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Dynamic control strategies for distributed microgeneration and waste heat recovery power plants

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Abstract

In this paper the modeling activity on a waste heat recovery microgeneration ORC plant is presented together with the results of the application of two different load diagrams and three different control strategies. The overall energy production and the average efficiency were compared and a proper control strategy was evaluated to optimize the energy recovery process as well as the dynamic response of the plant.

Keywords: low carbon city; urban energy system; renewable energy; sustainable development.

1. Introduction

In the last years the interest towards the low-enthalpy heat conversion into electrical energy has largely grown up, and this was due to the claim for pollutant emissions and fossil energy dependence reduction.

Organic Rankine cycles (ORC) are among the most studied solutions not only from the point of view of the overall efficiency but also for their operational flexibility and capability of following the energy input, which may be quite unpredictable especially when dealing with unprogrammable energy sources.

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Up to now, many systems are available with a minimum power output of about 0.5 MW, while in the lower range only few plants have been built so far. However, the minimum size of 500 kW might be oversized for many applications.

In the lower range of power output (up to 100 kW), several research paper about the use of volumetric expansion devices were published, due to the lower sensitivity to the moisture content of the operating fluid at the end of the expansion than turbines [1] and for their higher intrinsical flexibility of use [2-11].

This paper focuses on the improvement of the plant flexibility with respect to the variations of the energy input and a waste heat recovery system was taken as a case study. To the authors' knowledge, it is the first time that a working conditions dependent control strategy was evaluated.

2. Material and Methods

The work described in this paper was carried out through numerical analyses performed with the AMESim code and the volumetric expansion device employed was of the Wankel type, whose simulation model was presented in a previous paper [1]. This type of device proved to be suited for the power output range of 10-50 kW depending on the working fluid, which in this case is R-600a, based on previous published work [10]. The energy input diagram was also derived from literature and is representative of a typical application of waste heat recovery (from the exhaust of an internal combustion engine). The plant scheme is depicted in fig. 1 while the load diagrams considered are traced in fig. 2.

The differences between the two load cycles were in the amplitude of temperature variations, while mass flow rate was the same.

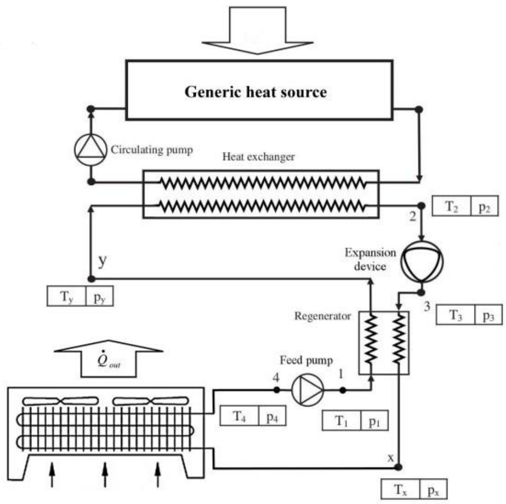

Fig. 1. Plant sketch

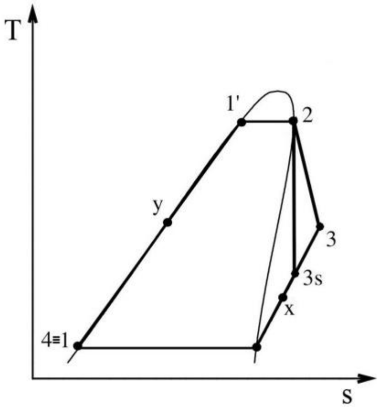

Fig. 2. Thermodynamic cycle on T-s diagram

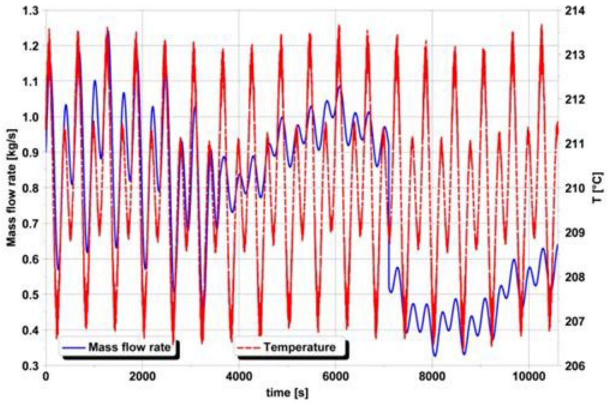

Fig. 3. Load diagram I

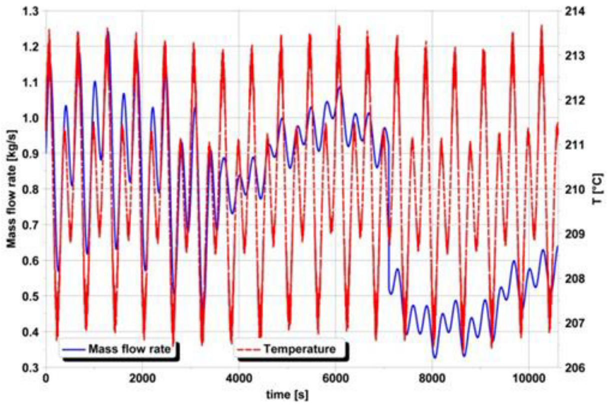

Fig. 4. Load diagram II

Three different control strategies were compared:

The first strategy was to keep constant the rotational speed of the expander and let the saturation pressure to vary freely.

The second strategy was to set a fixed saturation temperature set point while letting the rotational speed of the expansion device free of varying (by means of an inverter for instance). The rotational speed range was however limited in the range 500-3000 rpm.

The third strategy was to use a variable set point for the evaporating temperature in the range 100-120°C, so that the work output of the plant would be optimized. In this case it was necessary to find one or more control variables to be taken as a reference for the set-point management, to be easily measured in a really operating plant. These variables were selected by taking into account that the expansion device is a volumetric one, thus the delivered power is proportional to the volumetric flow rate $\dot{V}$ (given by the product of the rotational speed and the displacement) and to the inlet pressure pa. The study showed that the optimal set point temperature can be evaluated as a polynomial expression, whose coefficients an have to be evaluated for each considered case (depending on the device type, the rotating speed, the displacement and the working fluid):

$$\sum_{i=0}^{n} a_i (V \cdot p_a)^i \qquad (1)$$

3. Results and Discussion

The three different control strategies described in the above paragraph were compared. The average quantities are reported for each control strategy in Table 1 for the first load cycle (fig. 3) and in Table 2 for the second load cycle (fig. 4).

The first control strategy provided the lowest second law and overall efficiencies. The rotational speed had to be carefully chosen because higher values caused a significant decrease in the efficiency, while lower values let the saturation pressure approach and even overcome the critical value (these results are not reported here for the sake of brevity). This strategy is therefore suitable only for those plants whose load variation is limited.

The second strategy provided good results, but it was necessary, however, to find the optimum value of the set point. This type of control is suitable for those plants whose load variations are known and the optimal value of the set point could be defined a priori.

With the third type of control the maximum power output and overall efficiency were achieved with both the load cycles, as published in previous study [6]. This type of control is suitable for those systems characterized by unpredictable heat transfer fluid mass flow rate and temperature variations. Even if the resulting efficiency is similar to strategy II, in this case there was not the need for defining any parameter a priori, thus this kind of control can be defined as “self-adapting”, with evident advantages in terms of flexibility of use. Another advantages of the strategy III versus strategy II was the shortest start-up time (fig. 5).

Tab. 1. Calculation results

Load Cycle 1-2	Strategy I	Strategy II	Strategy III
Average net output [kW]	12.1 - 14.7	13.2 - 15.7	13.2 – 15.7
Exchanged heat[kW]	109 - 120	96 - 114	97 - 114
Average second law efficiency [%]	41.7 - 43.2	49.4 - 47.8	50 – 47.8
Average overall efficiency [%]	7.31 – 8.38	7.94 – 8.92	7.96 – 8.92

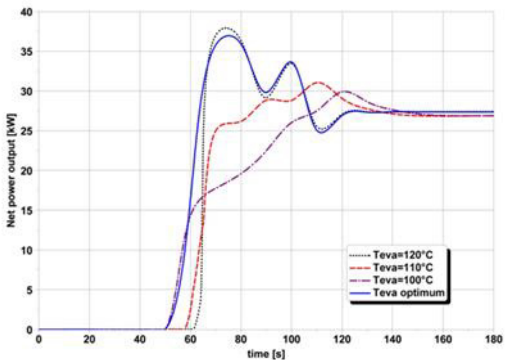

Fig. 5. Delivered power during a start-up process

4. Conclusions

In this paper the influence of the control strategy of an ORC cycle with a volumetric expander for waste heat recovery was analysed. R-600a was chosen as the operating fluid in a plant while the expander is of the Wankel type.

The calculations showed that while a constant rotational speed strategy provided the worse results in terms of net power output and overall efficiency, a constant saturation pressure strategy (at variable rotational speed) gave better results, but it also required the definition of an optimal value of the pressure set point as a function of the thermal input load. The better results were obtained by letting both the saturation pressure and the rotational speed vary, which in addition did not require the a priori definition of any thermodynamic parameter of the plant.

The second and the third strategy were then used to evaluate the startup process of the system. Also in this case the second and the third strategy gave similar results in terms of delivered power, but the third strategy also provided a faster start up process.

In conclusion, a fully flexible strategy not only provided a slight gain in delivered power, but also had the capability of self-accommodating the optimal saturation temperature set point as well as a faster response to load variations and start and stop operations.

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