SIMULATION MODEL OF AN EXPERIMENTAL SMALL SCALE ORC COGENERATOR

Stefano Clemente, Diego Micheli, Mauro Reini, Rodolfo Taccani

Dipartimento di Ingegneria Meccanica e Navale, University of Trieste, via A. Valerio 10, 34127 Trieste, Italy
E-mail: (sclemente, micheli, reini, taccani)@units.it

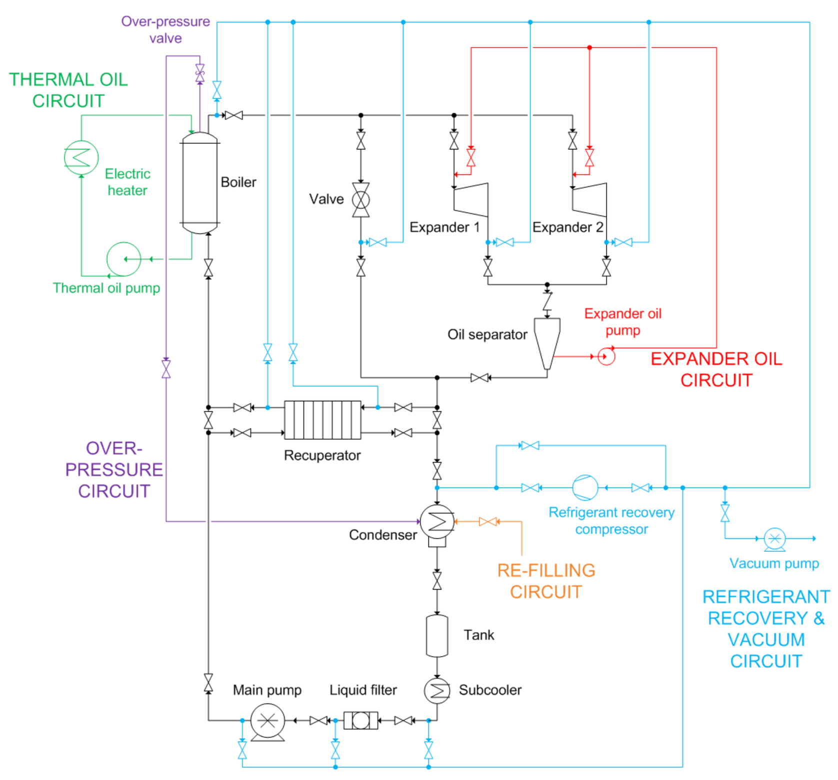
Introduction

The expected performance is presented of a small-scale ORC test bench, currently under development at the Energy Systems Laboratory (EneSysLab) of the Department of Mechanical Engineering and Naval Architecture of the University of Trieste.

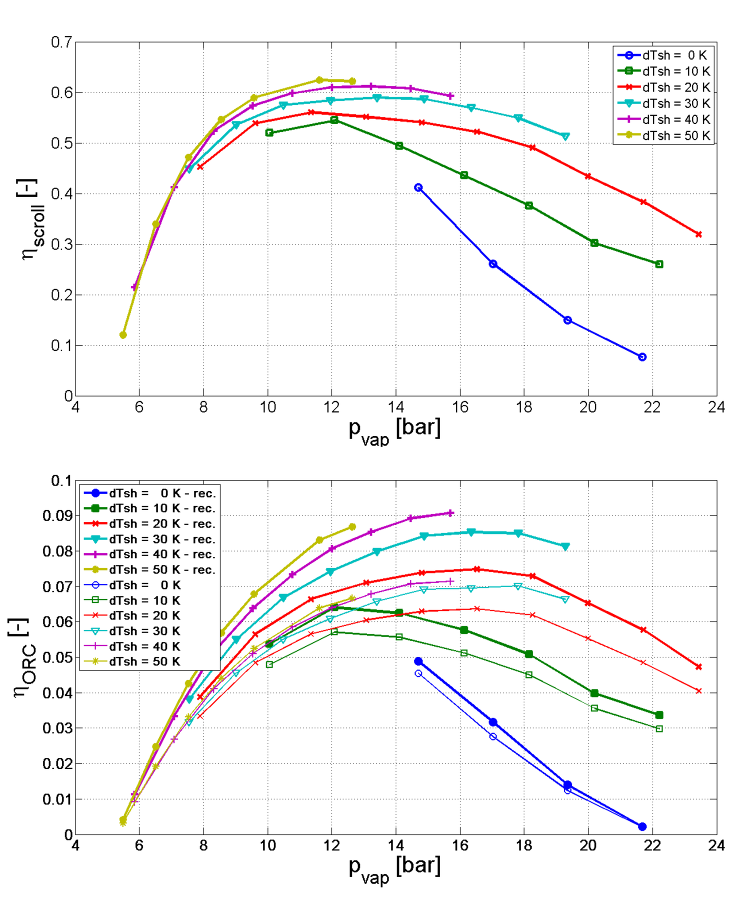
The simulation model has been implemented in Aspen®, taking into account the real behavior of the system components in design and off-design conditions. The heat exchangers have been modeled referring to the geometrical data provided by the manufacturer. Detailed one-dimensional models have been developed for the considered expanders, a compressor derived scroll and a newly designed piston type respectively.

R245fa and isopentane have been considered as working fluids.

ORC test bench scheme



Scroll expander and ORC efficiency

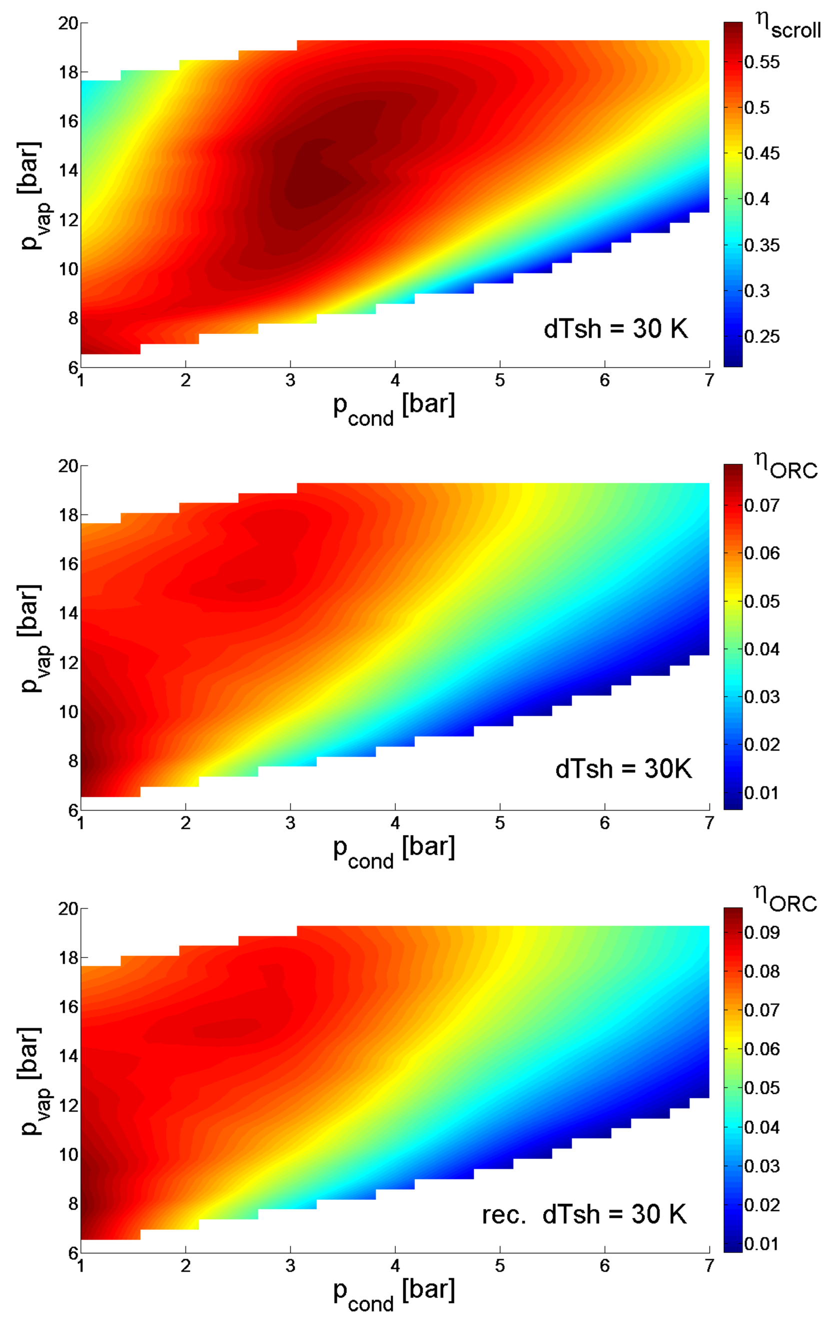


Figures refer to R245fa as working fluid, and show the efficiency vs. vapor pressure curves of the expander and of the ORC, in both the cases of simple and recuperative cycle. Condensing pressure is taken constant.

In particular, the effect of the superheating level is taken into account. Up to 30K of increment on the vaporizing temperature allows remarkably increasing cycle efficiency. Further increments lead to only negligible efficiency gains, while reduce the maximum achieving vapor pressure, because the highest allowed R245fa temperature is reached.

The maximum expected scroll efficiency is near to 0.65. If expansion ratio is lower than the best efficiency one, corresponding to the built-in volumetric ratio, efficiency decreases rather quickly. For higher values it keeps high up to vapor pressures corresponding to expansion ratios of several points higher than the optimal one. This asymmetry is typical of volumetric expanders, which are more penalized by over expansion than by under expansion phenomena.

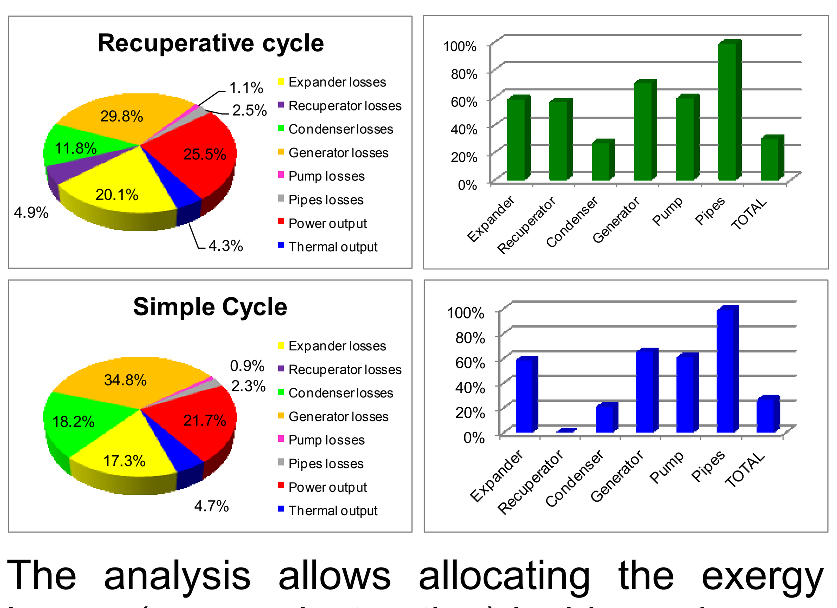
The ORC maximum efficiency is reached at higher vapor pressure values, because the increase of the theoretical cycle thermodynamic efficiency makes up for the decrease of the expander efficiency. Maximum expected efficiency is equal to 7% and 9% with the simple and regenerative cycle respectively.



Expander and ORC efficiency analysis is completed in these figures, which show the combined influence of vapor and condensing pressures. Superheating temperature increment is 30K and R245fa is the working fluid. Maximum ORC performance are obtained with vapor pressure between 7-12 bar and, obviously, with the lowest condensing pressure. But the good performance of the expander, even at high values of both the pressures, allows maintaining acceptable ORC efficiency values (red and yellow areas in the figures) also with high condensing pressures, if vapor pressure is also sufficiently high. This operating behavior is interesting in case of co-generative applications, because the temperature of the co-generated heat flow rate can be settled according to the specific application.

Recuperative cycle can be convenient, as it will be better explained in the next section.

Exergy analysis



The analysis allows allocating the exergy losses (exergy destruction) inside each component. In the pie-charts the input exergy is shared among exergy destructions and useful outputs, for both recuperated and simple cycle. Note that the recuperated cycle uses a smaller exergy input, with respect to the simple one (10.33 instead of 12.13 kW) for obtaining the same power output (2.6 kW), so that the former has a better exergy efficiency (29.8% instead of 26.4%) even if the latter produces more heat (33.2 kW instead of 26.1 kW) in energetic terms.

The bigger exergy destruction happens inside the generator, mainly due to the temperature difference between the thermal oil and the working fluid. Important exergy destruction also happens inside the expander and the condenser, while pump and pipes show contributions of about 1% and 2.5%, respectively. The condenser shows also a particularly low exergy efficiency, because of both the temperature difference between the working fluid and the hot water and the low output temperature of the same water, that is only 10°C higher than the ambient temperature (T°=25°C).

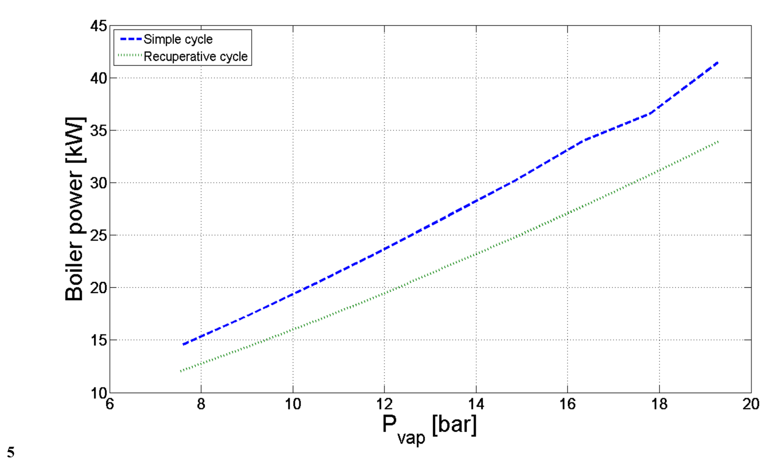
These results suggest that a better overall efficiency could be achieved in two ways:

- by reducing the exergy destruction in the two heat exchangers (generator and condenser);
- by reducing the exergy destruction in the scroll expander.

The first task can be addressed by adopting a mixture, instead of a pure working fluid. In this case, the phase changes happen at sliding temperature, allowing a reduction in the mean temperature differences between working fluid and the external thermal vectors and therefore in the exergy losses. Actual benefit can be obtained only if the mixture composition is properly defined and optimized.

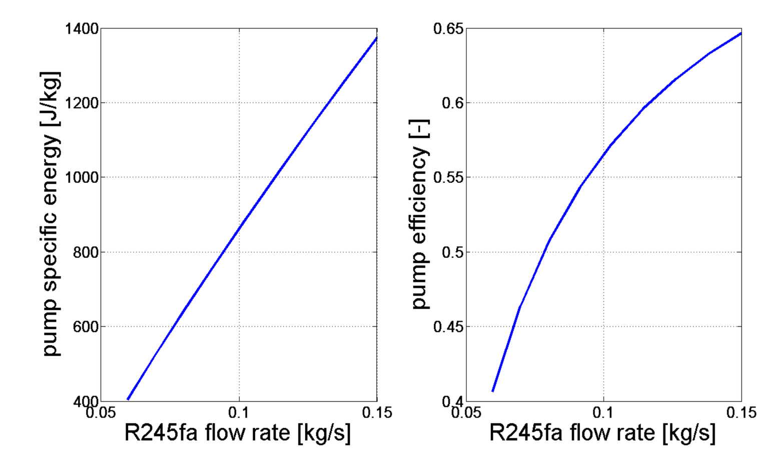
The second task implies a reduction in the leakage and mechanical losses, which have been recognized as the main causes of isentropic efficiency reduction in the scroll expander.

Boiler thermal input



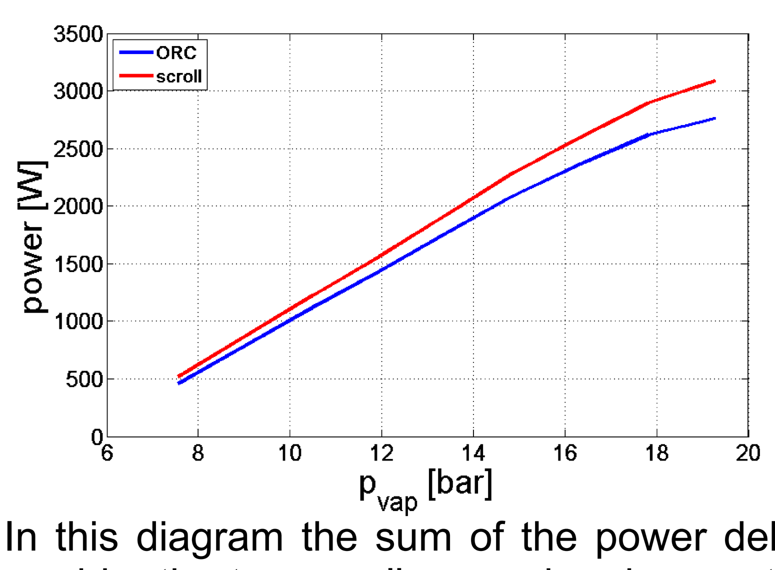
Here the thermal input of the boiler is reported as a function of the vaporization pressure, both for the simple cycle and for the recuperative one: it is interesting to note that the difference between the two curves (i.e. the saved thermal input) increases at higher pressures, where the recuperative system is even more convenient.

Pump performances



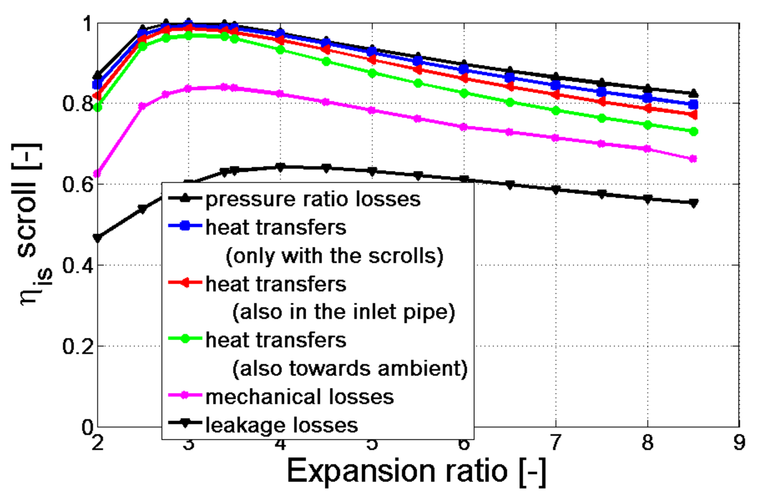
These diagrams show the performances of the considered diaphragm pump as functions of the delivered flowrate. Since this is a positive-displacement machine, the flowrate can be supposed, with good approximation, proportional to the crank speed of the machine. These curves, based on technical data from sheets provided by the producer, have been used to realize the pump model then inserted in Aspen.

Mechanical power



In this diagram the sum of the power delivered by the two scroll expanders is reported as a function of the evaporation pressure, together with the actual power achievable from the cycle: the difference between the two curves represents the power absorbed by the pump. It's clear, as obvious, that power of both the expanders and the pump increases with the vaporization pressure (for a given condensing pressure), but it is also interesting to note that, beyond a certain value of pressure, the gain in power becomes less-than-proportional.

Losses in the scroll expander



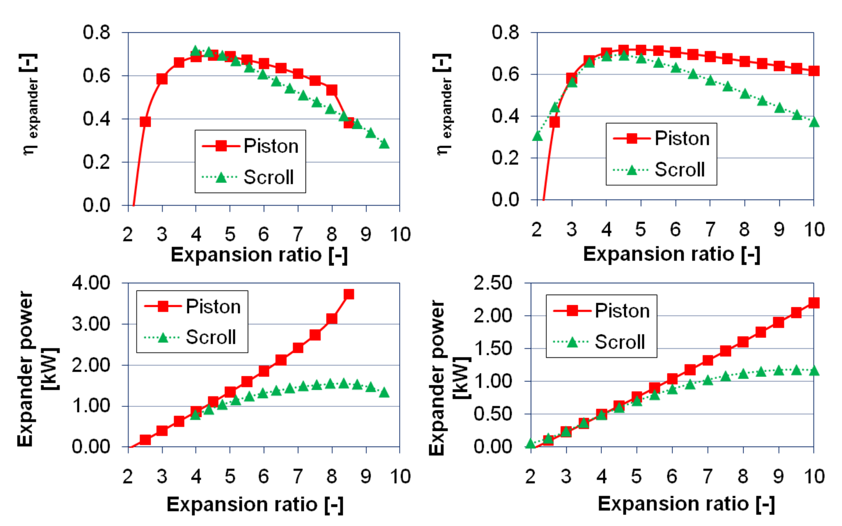
With reference to the losses due to an unadapted expansion ratio, unity efficiency is reached only if the scroll built-in pressure ratio is imposed between inlet and discharge.

Losses related to the heat transfers are quite important: the largest are relative to ambient losses, so the scroll expander has to be carefully thermally insulated.

Mechanical losses are approximately independent from expansion ratio, so they have a greater influence when less work is done, i.e. at smaller expansion ratios.

Leakage is by far the phenomenon most affecting the performances of a scroll machine.

Comparison between the performance of a scroll and a piston type expander



In some working conditions a piston expander could be considered as a valid alternative to the scroll one, since the first keeps working at high performances even at high expansion ratios.

In any case, the scroll taken into account has a built-in pressure ratio equal to 3, so better performances can surely be obtained by scroll machines designed to work with higher expansion ratios.

With isopentane the machine could achieve a higher efficiency than with R245fa in some conditions, but the delivered power is very lower, due to the fluid specific enthalpy and density.

Conclusions

The simulation model has given results consistent with the performance data made available by the component manufacturers.

The expected overall maximum mechanical efficiency is equal to 9%, or 7%, for recuperated and simple cycle, respectively. The simulated scroll expander is strictly derived from a commercially available scroll compressor, so it can be inferred that this kind of component could be used in the actual manufacturing of micro CHP units, producing about 2 kW of net electric power and 20-30 kW of thermal power.

Superheating of the working fluid enhances the efficiency of the unit, but temperature near to the chemical stability limit for the specific fluid can be approached, or even overcame. This is quite a strong limitation, adopting R245fa.

Exergy analysis suggests the convenience of reducing exergy destruction mainly in the generator and the condenser (for instance adopting a mixture as working fluid), and of reducing leakage and mechanical losses in the expander.

References

1. S. Clemente, D. Micheli, M. Reini, R. Taccani, "Numerical Model and Performance Analysis of a Scroll Machine for ORC Applications", ECOS2010, June 14-17, Lausanne (CH).
2. S. Clemente, D. Micheli, M. Reini, R. Taccani, "Preliminary design of Organic Rankine Cycles with scroll expanders" ICAE2011, May 16-18, Perugia (I).
3. S. Clemente, D. Micheli, M. Reini, R. Taccani, "Performance analysis and modelling of different volumetric expanders for small-scale Organic Rankine Cycles" ESFuelCell2011, August 7-10, Washington DC (USA).