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Multiple Expansion ORC for Small Scale – Low Temperature Heat Recovery

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Abstract

ORCs are widely recognized as one of the most suitable solution for energy recovery, if the temperature of the heat source is of about 200°C, or lower. In case of heat sources of about 100 kW or smaller, the more common solutions prescribe a simple cycle and a single stage expander, in order to reduce complexity and costs. Scroll expanders, derived from scroll compressors, are expected to be available at very low unit costs. The drawbacks of this kind of solutions, originally designed for automotive, or HVAC applications, are mainly two: the low fixed volumetric expansion ratio and the small volumetric flow rate, that are not always well-suited for the requirements of power production. In this paper, different ORCs with multiple expansions are evaluated with the aim of achieving a better exploitation of small scale-low temperature waste heat sources. The comparison takes in consideration different possible solutions for the multiple expansions, with internally recuperated and not-recuperated cycles, whilst the data describing the actual behaviour of compressors derived scroll expanders have been previously obtained by a test rig, set up at the University of Trieste, using R245fa as working fluid.

Keywords: *ORC; scroll expander; heat recovery; low temperature applications.*

1. Aims and Scope

In recent years, the interest for thermal energy recovery has been notably growing. In this field, Organic Rankine Cycles (ORC) are widely recognized (AA. VV., 2016) as one of the most suitable solution for waste thermal energy recovery in form of electric power, in particular for cases in which the temperature of the heat source is about 200°C, or lower. As can be easily inferred, the number of potential applications rapidly increases as smaller and smaller energy recoveries are considered. In case of thermal heat sources of about 100 kWth or smaller, the more common solutions are based on a simple Rankine cycle, in order to reduce the complexity of the system, and one single stage expander, in order to reduce the cost of this component, which strongly affects the whole cost of the ORC system.

From this standpoint, possible solutions are scroll expanders directly derived from the inversion of scroll compressors, originally designed for the automotive, or HVAC market. In many cases they are available in a hermetic construction, which includes the electric generator, at very low unit costs. The scroll is a positive displacement machine essentially formed by two identical spiral-shaped wraps fixed on back plates. One wrap has a hole in the back plate and is held fixed, while the other can orbit. If the machine is used as an expander, the working fluid enters from the central chamber through the fixed back plate hole, and then it moves towards the external endings of the wraps and exits. The drawbacks of this kind of solution are mainly two: the low fixed volumetric expansion ratio and the small displacement volume, that are not always well-suited for the requirements of power producing ORCycles.

A collection of systematic data describing the actual behaviour of compressors derived scroll expanders has been acquired on the test rig installed at the University of Trieste (Clemente et al., 2013), (Bracco et al., 2013). Values of expansion ratio, net produced power and isentropic efficiency have been obtained by testing an expander with a displacement volume of 9.1 cm^3, using R245fa as working fluid. The tests have shown that the maximum overall isentropic efficiency (about 62%) is achieved for an expansion ratio of the scroll machine equal to about four. Such a value of efficiency can be also found in literature, as in (Zanelli & Favrat 1994), (Yanagisawa et al. 2001), (Lemort et al. 2006) and (Saitoh et al. 2007), although higher values (up to 68-70%) have been reported in (Lemort et al. 2006), (Saitoh et al. 2007), (Kane et al. 2003) and (Lemort et al. 2009). In many cases, however, the value of efficiency measured in actual operating conditions is too small to achieve a good performance of the power cycle.

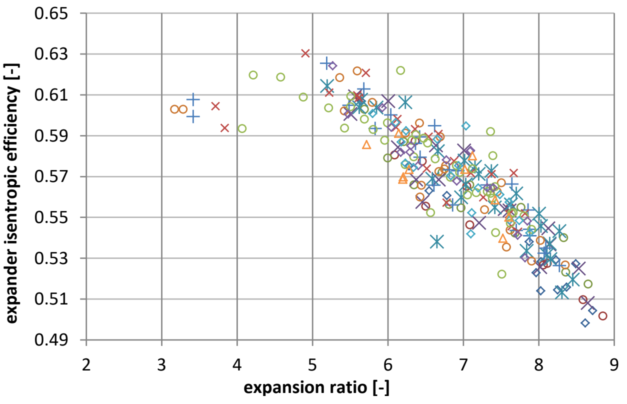

Figure 1. Isentropic efficiency vs. the expansion ratio at various rotational speeds (different markers: 5000-7750 rpm) for a scroll expander, with R245fa as working fluid. Experimental data obtained by the authors (AA. VV. 2016).

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In Figure 1, where the expander efficiency is shown as a function of the expansion ratio, it is possible to see that when high expansion ratios are imposed, according to the thermodynamic requirements of the cycle discussed in the next paragraph, the expander efficiency is much lower than its maximum value.

In the cited experimental tests, and in the related numerical simulations, the temperature of the heat source has been regarded to vary in the range 90°-150°C. This range can be typical for geothermal applications and medium temperature solar thermal collectors. In particular, the latter applications can be of interest if a solar cooling system is included as well. The coupling between a solar system and an ORC unit has been analyzed by several authors. For example Wang et al. (Wang et al. 2010a) built and tested an experimental small scale low temperature solar Rankine Cycle system featuring a direct evaporation of the working fluid R245fa. Two kinds of collectors were chosen for their installation: flat plate and evacuated tube types, respectively. In (Wang et al. 2010b) it is presented a small power ORC system, equipped with flat plate collectors, with the aim of comparing different working fluids, both pure substances and mixtures. A recent study by Freeman et al. (Freeman et al. 2015) discusses the technical and economic feasibility of a small scale Combined Solar Heat and Power (CSHP) system based on an ORC. Kane et al. (Kane et al. 2003) designed, built and optimized a mini-hybrid solar power system suitable for installation in remote areas of developing countries.

In all these low temperature applications, as their ideal Carnot efficiency is low, improving the effectiveness of the expansion process can be mandatory to reach profitable performance. In this regard, some papers in literature consider multiple-expanders ORC, in particular the configuration with two expanders in parallel, in order of increasing the flexibility of the system during variable load operation (Yun et al. 2015a), (Yun et al. 2015b). In (Rudenko et al. 2015) a super-critical cycle is considered with two expanders, but in this case they are small, custom designed, turbo-machines. Instead, two identical scroll expanders are considered in (Kaczmarczyk et al. 2015), both in series and in parallel, using HFE7100 as working fluid. Obviously, in the series configuration the two expanders have different rotational speeds.

In this paper, different ORCs with multiple scroll expanders are evaluated with the aim of achieving a better exploitation of small power low temperature waste heat sources. The expectation is that multiple expansion solutions allow to overcome some of the drawbacks of single scroll expanders. In particular, the aims are to maintain a high overall expansion isentropic efficiency and to increase the power production while keeping low the cost of the system.

The comparison takes in consideration the following different possible solutions for the expanders: a series of two stages, with one scroll in the first stage and two to five scrolls in parallel (or a single expander with the equivalent total displacement) in the second stage, with and without intermediate reheat. Both internally recuperated and not-recuperated cycles are taken into account, while R245fa is considered as the reference working fluid.

2. ORC Basic Model

On the basis of the experience gained from the previously mentioned experimental activity, a simple ORC model (either internally recuperated, or not) was implemented and validated. A design approach has been adopted, consistently with the purpose of the study, with the only exception of the scroll expander, which is considered to have defined dimensions and characteristics, as described by Clemente et al. (Clemente et al. 2013).

The scroll expander model is defined by the allowable volumetric flow rate, which is a function of its internal geometry and rotational speed, by the energy balance and by the isentropic efficiency vs. expansion ratio curve. This curve has been obtained interpolating the comprehensive set of experimental data shown in Figure 1. Condensing temperature, evaporating pressures, superheating temperature and operating fluid mass flow rate have been varied in order to analyze a wide range of operating conditions. In particular, the figure highlights the considered values of the rotational speed, which varied in the range 5000 – 7750 rpm. The expander was lubricated by mixing with the R245fa a mass fraction between 3% and 5% of a suitable synthetic oil (the lubricant must have good miscibility with the refrigerant in the whole required temperature range). As shown in the figure, the scroll characteristic curve can be regarded as almost independent by the thermodynamic cycle settings and, in particular, by the rotational speed of the expander, at least inside the operating condition range allowed for the tested component.

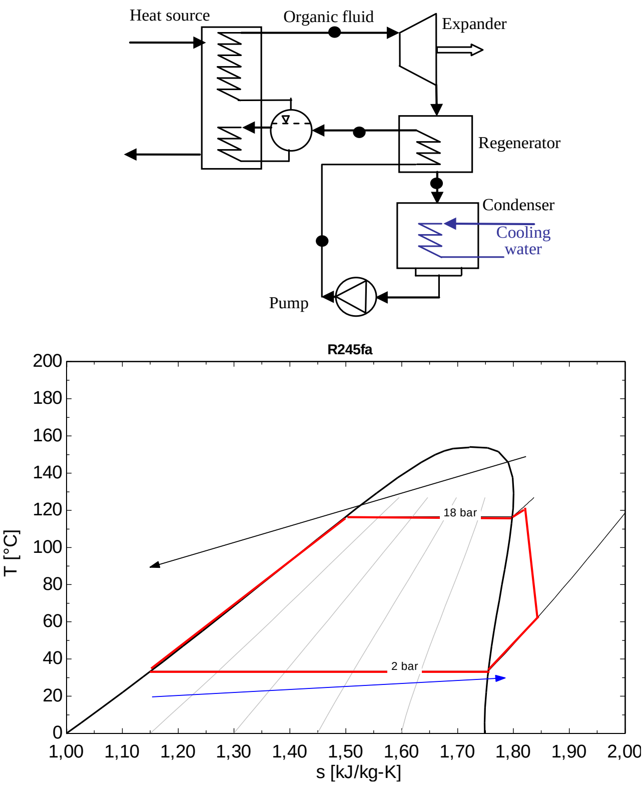

Figure 2. The basic ORC cycle (top) and its T-s diagram (bottom).

According to the design approach, all heat exchangers have been described by means of their energy balance. The minimum temperature differences at the evaporator and at the condenser have been assumed equal to 7°C and 6°C, respectively. These values are typical of the plate heat exchangers that can be used in this kind of application and consistent with experimental data. The effectiveness of the regenerator, which strongly affects the performance of the regenerated cycles, has been assumed as a constant value, equal to 0.80. This assumption is consistent with the experimental test results, which have given values from 0.80 to 0.83 with the considered plate heat exchanger.

A small degree of super heating, equal to 5°C, is introduced at the expander inlet with respect to the condition of saturated vapor. Pressure losses have been regarded as concentrated in the heat exchangers. They have been derived from the results of the experimental activity and assumed to be constant in the different configurations considered in the following. A value of 0.85 bar, comprehensive of all the losses of the circuit between pump and expander, has been assumed for the evaporator while 0.15 bar have been considered for both the sides of the regenerator.

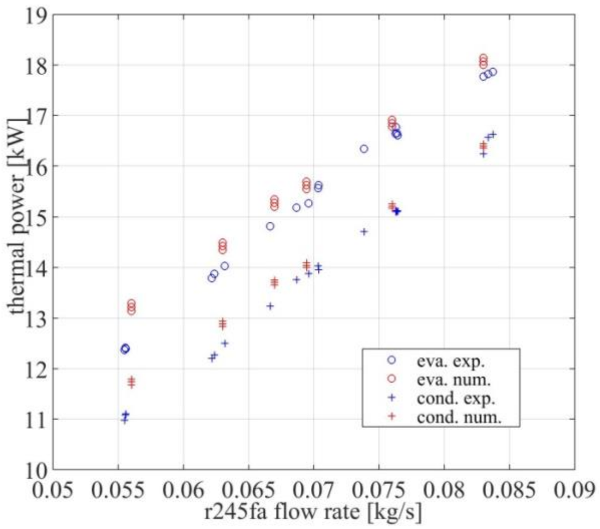

Figure 3a. A comparison between numerical evaluation and experimental results of the basic ORC cycle performance; heat exchanged in the evaporator and in the condenser vs. the working fluid mass flow rate.

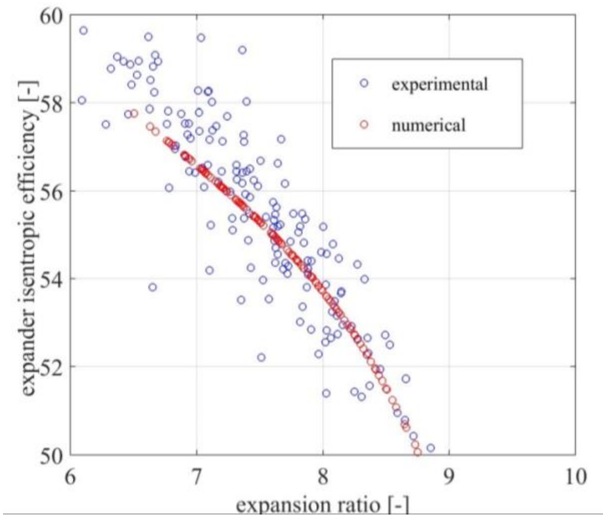

Figure 3b. A comparison between numerical evaluation and experimental results of the basic ORC cycle performance; efficiency vs. the expansion ratio of the expander.

The overall efficiency of the feed pump has been considered to be constant and the conservative value of 0.30 has been adopted, taking into account the experimental data and considering the adoption of a common low cost electric motor. The R245fa properties are calculated by means of the fluids library of the Engineering Equation Solver (EES) software.

The schematic of the components of the basic regenerated ORC cycle is shown in Figure 2.

The simplified ORC model has been validated by comparison with some experimental data, as can be inferred from Figure 3a and 3b. In the first one the thermal power exchanged with the hot and cold sources is reported at various working fluid mass flow rates while, in the second one, the experimental and numerical values of expander efficiency are compared as a function of expansion ratio. A 4th order polynomial relation of the following kind:

$$\eta_{isT} = a_0 + a_1\beta + a_2\beta^2 + a_3\beta^3 + a_4\beta^4 \quad (1)$$

has been used (a_0 = -22.5292, a_1 = 60.4716, a_2 = - 15.3566, a_3 = 1.6573, a_4 = - 0,0667).

In order to carry out the validation, the condensing temperature has been set to 27°C, corresponding to a condensing pressure of 1.6 bar. The evaporating pressure, instead, is determined by the chosen value of mass flow rate, given by the pump, and by the volumetric flow rate allowed by the expander at a given rotational speed. The obtained correspondence between numerical and experimental results has been considered acceptable for the purposes of a preliminary analysis. In order to define the uncertainty ranges of the two main global working parameters, i.e. the delivered electric power and the cycle efficiency, the theory of errors propagation has been applied on the data measured on the test facility. The results obtained from the different measurement points vary in a narrow range, so constant error values can be assumed. They are less than 5% for the electric power and 13% for the cycle efficiency (corresponding to less than one efficiency point when the cycle efficiency is 0.08).

The above described ORC model has been then used to carry out the analysis of multiple expansion arrangements. First, authors have quantified the theoretical possibility of improving performance with these kind of configurations. In Figure 4 the simulated efficiencies of the regenerative and non-regenerative cycle with the actual expander ($\eta_{cycle-Rig}$ and η_{cycle}) and with an ideal expander ($\eta_{ideal-Rig}$ and η_{ideal}) are shown as a function of the evaporation temperature, T_{ev}. The ideal expander has a constant isentropic efficiency equal to the maximum value measured with the actual one, equal to 62%.

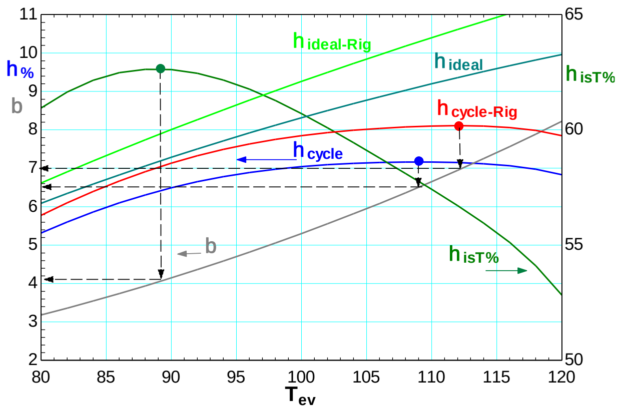

Figure 4. Isentropic efficiency of expander (right y-axis), efficiency of the cycle and expansion ratio (left y-axis) vs. the evaporation temperature, for the simple basic ORC cycle, in both regenerative and non-regenerative configurations.

Figure 4 shows also the curves of the simulated isentropic efficiency, η_{isT}, and of the pressure ratio, β, of the actual expander. The figure highlights that, while the maximum value of η_{isT} is reached for $\beta = 4$ and T_{ev} of about 90°C, the maximum values of $\eta_{cycle\text{-}Rig}$ and η_{cycle} are reached for $\beta = 7$ and evaporation temperatures of about 110°C-112°C, even if the corresponding value of η_{isT} is reduced of five percentage points with respect to its maximum. Instead, with the ideal expander, the cycle efficiency in the same operating point could be about two percentage points higher than that obtained with the actual one, and it could further increase at higher evaporating temperatures and pressure ratios.

In conclusion, it can be inferred by Figure 4 that significantly higher cycle efficiencies would be obtained if the expander were specifically designed for pressure ratios of about 7-9.

Unfortunately, such a kind of scroll expander is not available in the low cost component market, where components produced in very large series for automotive, or HVAC applications can be found. In addition, the working principle and the production technology of industrial scroll compressors are generally regarded as not proper for compression ratio higher than 6-7. The expectation is that a global efficiency improvement of about two percentage points could be obtained by adopting a multiple expansion configuration. With such a solution, each expander, of the same kind of the scroll considered in the basic (single stage) model, could operate with a pressure ratio and isentropic efficiency values closer to the maximum (design) ones.

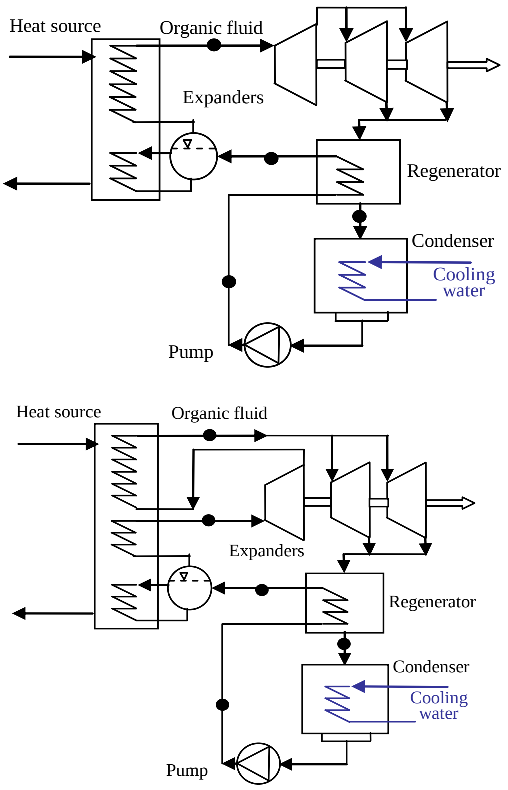

Figure 5. The ORC cycle with 1+2 series and parallel configuration, with RH and without RH.

In principle, the limitations related with built-in volumetric expansion ratio of a volumetric (scroll) expander could be overcome (to some extend) by changing the operating fluid. For instance, if isobutene was used instead of R245fa, a higher evaporation temperature would be achieved with the same built-in volumetric expansion ratio and condensation temperature, so higher cycle efficiency would be expected.

Alkanes have not been considered here because of their high flammability. But, even if isobutene (or a different fluid with similar characteristics) were used, the issue of further increasing the cycle efficiency and the power output could be addressed in a similar way by adopting multiple expansion configurations, the only difference lying in a higher level of the evaporation temperatures. Therefore, the conclusions inferred in the following for the multiple expansion ORCycles, adopting fixed size scroll expanders, can be regarded as quite general, and do not depend on the specific working fluid considered in the study (R245fa).

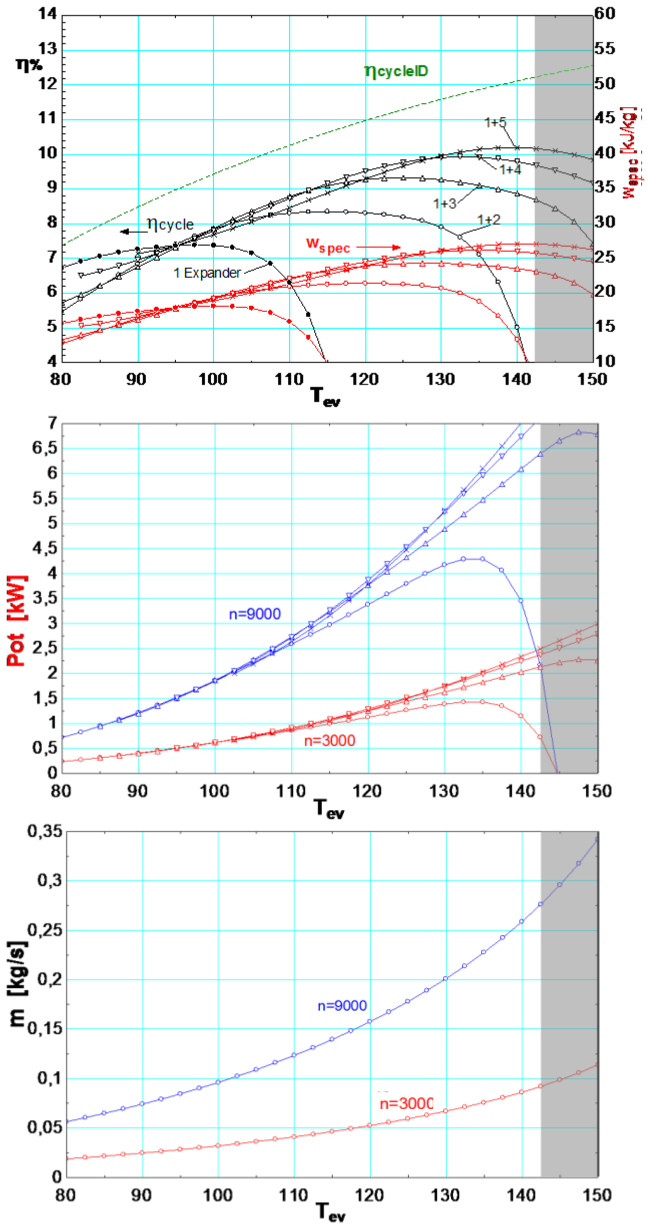

Figure 6. Electrical efficiency, specific work (w_{spec} - top), total power output (P_{ot} - centre) and working fluid mass flow rate (m - bottom) for different evaporation temperatures and different configurations of multi-expanders not recuperated ORC, with T_c=25°C. In grey area the conditions not consistent with the minimum pinch point in the evaporator (Δ_{pp}=7°C).

3. The Multiple Expansions ORC

In the following, different kinds of multiple expansions ORC are considered, by combining series and parallel configurations. In particular, a series of two stages, with one scroll in the first stage and 2 to 5 scrolls in the second one are considered, with and without re-heating. In Figure 5 the configurations 1+2 with and without re-heating are shown.

All the expanders are supposed to be equal to the scroll unit cited in section 1, and rotating at the same speed, i.e. on the same shaft, in order of allowing the adoption of a unique electric machine and a unique power-conditioning unit. Maintaining the same speed for all the expander should not penalize the cycle efficiency, due to the almost negligible influence of the scroll speed on its efficiency, as previously shown in Figure1.

The simple configurations of 1+1 identical expander in series and of two in parallel are not considered in this study. The latter, taking into account the design approach adopted in the thermodynamic model, is completely equivalent to consider a duplication of the whole basic ORC shown in Figure 2, therefore it cannot imply any advantages from the standpoints of the operating condition and of the efficiency of the scroll. In principle, the former (1+1) configuration could be adopted, but the two scrolls cannot be operated at the same rotational speed because of the increasing volumetric flow rate during the expansion (Kaczmarczyk et al. 2015). Thus, it could be necessary to adopt a gear reduction, or two separated electric machines and power conditioning units. In the multiple expansion cases, two values of the condensation temperature, T_c, have been considered: 25°C, excluding thermal cogeneration, and 35°C, consistent with low temperature space heating applications.

4. Simulation Model Results

The results obtained for the non-regenerative configurations, with T_c=25°C, are summarized in Figure 6. The electrical efficiency, the specific work, the power output and the working fluid mass flow rate are shown as a function of the evaporation temperature for different number of expanders in the second stage. The grey zones of the figure correspond to operating conditions not consistent with the constraint on the minimum value of the pinch point in the evaporator, Δpp=7°C.

Notice that the electrical efficiency and the specific work (in the upper part of the figure) do not depend on the rotational speed of the expanders, because the thermodynamic cycle in the chart (T-s) and the efficiency curve of each expander remain unchanged. The mass flow rate (in the lower part of the figure) increases with the rotational speed but, because of the fixed geometry of the expander in the first stage, it does not depend on the number of expanders of the second stage.

On the contrary, the net power output (P_{ot}, in Figure 6-centre) depend on the expander rotational speed. In the figure the results obtained for the maximum and minimum speed allowed for the tested expander are shown, equal to 9000 and 3000 rpm, respectively.

It can be seen that the maximum cycle efficiencies are higher than that of the basic cycle and are obtained at a higher evaporation temperature. This behavior is more evident by increasing the number of expanders in parallel in the second stage but, as it was expected, the improvement obtained with one additional expander is each time lower. In particular, with a configuration 1+4, an efficiency of 10% is reached with an evaporation temperature T_{ev}=130°C. In this condition, the power output is expected to be in the range 1.7 – 5.2 kW, depending on the rotational speed of the expanders. The corresponding working fluid mass flow rate is always lower than 0.2 kg/s.

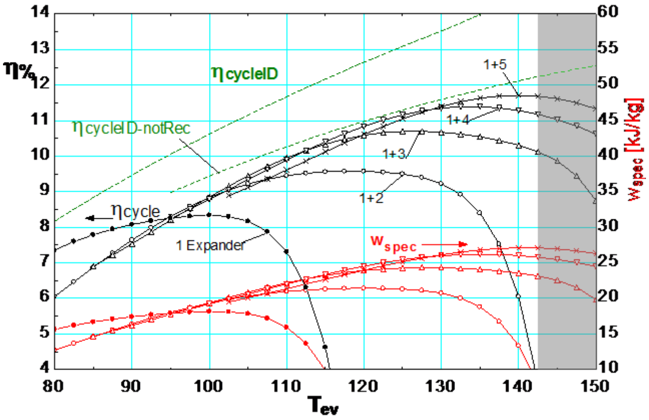

Figure 7. Electrical efficiency and specific work for different evaporation temperatures and different configurations of multi-expanders recuperated ORC, with T_c=25°C.

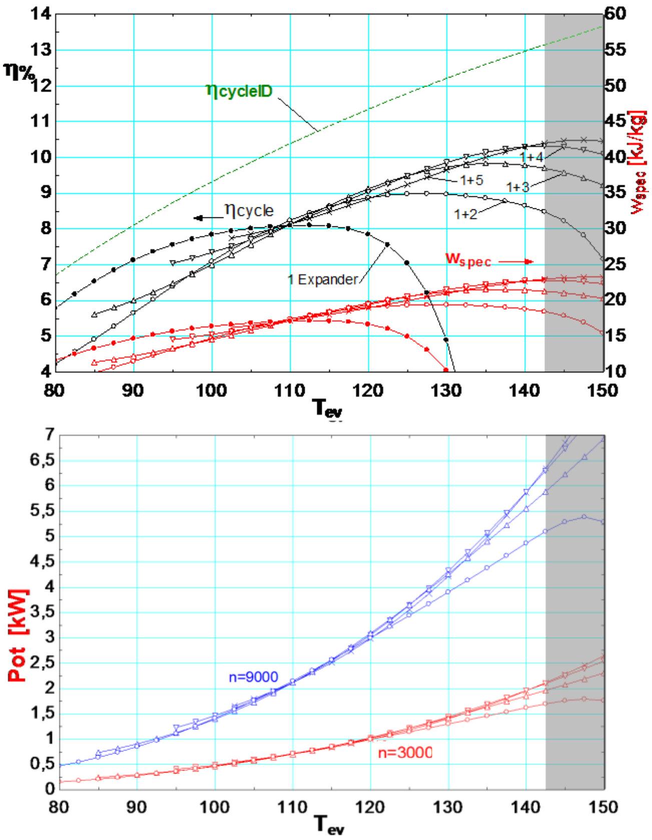

Figure 8. Electrical efficiency, specific work, power output for different evaporation temperatures and different configurations of multi-expanders recuperated ORC, with Tc=35°C.

Better performances can be obtained adopting a regenerated cycle, as summarized in Figure 7, where the electrical efficiency and the specific work are shown for the same operating conditions of previous Figure 6. Notice that the specific work, the power output and the working fluid mass flow rate are the same of the not recuperated case. The efficiency gain in the optimal conditions are generally of about one percentage point or greater, as it happens between the regenerated and non-regenerated basic cycle.

Notice that in all the considered cases, either regenerative or not, the maximum efficiencies and the maximum specific work are obtained for evaporation temperatures lower than 140°C, consistent with a temperature of the hot fluid in the evaporator of 150°C and the minimum pinch point, Δ_{pp}=7°C.

When the temperature of the heat source does not allow reaching an evaporation temperature higher than 95°C, the basic cycle obtains the same performance of the multiple expansion configurations, or even better. Therefore, multiple expansion configurations may be taken into account only for evaporation temperatures higher than 105°C, when they allow an efficiency gain of one percentage point, or more. The adoption of recuperated cycles is generally recommended, even with low evaporation temperatures, if consistent with the acceptable exploitation of the available hot source.

Figure 8 shows the electrical efficiency, the specific work and the power output for different evaporation temperatures and different configurations of multi-expanders recuperated ORC, as in Figure 6 and 7, but with a higher condensation temperature, T_c=35°C. Notice that the working fluid mass flow rate is the same of previous Figure 6, because of the fixed geometry of the volumetric scroll expander in the first stage. It can be inferred that, by increasing the condensation temperature, the maximum of all curves (both the efficiencies and the specific works ones) move to the right and become lower, as expected for a Rankine cycle. In particular, the maximum for the configurations 1+4 and 1+5 are inside the gray area of the diagram, highlighting that a heat source temperature higher than 150°C is required in order to allow these configurations operating in their optimum point. For the same reason, a maximum power output of about 5.5-6 kW can be obtained, lower than the values of about 6.5-7 kW previously obtained with the lower condensation temperature.

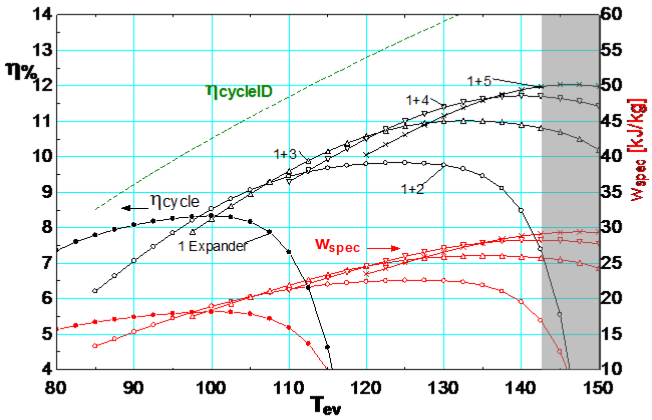

Figure 9. Electrical efficiency and specific work for different evaporation temperatures and different configurations of multi-expanders recuperated and re-heated ORC, with T_c=25°C.

It is worth noting that also the evaporation temperature necessary for overcoming the efficiency of the basic cycle becomes higher so that, if the temperature of the heat source does not allow reaching an evaporation temperature higher than 110°C, the basic cycle is certainly more efficient. An evaporation temperature higher than 115-120°C is recommended to ensure an efficiency gain of the multiple expansion configurations of at least one percentage point.

The regenerated and re-heated configurations (schematically shown in Figure 5) are considered in Figure 9, with the lower condensation temperature (T_c=25°C). The re-heating temperatures have been considered equal to the super heating ones, for all evaporation temperatures, verifying that this is consistent with the temperature diagram of the heat source. The results show that some increments can be obtained in the maximum efficiency and in the maximum specific work, by adopting the re-heating, but their amounts are generally smaller than those guaranteed by internal regeneration. In addition, the maximum of the electrical efficiency and of the specific work are obtained at higher evaporation temperatures, with respect of the same configurations, with internal regeneration but without re-heating. In any case, the maximum cycle efficiency of this study, η=12%, has been obtained for a 1+5 regenerated and re-heated configuration with an evaporation temperature T_{ev}=142°C. It has to be carefully evaluated if such a solution could be actually more convenient of, for instance, a regenerated (not re-heated) 1+4 configuration, with an evaporation temperature T_{ev}=130°C, that reaches an efficiency η=11,5%.

4. Conclusions

A design simulation model of a small ORC unit has been implemented and validated on the basis of the experimental results obtained, with a basic cycle configuration, on the test rig set up at the University of Trieste. The model has been then used to study different cycle lay-outs, with the aim of foreseeing the performance of more complex ORCs with two expansion stages in series, composed of a single scroll in the first stage and two to five scrolls in parallel in the second one, with and without internal regeneration and re-heating. The hypotheses of scroll expanders of the same kind, all rotating at the same speed have been assumed.

The considered multiple expansion configurations (from 1+2, to 1+5) allow reaching higher cycle efficiency and higher specific work, because each expander is operated at an expansion ratio closer to its optimum value. As the number of expanders in series in the second stage increases, the maximum efficiency also increases, but it is obtained at a higher evaporation temperature. As a consequence, the heat source too has to be at a higher temperature, or it is necessary to increase its mass flow rate. The performance improvement obtainable with one additional expander is each time lower.

When the evaporation temperature is lower than 95°C, the basic cycle obtains the same performance of the multiple expansion configurations, or even better. Therefore, multiple expansion configurations may be taken into account only for evaporation temperatures higher than 105°C, when they allow an efficiency gain of one percentage point, or more. These two limits become 110°C and 115-120°C, respectively, if the condensation temperature rises to 35°C from the minimum design value here considered (25°C).

By adding the internal regeneration, the efficiency gain in the optimal conditions are generally of about one percentage point or greater, while an additional gain of 0.5-1 percentage points can be obtained introducing both regeneration and re-heating. A careful evaluation of the required heat transfer surfaces will be necessary in possible future applications, in order of evaluating the economic advantages of these kinds of configurations.

If the hypothesis of using only scroll expanders of the same kind is relaxed, the second stage can be designed as a single scroll, for the same rotational speed of the first one. In this case, the advantage of using component directly derived from the automotive, or HVAC component market can disappears: it can be easily inferred, from the results here obtained, that a displacement volume of the second stage 2-5 times greater than the first one has to be considered, depending on the design value assumed for the evaporation temperature.

Acknowledgments

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References:

[1] AA. VV. (2016), Organic Rankine Cycle (ORC) *Power Systems: Technologies and Applications*, edited by E. Macchi and M. Astolfi, Elsevier, the Woodhead Publishing Series in Energy.

[2] S. Clemente, D. Micheli, M. Reini, R. Taccani, "Bottoming organic Rankine cycle for a small scale gas turbine: a comparison of different solutions," *Applied Energy*, *106*, 355–364, 2013.

[3] EES, F-Chart Software, http://www.fchart.com/ees .

[4] R. Bracco, S. Clemente, D. Micheli, M. Reini, "Experimental tests and modelization of a domestic-scale ORC (Organic Rankine Cycle)," *Energy*, *58*, 107-116, 2013.

[5] R. Zanelli, D. Favrat, "Experimental Investigation of a Hermetic Scroll Expander-Generator," in *Proc. International Compressor Engineering Conference*, Purdue, pp. 459-464, 1994.

[6] T. Yanagisawa, M. Fukuta, Y. Ogo, T. Hikichi, "Performance of an Oil-Free Scroll-Type Air Expander," *IMechE C591/027/2001*, pp. 167-174, 2001.

[7] V. Lemort, I. V. Teodorese, J. Lebrun, "Experimental Study of the Integration of a Scroll Expander into a Heat Recovery Rankine Cycle," in *International Compressor Engineering Conference*, Purdue, pp. 1-8, 2006.

[8] T. Saitoh, N. Yamada, S. I. Wakashima, "Solar Rankine cycle system using scroll expander," *J. Environ. Eng.*, *2 (4)*, pp. 708-719, 2007.

[9] M. Kane, D. Larrain, D. Favrat, Y. Allani, "Small hybrid solar power system," *Energy*, *28 (14)*, pp. 1427-1443, 2003.

[10] V. Lemort, S. Quoilin, C. Cuevas, J. Lebrun, "Testing and Modeling of a Scroll Ex-pander Integrated into an Organic Rankine Cycle," *Appl. Therm. Eng. 29*, pp. 3094-3102, 2009.

[11] X. D. Wang, L. Zhao J. L. Wang, W. Z. Zhang, X. Z. Zhao, W. Wu, "Performance evaluation of a low-temperature solar Rankine cycle system utilizing R245fa," *Solar Energy*, *84*, pp. 353–364, 2010a.

[12] J. L. Wang, L. Zhao, X. D. Wang, "A comparative study of pure and zeotropic mixtures in low-temperature solar Rankine cycle," *Appl. Energy*, *87(11)*, pp. 3366–3373, 2010b.

[13] J. Freeman, K. Hellgardt, C. N. Markides, "An assessment of solar-powered organic Rankine cycle systems for combined heating and power in UK domestic applications," *Appl. Energy*, *138*, pp. 605–620, 2015.

[14] Y. A. M. Kane, D. Larrain, D. Favrat, "Small hybrid solar power system," *Energy*, *28 (14)*, pp. 1427–1443, 2003.

[15] E. Yun, D. Kim, S. Y. Yoon, K. C. Kim, "Experimental Study on Parallel Expander Organic Rankine Cycle With Two Different Capacity Expanders," in *3rd International Seminar on ORC Power Systems*, October 12-14, 2015, Brussels, Belgium, 2015a.

[16] E. Yun, D. Kim, S. Y. Yoon, K. C. Kim, "Experimental investigation of an organic Rankine cycle with multiple expanders used in parallel," *Applied Energy*, *145*, pp. 246-254, 2015b.

[17] O. Rudenko, L. Moroz, M. Burlaka, C. Joly, "Design of Waste Heat Recovery Systems Based on Supercritical ORC for Powerful Gas And Diesel Engines," in *3rd International Seminar on ORC Power Systems*, Brussels, Belgium, October 12-14, 2015.

[18] T. Z. Kaczmarczyk, E. Ihnatowicz, G. Żywica, J. Kiciński, "Experimental Investigation of the ORC System in a Cogenerative Domestic Power Plant with a Scroll Expanders," *Open Eng.*, *5*:411–420, 2015.