heat source power);
- Functional Parameters (expander isentropic efficiency).

For a specified hot source, hot source temperature, working fluid, and temperature of condensation, all the possible cycles are calculated from their temperature of evaporation and expander's inlet entropy. Considering that the evaporation temperature represents the thermal level at which most of the energy is supplied to the Rankine cycle, it has been chose as a main functional parameter. Also, the expander's inlet entropy at the evaporation pressure level provides an indication of the vapor conditions, i.e. if the cycle is superheated or wet.

The evaporation temperature was varied from 100°C to 300°C for water, and from 100°C to 280°C for both the organic fluids. Clearly, for each fluid a different entropy interval has been used. Nevertheless, the chosen interval was wide enough to consider also superheated vapor conditions at the expander inlet.

The expander isentropic efficiency strongly affects the cycle layout on the T-S plane. For the case of the scroll expander, two values of η_{exp} have been chosen, namely 0.4 and 0.6, representing the limiting values of the range within the efficiency of a scroll expander should fall.

This assumption has been made with reference to the scientific literature [5-9]. It should be noticed that there is not many data available on the performances of the scroll-type expander. Also, it should be considered that the expander efficiency may vary significantly depending on vapor conditions and expander orbiting speed, as reported in [6]. Considering that no other specific data is available and according to the scope of the present work, the choice to limit the analysis to two extreme values of η_{exp} and to a constant orbiting speed of 3600 rpm seems to be the most reasonable one.

For the inward flow radial turbine, the total-to-static isentropic efficiency, η_{id}, has been computed by means of a turbine model previously developed by [10] and described in the next section.

Once all the cycles' characteristics points are defined in terms of temperature and specific entropy, the working fluid mass flow has to be calculated, in order to obtain the total power output. The mass flow basically depends on the enthalpy raise required during preheating, evaporation and eventually superheating. Furthermore, the thermal power of the hot source and the temperature at which it is available clearly affect the mass flow. Once a reference value of the ambient temperature has been defined, the hot source mass flow may be calculated. Then it is possible to solve the energy balance on the recuperator, and finally acknowledge the working fluid's mass flow. Constraints are the minimum value of the pitch point temperature difference between the hot stream and the working fluid (which value was fixed at DT_{pp} =10°C) and the minimum value of the hot stream's temperature at the exhaust (see Fig.1). These latter are fixed at the following sets of values depending on the considered hot source:

- hot gasses at T_{HS}=*600 °C*: the minimum exhaust gas temperature, $T_{g,out}$, is fixed at *70 °C*, *100 °C* or at least *20 °C* above the temperature of the fluid entering the pre-heater;
- diathermic oil at T_{HS}=*300 °C*: *20 °C* above the temperature of the fluid entering the pre-heater.

These constraints limit the solution of the recuperator energy balance, and consequently the cycle cogeneration performances, in different ways, as will be lately clarified in the results discussion.

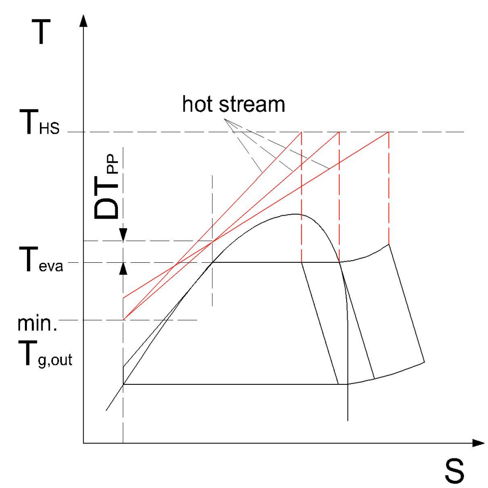

Figure 1. Rankine Cycle Layout and Hot Stream path for three different turbine inlet conditions.

The advantage in terms of cogeneration performance brought by a regenerated Rankine cycle, if feasible, has to be evaluated. Indeed, it is well known that dry fluids such as the organic ones here considered, pass through superheated conditions even during expansions from saturation. This behavior is even more clear due to the low isentropic efficiency of the scroll expander which may lead to superheated conditions also with wet vapor inlet conditions . However, in the case of water, the regeneration yield very little benefits with respect to the non regenerated case and, for this reason, this option has not been examined in this work. On the contrary, in the case of cyclohexane and toluene, the regeneration might be recommended to raise the unit performances in terms of power output. Nevertheless, the regenerative option must be carefully evaluated, since it yields to a higher exit temperature of the hot stream with a consequent decrease of the recovery efficiency. Assuming that the unit is adiabatic, the recovery efficiency corresponds to its cogeneration efficiency. It is clear that this parameter should be as high as possible, in order to exploit the heat source energy as much as possible.

The last thing to point out is that not all the calculated cycles are not technically feasible. Indeed, the volumetric expansion ratio and the expander charge volume represent the two main limitations imposed by a scroll expander. It might be underlined that also the expander inlet temperature may represents a problem for the thermal resistance of such machine. However, once the T_c and the maximum acceptable ρ_v are fixed, the resulting expander inlet temperature (the maximum expected) turns out to be always below about 250°C, certainly a safe operating temperature for such machines. For the case as for the inward flow radial turbine, constraints arise by the relatively low values of the working fluid mass flow rate, which requires too small rotor inlet blade height, as will be pointed out later on in the discussion.

ONE-DIMENSIONAL MODEL OF THE INWARD FLOW RADIAL TURBINE

Inward flow turbine are widely discussed in literature [11, 12], but their design usually refers to very hot gases as operating fluid and to the simultaneous presence of high thermal and mechanical stresses affecting rotor material, so that radial blades are generally considered as a standard. On the other hand, while in a gas turbine expander the inlet temperature is about 1000°C or more, the one of a micro-ORC-cycle will be in contact with a working fluid at about 250°C or less. In these conditions, thermal and mechanical stresses will not differ very much from those usually found in an outward flow air compressor of the same diameter and rotational speed. For this reason, some limitations usually affecting radial turbine design are not considered in this paper in order to improve the possibility to achieve feasible and efficient configurations.

The preliminary design of the inward flow turbine has been carried out on the basis of a simple one-dimensional approach developed by [10]. Working fluid composition as well as vapour pressure and temperature at nozzle inlet and condensing pressure (i.e. rotor outlet pressure) are preliminarily chosen. Rotational speed and rotor diameter must also be selected on the basis of maximum allowed centrifugal strength. Main design variables are then the mass flow rate and the isentropic degree of reaction.

Nozzle exit velocity, C_1, can be obtained with equation (1), where the nozzle loss coefficient is calculated with the Rodgers correlation [11].

$$\left(1 - R_{id}\right)\frac{\Delta h_{02is}}{C_1^2} = \sqrt{\frac{1}{1+\varsigma_n}} \qquad (1)$$

Flow is assumed to follow the free-vortex law in the radial distance, Δr, between nozzle exit and rotor inlet. Δr is obtained according to Watanabe [11]:

$$\Delta r / \left(b \cos\alpha\right) = 2 \qquad (2)$$

Rotor relative exit velocity can be calculated on the basis of rothalpy (i.e. rotational stagnation enthalpy) conservation with equation (2).

$$W_2 = \left(1 + \frac{\sum \Delta h_{losses}}{\frac{1}{2}W_{2is}^2}\right) \cdot \sqrt{W_1^2 + 2\Delta h_{12is} - (U_1^2 - U_2^2)} \qquad (3)$$

Enthalpy losses are computed as an addition of the various contributions given by equation (4), where dimensionless terms Δq refer to losses due to friction, clearance and secondary flows, respectively. For the latter loss contribution, the effects of curvature and blade loading are considered separately. The different terms in Eq. 4 are quantified using literature correlations [11].

$$\sum \Delta h_{losses} = \left(\Delta q_{sf} + \Delta q_{cl} + \Delta q_{curvature} + \Delta q_{blade}\right)U_1^2 \qquad (4)$$

As ς_n and Δq terms depend on geometry and flow parameters, equations (1-4) must be resolved with an iterative procedure. For this purpose, the Engineering Equation Solver (EES) software has been used [13].

In order to compute the velocity triangles, fluid properties and the rotor meridional shape must be defined. The first ones are given by the EES libraries while the last one is obtained by fixing the following four geometrical parameters:

- $\nu = r_{2h}/r_{2s}$ hub and shroud exit radius ratio;
- $\lambda = \overline{r_2}/r_1$ mean exit and inlet rotor radius ratio;
- $zzl = L/r_1$ axial length and inlet rotor radius ratio;
- θ angle of the blade trailing edge projection on the meridional plane, relative to the axial direction (0° = axial edge, minimum blade length, 90° = radial edge, maximum blade length).

The selected values of these latter parameters are reported in table 1. The condition $\theta = 90°$ has been imposed because it is representative of a commonly accepted rotor design choice [11, 12]. Moreover, in the previous work by Micheli and Reini [10] it has been shown that, the rotor efficiency obtained with $\theta = 90°$ is always close to the optimal rotor efficiency obtained with θ as a free variable, keeping other design parameter constant. In the same work further details about the model can be found. An example of the resulting rotor meridional shape is shown in Fig. 2.

Table 1: Parameters values assumed for the rotor design.

parameter	*value*
ν	*0.4*
λ	*0.5*
zzl	*0.7*
θ	*90°*

Blade angles are calculated on the basis of velocity triangles but, to obtain optimum efficiency at design point, an incidence factor, analogous to the slip factor used in centrifugal compressors, is taken into account at inlet. According to [11, 12] it is determined by means of the classic Stanitz correlation. It must be noted that blades are not necessarely considered radial at inlet, to improve the design degree of freedom. Such a possibility was taken into account considering that fluid temperature is not as high as in other and more common radial turbine applications, as small gas turbine and turbochargers, and consequently blade stresses are analogous to a compressor wheel ones, that often presents backswept blades. Moreover,

Table 2: model degrees of freedom.

parameter	*value*
organic fluid	Steam, ciclohexane, toluene
rotational speed ω	*3000-8000 rad/s*
T_{eva} Vaporizing temp.	*100 - 200 °C*
T_{sh} super heating temp.	*10 - 85 °C*
T_c condensing temp.	*50 - 90 °C*
mass flow rate	Depending on considered heat source
rotor inlet radius r_1	*40 - 60 mm*
R_{id} degree of reaction,	*0 - 1*

the usual design condition of having no swirl at outlet is here not imposed. This is due to combined geometrical and volume flow rate limitations that can make such a condition impossible to be fulfilled. Velocity triangle is simply defined to obtain a workable design condition that verifies all active constraints, or to obtain maximum work output, assuming some reduction in efficiency as acceptable [11].

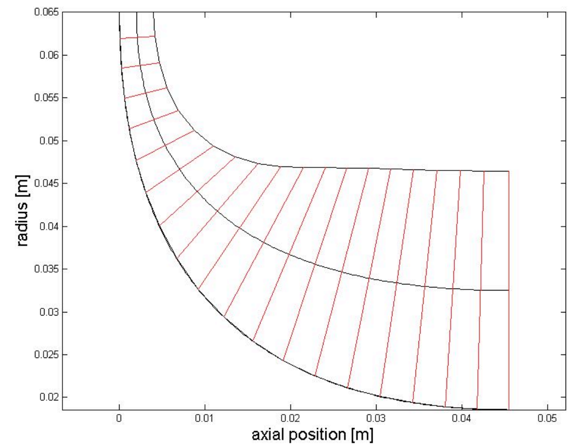

Figure 2. Example of rotor meridional shape

The described model presents eight degrees of freedom, as shown in table 2. In the following a fixed degree of reaction, $R_{id} = 0{,}35$ has been introduced becouse optimal degree of reaction generally does not allow power output increment greater than 5% [10].

RESULTS DISCUSSION: SCROLL EXPANDER

The results obtained using water as working fluid are presented first. Their analysis allows to highlight the effect on the cycle performances of the selected operational and functional parameters.

Water steam Rankine cycles

In the case that a hot stream of gases is available, such as the exhaust of a micro gas turbine, the performances and the required expander characteristics (ρ_v and V) are shown in Fig. 3 for two different expansion efficiencies. The black solid line in the figure represents the locus of the saturated liquid and vapor. For this data set, the minimum allowable hot gasses exhaust temperature was fixed at *100 °C*.
It is clear that the power delivered by the expansion increases as the evaporation temperature increases, and as the vapor conditions tends to be superheated, as it is widely demonstrated by basic thermodynamic considerations. However, the highest power levels are not approachable, due to the limitations imposed by the expander. As shown in the figure, the required ρ_v raises drastically with the temperature of evaporation. But this is not the only restriction imposed, also the required fluid-dynamic charge volume limits the operational field of the plant: the values of V quickly decrease with the pressure raise, i.e. with the evaporation temperature raise. For the charging volume, the minimum value of 1 cm^3 is considered the maximum limit. Under that limit the machine becomes technically unaffordable.
The latter constraint can be overcame by reducing the rotational speed of the expander, but this could yield to a

decrease of the expansion efficiency as reported in [6]. Anyway, for the specific case in Fig. 3, the restrictions imposed by ρ_v are stronger.

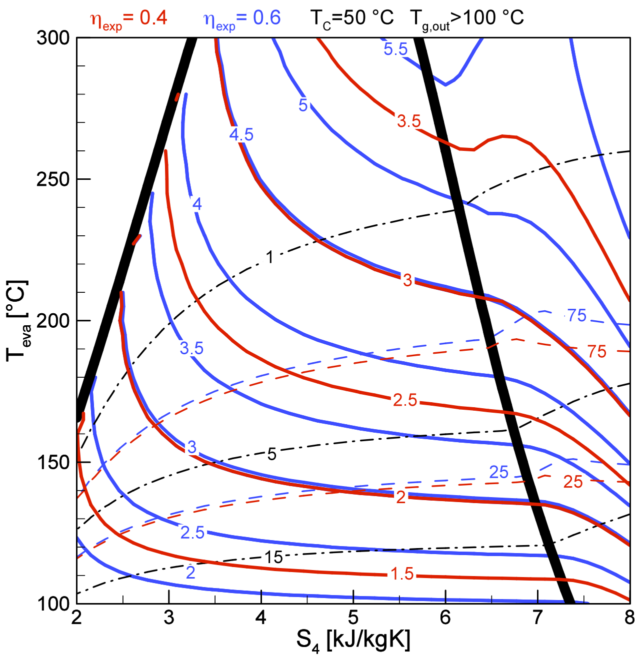

Figure 3. Cycle operational parameters and performances. Simple cycle. Working fluid: water. Hot source: hot gasses at T_{HS} =600 °C (solid lines: expander power output, dashed lines: ρ_v, dashed-dot lines: V).

Finally, it has to be observed that the expander must be at least a two stage one. Indeed, a single stage scroll expander has maximum expected ρ_v of approximately 5, while interesting possibilities are offered by the cycle lay-outs in Fig. 3 only for $\rho_v > 25$.

During the selection of the optimum operational parameters of a bottoming cycle, the recovery efficiency, i.e. the capacity of recovering the thermal energy available from the hot source, is very important, especially in cogeneration mode. In fact in cogeneration mode the recovery efficiency is equal to the first law efficiency of the whole unit since it is considered adiabatic. In Fig. 4 the cycle efficiencies for the case of T_c=50°C, η_{exp}=0.4 are reported. In the figure, two series of data are presented, which refer to the possibilities to cool down the hot stream at the temperatures of *70* and *100 °C*. For both situations, since T_{HS} is much higher than the water T_{CR}, the recovery efficiency is satisfying over a wide region of operational parameters. Superheating the working fluid yields to a decrease of the recovery efficiency, due to the augmentation of $T_{g,out}$ that is necessary in order to fulfill the conditions of minimum DT_{pp}, see Fig 1.

Raising the temperature of condensation, as could be required by a certain heat demand, leads to a lower cycle thermal efficiency and to a consequent reduction of the power output, as shown in Fig. 5. On the other hand, the maximum values of ρ_v and V are obtained at higher temperatures, consistently with the reduced pressure ratios of the cycle, so extending the region of the technically possible choices.

Figures 3 and 5 show also the influence of the expander's isentropic efficiency on the unit performances. From the lower value of *0.4* to the higher of *0.6*, an augmentation of about *1 kW* of expander power output is observed.

If the characteristics of the hot source are different, as in the case of diathermic oil, the performances and the required operational parameters change significantly. In fact, if the fluid T_{CR} is equal or higher than T_{HS}, a trade-off is to be taken into account while setting the optimum evaporating temperature, as demonstrated by the results given in Figs 6 and 7. For these data sets, the minimum allowable exhaust oil temperature was set *20°C* above the condensation one.

The delivered power output reaches zero for values of T_{eva} equal to the temperature of the hot stream. Again, the power output is zero for T_{eva} next to the condensation value, as a logical consequence. On the contrary varying the entropy at the expander inlet (S_4), i.e. changing the vapor conditions, the power output increases for superheated conditions (the cycle thermal efficiency is higher) and, at the same time, it raises for conditions close to the saturated liquid (the mass flow of working fluid is much higher). These observations prove the existence of a saddle point, which can be located around T_{eva}=170°C and S_4=6.5 kJ/kgK. This evidence is in accordance with the results obtained by Invernizzi et al. [14]. Limiting the analysis to superheated vapor conditions, they showed that if the difference between T_{HS}. and T_{CR}. is below about *80°C*, the optimum T_{eva}, in terms of power output, is approximately *130°C* below T_{HS}. Conversely, if the T_{CR} is much lower than T_{HS}, the optimum T_{eva} has to be found near the critical point. The same conclusions can be drown from the data in Figs 3 and 6, so extending the results by [14] also to the wet steam region. By comparison of the data in Figs 6 and 7 with those obtained exploiting hot gasses as thermal input, Figs 2 and 4, an overall decrease of the delivered power can be noticed, due to the lower thermal level of the hot source.

The influence of T_c on both ρ_v and V is the same as discussed previously. Moreover, the technically feasible cycles have an expander inlet temperature always below 250°C, certainly a safe operating temperature for the scroll machine.

Figure 8 reports the power output and recovery efficiency obtained for T_c=50°C, η_{exp}=0.4. The best heat source exploitation in cogeneration mode is obtained for moderate evaporation temperatures and small vapor fraction. This evidence, lessens the interest for the operational conditions characterized by superheated vapor and requires the use of a volumetric expander.

As already mentioned in the previous section, for the case of water the regeneration, which is feasible in some of the analyzed cases, does not bring any significant performances augmentation and, therefore, these results are not reported.

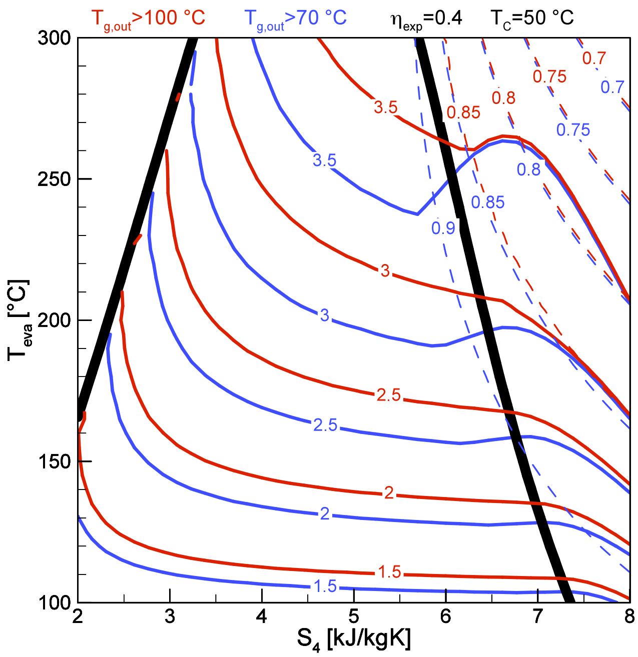

Figure 4: Cycle operational parameters and performances. Simple cycle. Working fluid: water. Hot source: hot gasses at T_{HS}=600 °C (solid lines: expander power output, dashed lines: η_r).

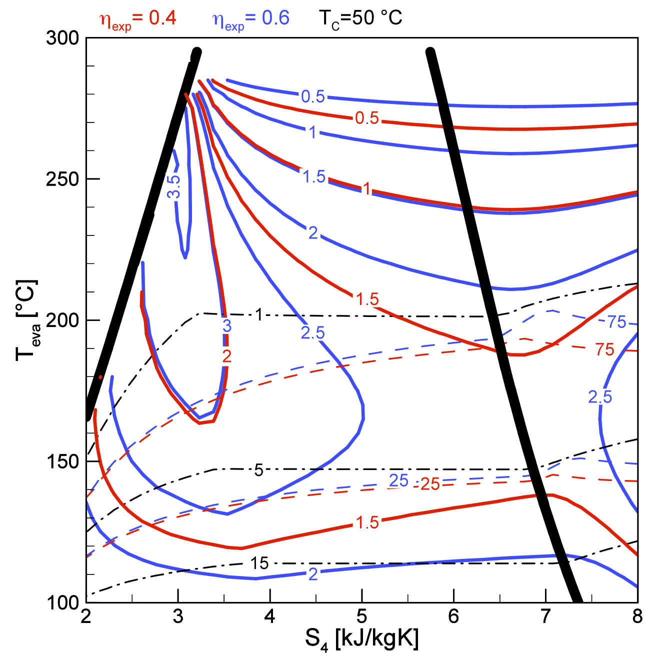

Figure 6: Cycle operational parameters and performances. Simple cycle. Working fluid: water. Hot source: diathermic oil T_{HS}=300 °C (solid lines: expander power output, dashed lines: ρ_v, dashed-dot lines: V).

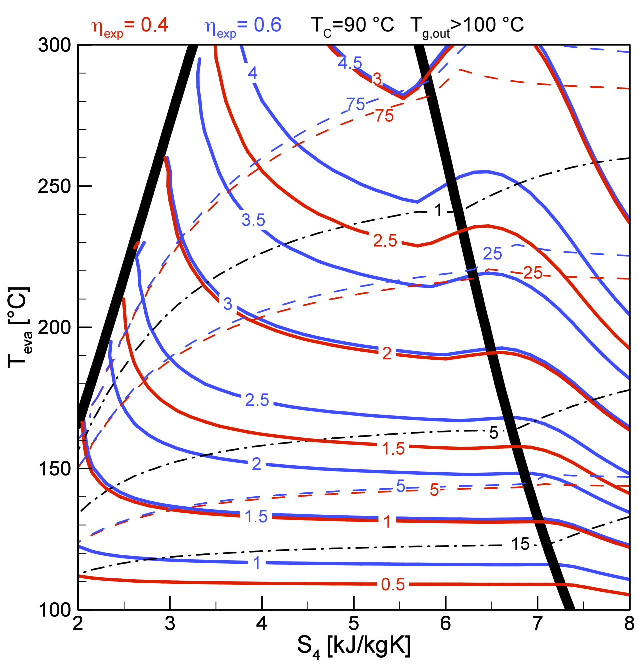

Figure 5: Cycle operational parameters and performances. Simple cycle. Working fluid: water. Hot source: hot gasses at T_{HS}=600 °C (solid lines: expander power output, dashed lines: ρ_v, dashed-dot lines: V).

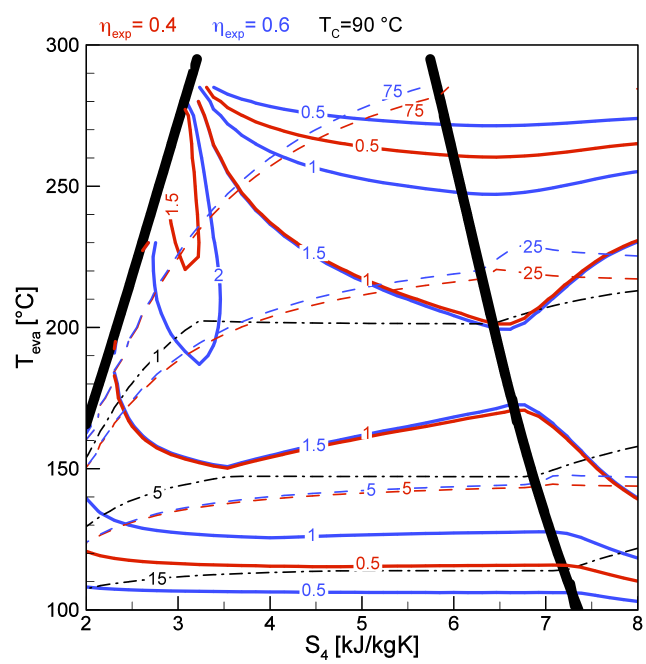

Figure 7: Cycle operational parameters and performances. Simple cycle. Working fluid: water. Hot source: diathermic oil at T_{HS}=300 °C (solid lines: expander power output, dashed lines: ρ_v, dashed-dot lines: V).

In conclusion, using water as working fluid, some interesting cycle lay-outs may be suitable for the cogenerative application here considered. All of them show a T_{eva} in the range *140-200°C* and require at least a two stage scroll expander. In particular, for the case of hot gases heat source, assuming to use a two stage scroll expander with $\rho_v=25$, the power output delivered is about *2 kW*, with $\eta_{exp}=0.4$, or 3 kW, with $\eta_{exp}=0.6$. This means a conversion efficiency (power output/heat source power) that varies from *0.06* to *0.10*. The related T_{eva} are *140°C* and *200°C* for $T_c=50°C$ and *90°C,* respectively. The performances for the case of diathermic oil are lower, due to the low temperature of the hot source. A maximum of *2kW* ($\eta_{exp}=0.4$) or *3kW* ($\eta_{exp}=0.6$) is achievable when $T_c=50°C$ only with full wet expansion. It has to be mentioned that some of this operational conditions are characterized by very low vapor fraction and might be unattainable even for a volumetric expander. Anyway, in this case, great care has to be taken when setting the optimum operating temperature because the recovery efficiency varies strongly with T_{eva} and S_4.

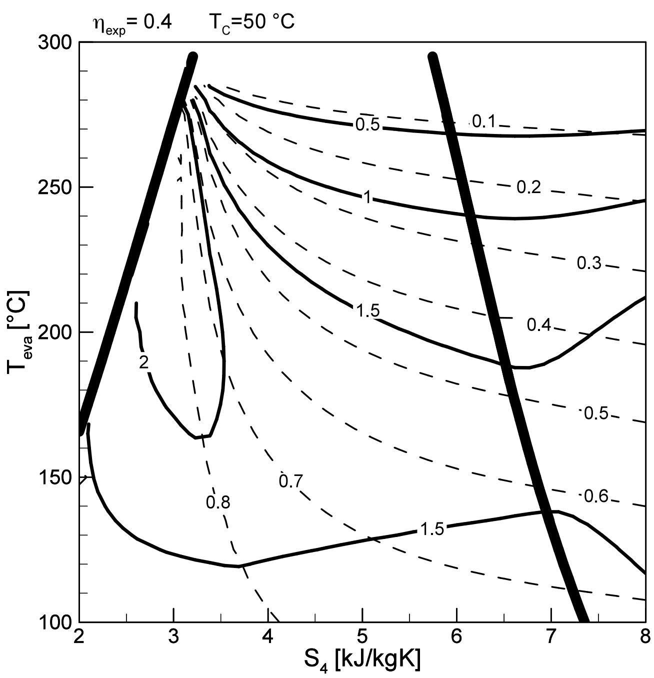

Figure 8: Cycle operational parameters and performances. Simple cycle. Working fluid: water. Hot source: diathermic oil at T_{HS}=300 °C (solid lines: expander power output, dashed lines: η_r).

Organic Rankine Cycles

In this section, only the data at $T_c=50°C$ will be discussed for both cyclohexane and toluene, since the influence of a higher T_c is the same as reported for water vapor.

For the hot gasses heat source, $T_c=50°C,$ and $\eta_{exp}=0.4\text{-}0.6$, the performances of the resulting ORC are reported in Figs 9 and 10. For these data sets, the cycles are computed without regeneration, which case will be addressed later on in the discussion.

Again, the maximum power output is obtained for evaporation temperatures approaching the T_{CR}, the latter being much lower than T_{HS}. However, as it was noticed for water vapor, the highest levels are not acceptable due to the expander's technical limitations. A comparison of the data obtained for the ORC with the performances of water vapor cycles shows approximately the same power output levels for the same values of ρ_v and η_{exp} but higher T_{eva} for the organic fluids. The paths of the iso-power lines suggest to keep the fluid near the condition of saturated vapor (Figs. 9 and 10), while in the case of water (Fig. 3) a certain superheating level leads to a more efficient operation, thanks to an increased cycle efficiency. On the contrary, using fluids such as cyclohexane or toluene, superheated conditions at the expander inlet yield to an even more higher grade of superheating at the outlet (referred to the pressure of condensation). The effect is a drop in the cycle thermal efficiency. Anyway, a certain grade of superheating at the expander outlet cannot be avoided, but this can be exploited to realize a regenerated Rankine cycle, improving the delivered power output. Figures 11 and 12 present the comparison of the data in Figs. 9 and 10 with those of regenerated cycles, obtained for $T_c=50°C,$ $\eta_{exp}=0.4$, and with minimum allowed outlet temperature of the hot stream set at *100°C* or *70°C*. Both the regenerated solutions show improved performances, with the best results for lower the exhaust gas temperature, as long as $T_{g,out}$ is different in the two conditions (after that point, the performances are obviously identical)

In the case of not regenerated cycle, the recovery efficiency is at its maximum value of about *0.85* for all the operational conditions (and for this reason is not reported in the Figs 11 and 12). For the two regenerated cases, the recovery efficiency drop quickly moving to superheated conditions, because of the consequent augmentation of $T_{g,out}$. Only in the region of wet vapors, a significant power output augmentation is associated to recovery efficiencies comparable to those of the not regenerated cycles.

If the thermal power exploited is associated to a flow of diathermic oil at $T_{HS}=300°C$ (Figs. 13 and 14), as seen previously for water cycles, the path of the power contours is justified by the comparable values of T_{CR} and T_{HS}. As reported by [14] for superheated vapor conditions, the optimum evaporation temperature is about *130°C* below T_{HS}, i.e at about *170 °C*. In addition, the present data show that, if wet vapor expansions are allowed as in this case, higher performances can be obtained by raising the evaporation temperature with respect to the one suggested in [14]. Anyway, at the highest power output levels the required ρ_v and V are technically not acceptable.

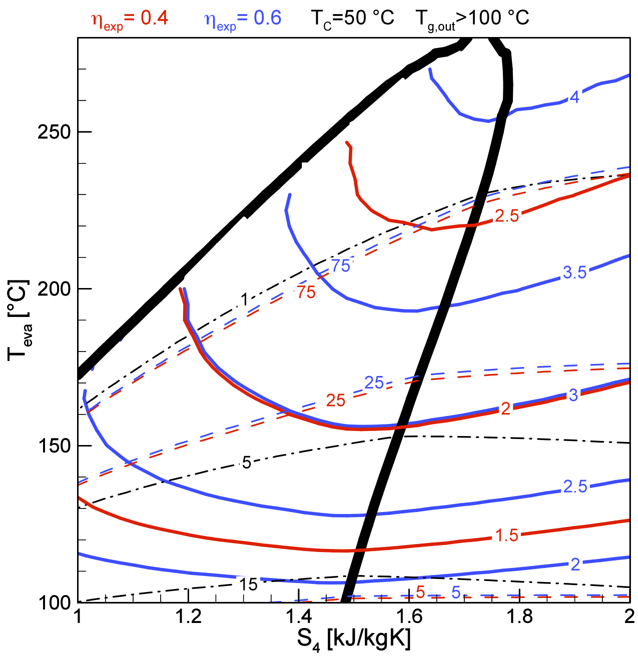

Figure 9: Cycle operational parameters and performances. Simple cycle. Working fluid: cyclohexane. Hot source: hot gasses at T_{HS}=600 °C (solid lines: expander power output, dashed lines: ρ_v, dashed-dot lines: *V*).

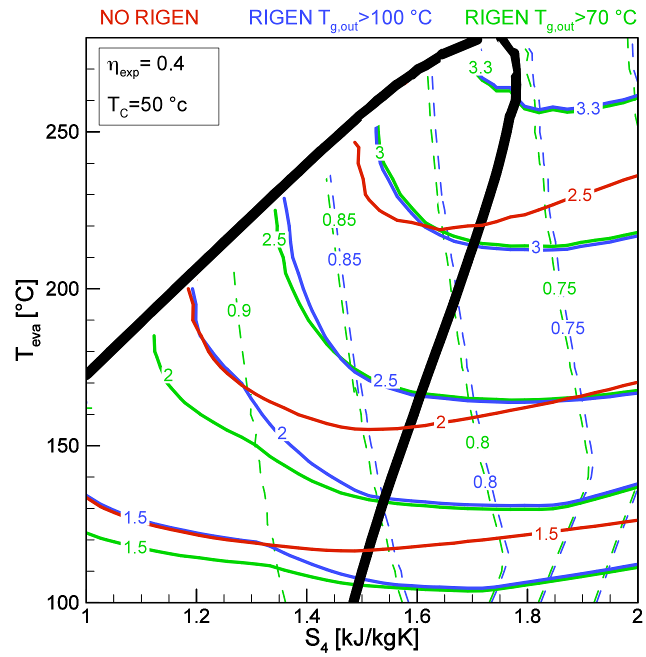

Figure 11: Cycle operational parameters and performances. Working fluid: cyclohexane. Hot source: hot gasses at T_{HS}=600 °C (solid lines: expander power output, dashed lines: η_r,).

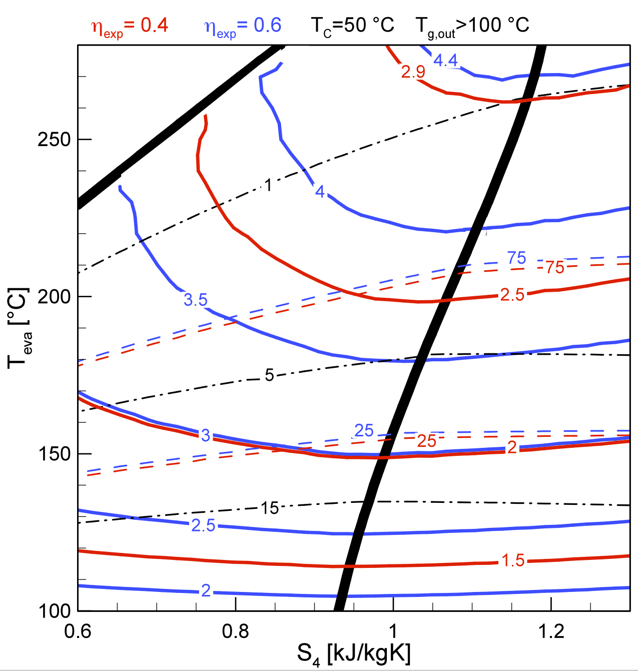

Figure 10: Cycle operational parameters and performances. Simple cycle. Working fluid: toluene. Hot source: hot gasses at T_{HS}=600 °C (solid lines: expander power output, dashed lines: ρ_v, dashed-dot lines: *V*).

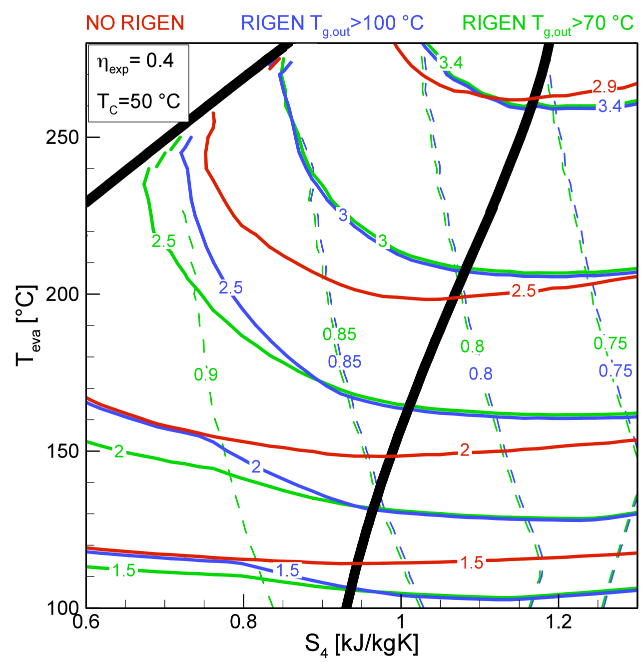

Figure 12: Cycle operational parameters and performances. Working fluid: toluene. Hot source: hot gasses at T_{HS}=600 °C (solid lines: expander power output, dashed lines: η_r,).

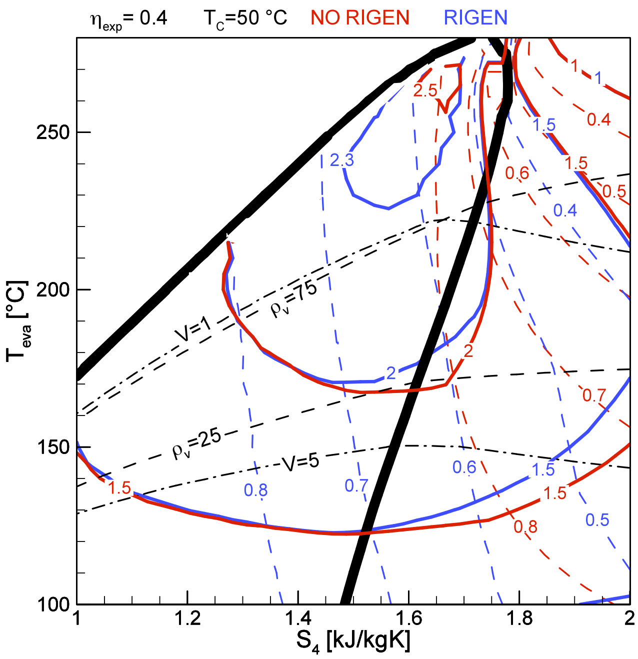

Figure 13: Cycle operational parameters and performances. Working fluid: cyclohexane. Hot source: diathermic oil at T_{HS}=300 °C (solid lines: expander power output, dashed lines: η_r,).

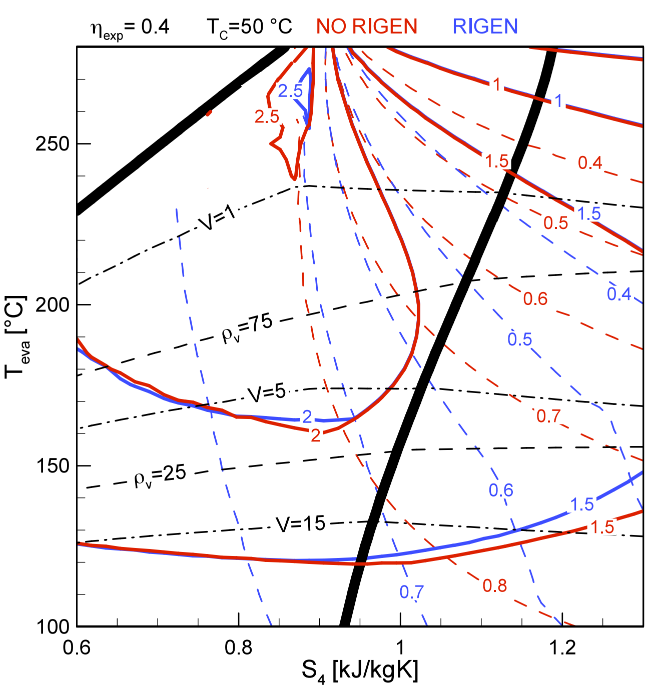

Figure 14: Cycle operational parameters and performances. Working fluid: toluene. Hot source: diathermic oil at T_{HS}=300 °C (solid lines: expander power output, dashed lines: η_r,).

Figures 13 and 14 also compare the performances of simple and regenerated cycles. In this cases, because of the moderate T_{HS}, the regeneration yields to a slight decrease of the power output due to a significant drop of the corresponding recovery efficiency. Indeed, with the realistic low values of η_{exp} here assumed, the thermal power from the regeneration is a consistent part of the power required by pre-heating, yielding to a higher $T_{g,out}$.

The data set for the case $\eta_{exp}=0.6$, show the same characteristics as for $\eta_{exp}=0.4$ with the exception of an obviously higher delivered power output.

In conclusion, the performances of the ORC showed power levels and operational conditions comparable with those obtained with water steam.

Only if the T_{HS} is sufficiently high, as in the present case of hot gasses at $T_{HS}=600$ °C, the regeneration is worth. However, the power output levels of the regenerated ORC are still comparable with those of water steam.

The only advantage brought by the employ of the ORC with respect the simple water steam cycles is that the same cogenerative performances can be achieved with higher vapor fraction ad lower ρ_v, i.e surely affordable by a scroll expander.

With respect to the cycloexhane, the toluene requires scroll expanders of bigger dimensions (see the V values in Figs. 9,10,13 and 14), which might be an advantage in some very low power applications.

RESULTS DISCUSSION: RADIAL TURBINE

In this case, the search of possible cycle layouts/turbine designs was obviously limited to super-heated conditions at the expander inlet.

Water steam Rankine cycles

From the calculations it turned out that proper values of the design variables are:

- a fixed level of super heating, namely *45 °C* above T_{eva}, sufficiently high to avoid wet expansions;
- rotor speed in the range *4000-8000 rad/s*;
- rotor radius, $r=45$ *mm*.

The results for the *30 kW* heat source of hot gases at T_{HS}=600 °C are provided in Fg.15. They have been obtained for $T_c=50°C$ and $T_{g,out}>100°C$. Unsatisfactory power outputs of about *1.5 kW* are achieved at $\omega = 4000$ *rad/s* as a consequence of the poor turbine efficiency, always below *0.4*. It is a matter of fact that such small turbines must have very high rotational speed in order to reach acceptable efficiencies [12]. Indeed, Fig 15 shows that for $\omega = 8000$ *rad/s* the turbine efficiency, and consequently the power output, are much higher. However, aside from the selected rotational speed, the required blade inlet height b is always comparable or much lower than *1 mm* which can be considered as the limit below which manufacturing constraint and serious fluid-dynamic limitations can arise. This last observation suggests that the choice of a dynamic expander to realize this low power micro-CHP unit is not the best one. Figure 15 shows also the expected power if a scroll expander with $\eta_{exp} = 0.6$ is used instead of the turbine. The data are extracted from those reported in Fig. 3 for *45 °C* of super heating and $\rho_v < 75$. From the comparison of the power curves for the two types of expander, and considering the limitations above mentioned about the turbine, the employ of a multi-stage scroll expander appears as the most reasonable choice, at least for a cycle thermal input of *30 kW*. On the opposite, if the available power at the hot source is higher, e.g. *60 kW*, the radial turbine becomes feasible, as shown by the values of *b* reported in Fig. 15, while the resulting mass flow rate could be too high in order to be efficiently discharged by a scroll type expander.

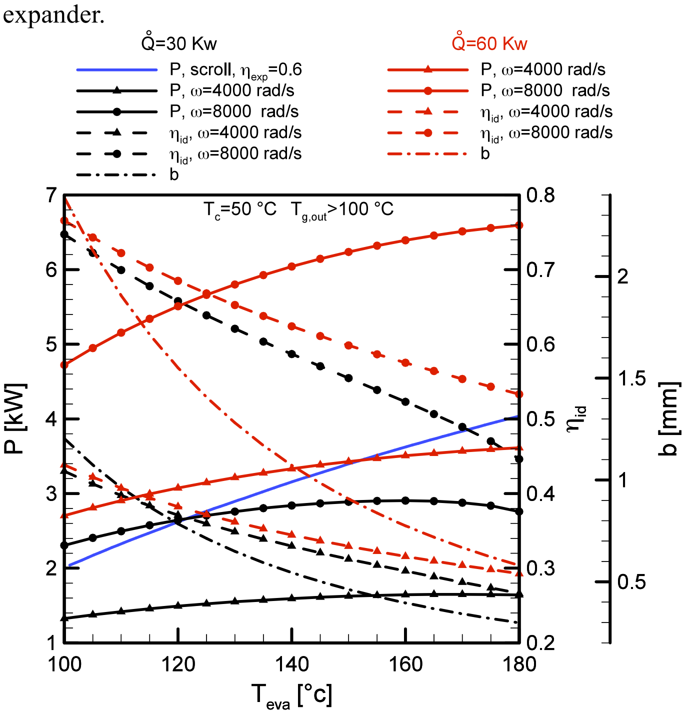

Figure 15: Performances, efficiency and characteristic rotor dimension for the turbine. Comparison with scroll expander at the same vapor inlet conditions. Simple cycle. Working fluid: water. Hot source: hot gasses at T_{HS}=600 °C.

Figure 15 shows also that the turbine efficiency decreases with T_{eva}. Indeed, to an increase of T_{eva} is associated an increase of ρ_v, which becomes more and more difficult to be performed with satisfactory efficiency by a dynamic machine with the geometrical configuration here considered. This is the reason why a maximum power output is found at $T_{eva}=170°C$, $\omega = 4000$ *rad/s* and *30 kW* as thermal input.

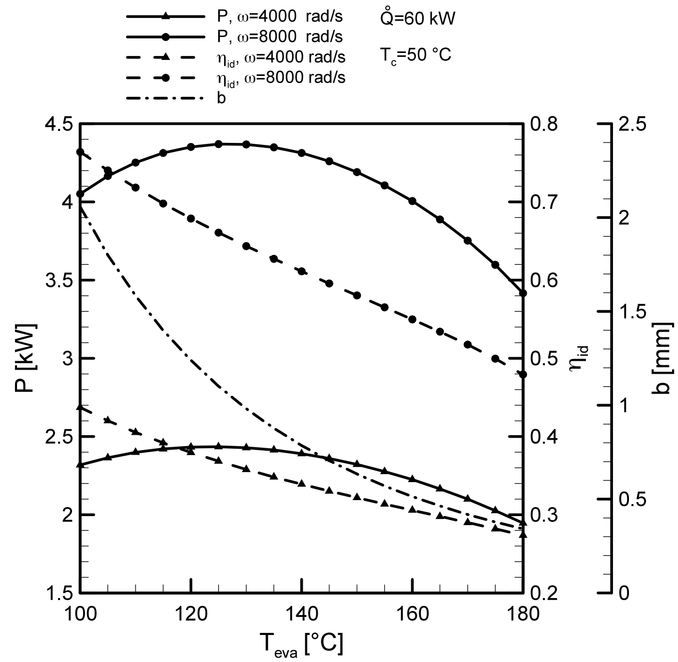

Figure 16: Performances, efficiency and characteristic rotor dimension for the turbine. Simple cycle. Working fluid: water. Hot source: diathermic oil at T_{HS}=300 °C.

Figure 16 show the calculated performances if the hot source of diathermic oil at T_{HS}*=300 °C* and *60 kW* is used. In the case of *30 kW* power input, the turbine design is critical (i.e. the machine turns out to be too small) due to the low power extracted from the hot source consequent to its moderate thermal level. Therefore the case at 30 kW loses interest and the results are not reported. In Fig. 16 the data suggest an optimum T_{eva} at about *130 °C*, which is under the value of *170 °C* suggested by [14]. The reason of this has to be found in the different values of the turbine expansion efficiency. Indeed, in the work by Invernizzi et al. [14] the turbine efficiency was kept constant at the value of *0.75*, differently from the present case where the efficiency varies significantly with T_{eva}.

Organic Rankine cycles

From the calculations it turned out that proper values of the design variables are:

- a fixed level of super heating, namely *10 °C* above T_{eva};
- rotor speed in the range *3000-5000 rad/s*;
- rotor radius, *r=40 mm* and *r=45 mm* for cycloexhane and toluene respectively.

The super-heating level is much lower than that one used for water steam (*45 °C*) basically for two reasons. First, with this type of fluids, dry expansions are always guaranteed as soon as saturated conditions are reached at the expander inlet. Second, an excess of super-heating leads to a reduction of both the cycle and recovery efficiencies, as commented when the results for the scroll expander were addressed.

The results for not regenerated cycles exploiting a *30-60 kW* heat source of hot gases at T_{HS}*=600 °C* are provided in Figs. 17 and 18.

Again, as it was for the water steam and for the same reasons there explained, the turbine performances are more satisfactory at the highest rotational speed of *5000 rad/s*, for both the organic fluids. Only for the cycloexhane at *30 kW* power input, the maximum power output is found for T_{eva}*=170 °C*, as a consequence of the unsatisfactory η_{id}. On the contrary, for the toluene, the power output is always increasing with T_{eva}. Indeed, the estimated η_{id} are higher for toluene than for cycloexhane (see Figs. 17 and 18), while in the case of water steam the values of η_{id} are the lowest obtained. Indeed, the working fluids mass flow for the ORC is much higher than those with water steam, because of the reduced enthalpy drop necessary for fluid pre-heating, vaporization and super heating. This condition is more favorable for the turbine which results of bigger dimension and more efficient. The values of *b* in Fig. 18 demonstrate that the turbine design working with toluene is acceptable in all the considered operating conditions, while for cycloexhane not all the results are acceptable (*b<1* for T_{eva}*>140 °C* and $\dot{Q}$*=30 kW*). In these cases, the scroll expander is a better choice.

If regeneration is taken into account, Figs. 19 and 20, the resulting performances are altered in the same way as commented when the scroll expander results were addressed (see the comment about Figs. 11 and 12). An overall power augmentation of about *15%* with respect to the not regenerated cycle performance are obtained when the minimum $T_{g,out}$ is fixed at *100 °C*. Lowering further the minimum $T_{g,out}$ at *70 °C*, has a positive effect only for T_{eve}*<140.* Further up of this evaporation temperature, the recovery efficiency is always far below the value reported for the simple cycle case (constant at about 0.85). The cycle operating conditions must be therefore carefully selected depending on the required unit characteristics, i.e. high delivered power and poor recovery efficiency or vice versa.

Figure 21 shows the results obtained for the hot source of diathermic oil at *300 °C*, cycloehxane and simple cycle. The regenerated option is not considered because it is associated to a diminished power output due to a poor recovery efficiency, as already commented for the data in Figs. 13 and 14. The power output increases until T_{eva}*=170 °C*, after this point, the condition on the minimum DT_{pp} is reached and the power starts to diminish consistently with a higher $T_{g,out}$ (see also the path of the recovery efficiency).

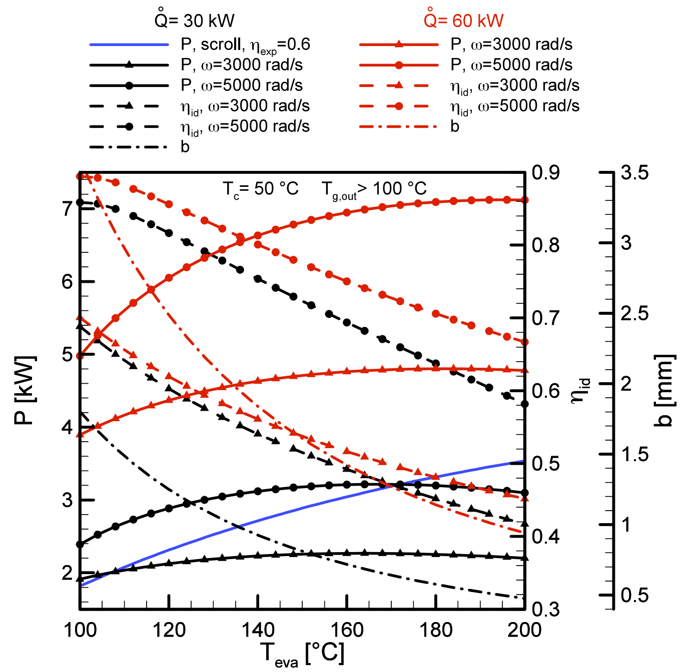

Figure 17: Performances, efficiency and characteristic rotor dimension for the turbine. Comparison with scroll expander at the same vapor inlet conditions. Simple cycle. Working fluid: cycloexhane. Hot source: hot gasses at T_{HS}=600 °C.

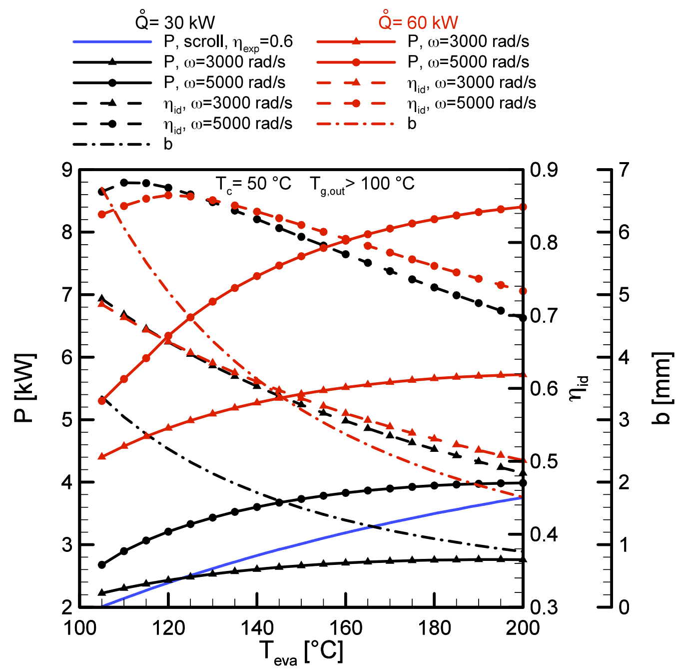

Figure 18: Performances, efficiency and characteristic rotor dimension for the turbine. Comparison with scroll expander at the same vapor inlet conditions. Simple cycle. Working fluid: toluene. Hot source: hot gasses at T_{HS}=600 °C.

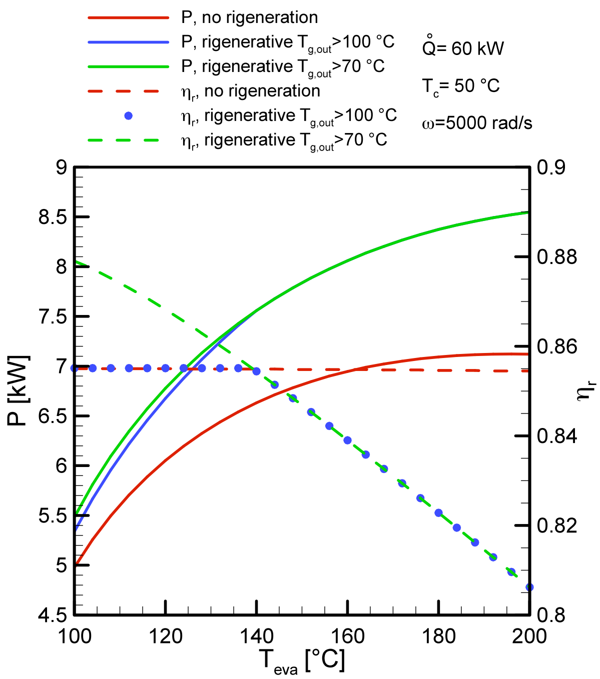

Figure 19: Comparison of simple and regenerative cycles. Working fluid: cycloexhane. Hot source: hot gasses at T_{HS}=600 °C.

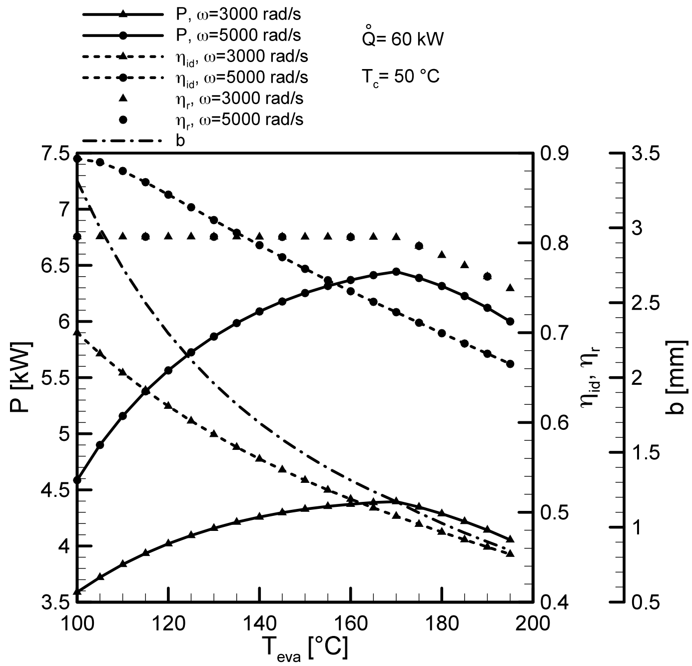

Figure 21: Performances, efficiency and characteristic rotor dimension for the inward flow turbine. Simple cycle. Working fluid: cycloexhane. Hot source: diathermic oil at T_{HS}=300 °C.

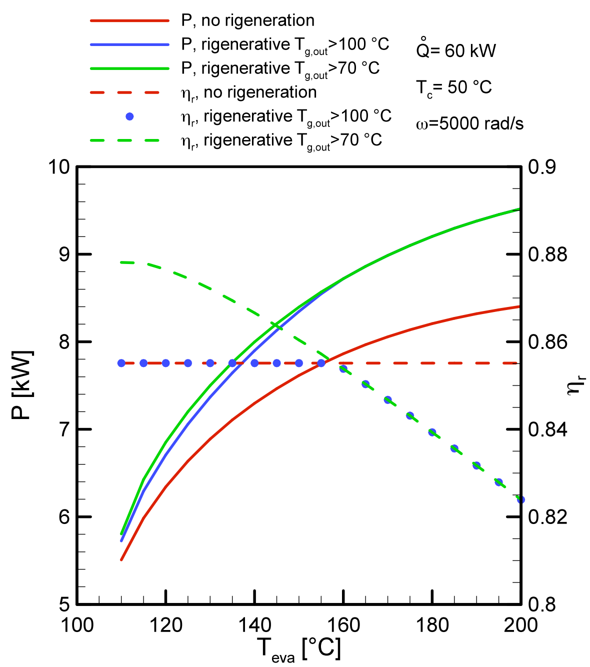

Figure 20: Comparison of simple and regenerative cycles. Working fluid: toluene. Hot source: hot gasses at T_{HS}=600 °C.

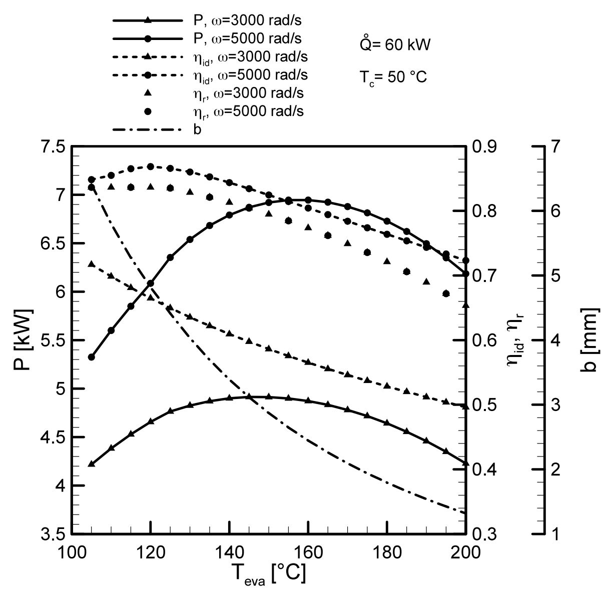

Figure 22: Performances, efficiency and characteristic rotor dimension for the inward flow turbine. Simple cycle. Working fluid: toluene. Hot source: diathermic oil at T_{HS}=300 °C.

For the toluene, Fig. 22, this condition is reached at *125°C*, but the power continues to rise until T_{eva}<*150 °C* thanks to the augmentation of the cycle efficiency and to a still good η_{id}. With respect to the data for the other hot source (Figs. 17 and 18), the turbine design is more difficult at the higher evaporation temperatures because of the reduced working fluid mass flow, consequent to the reduction of the recovery efficiency.

CONCLUSIONS

In this work, an assessment of the performance of micro-CHP unit has been carried out considering the possibility to operate a bottoming Rankine cycle with different hot sources, different working fluids and two types of expander. In particular, a volumetric scroll expander and an inward flow radial turbine have been considered as possible choices to realize the expansion of water, cycloexhane and toluene vapors. A one dimensional model based on losses correlations from literature has been used for a preliminary radial turbine design and expansion efficiency estimation. For the scroll machine the expansion efficiency has been assumed a priori equal to 0.4 or 0.6.

The performed analysis demonstrated the great difficulty to design the radial turbine if the thermal input is below about 30 kW, because of the too small working fluid mass flow rate that would require an excessively small turbine, unaffordable from both technical and fluid dynamic point of views. Only with toluene workable turbine designs have been achieved. On the opposite, the volumetric machines open the possibilities to explore region of operating conditions not possible with the dynamic expanders, i.e. full wet expansions. Indeed, it has been shown that if temperature of the heat source is comparable with the fluid critical temperature, interesting cogenerative performances are obtained by cycle layouts with expansions from wet conditions. In this case, the energy conversion efficiency is expected at the maximum of *8%* for water steam cycles, and about *7%* for the two organic fluids. In these latter cases, the cycle regeneration has not to be considered because it is not convenient in terms of performances and so an advantage for the cost and simplicity of the CHP unit is also obtained. If the hot source is available at a temperature above the fluid critical temperature the limitations imposed by the required volumetric expansion ratio and charging volume do not allow to reach the best possible performances. Nevertheless, the best possible cycles layouts allow energy conversion efficiencies of about *11%*, with slightly advantages for the water steam with respect to toluene and cycloexhane.

If the thermal input is higher, e.g. *60 kW*, interesting solutions are obtained using the radial turbine, especially with the organic fluids. Energy conversion efficiency of about *16%* have been reached with toluene operating in a regenerated cycle that exploit a hot stream of gasses at *600 °C*. As expected, the higher energy conversion efficiencies can be obtained only accepting a reduction of the recovery efficiency.

Increasing the condensation temperature from *50°C* to *90°C* leads to a power output penalization of about 3 percentage points.

In conclusion, this contribution showed that micro-CHP unit with thermal input below *30 kW* can hardly be realized with a traditional dynamic expander. The use of a volumetric machine such as the scroll one, opens new interesting scenarios to relatively simple and low temperature devices.

Further development of the present analysis would be possible only when reliable data about the efficiency of the scroll expander under different working conditions (e.g. type of fluid, vapor fraction) will be available. This goal can only be achieved by a detailed model of the expansion process in the machine, supported by an exhaustive experimental validation.

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