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Numerical investigations of a Trilateral Flash Cycle under system off-design operating conditions

Matteo Marchionni[a], Giuseppe Bianchi[a,*], Savvas A.Tassou[a], Obadah Zaher[b], Jeremy Miller[b]

[a]*Brunel University London, Institute of Energy Futures, RCUK Centre for Sustainable Energy use in Food chains (CSEF), Uxbridge, UB8 3PH, United Kingdom*
[b]*Spirax Sarco Engineering PLC, GL53 8ER Cheltenham, United Kingdom*

Abstract

The current research presents a comprehensive model of a 100 kWe Trilateral Flash Cycle TFC system composed of sub-modules for the plate heat exchangers and twin-screw expanders to assess and identify the operating parameters that mostly affect the TFC performance at off-design conditions. The modelling approach for heat exchangers and piping is based on 1D CFD while considering the operating maps for pump and expander. The simulation process takes into account variations of the heat source temperature and mass flow rate from the design point. The TFC performance is analysed in terms of efficiency and power output while the expander performance is discussed in terms of volumetric and isentropic efficiencies. A sensitivity analysis is carried out to assess the most suitable parameter for control purposes and system performance optimisation. The results point out a large sensitivity of the inlet quality at the expander due to the revolution speed of the machine but also the off-design behaviour of the heater.

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Keywords: Trilateral Flash Cycle; control valve; system modelling; low grade thermal WHR;

* Corresponding author. Tel.: +44-1895-267707; fax: +44-1895-269777.
E-mail address: giuseppe.bianchi@brunel.ac.uk

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1. Introduction

Low-grade heat is the most common form of thermal energy which industrial processes make available as a by-product. Recent assessments on the industrial waste heat potential conclude that nearly 470TWh of primary energy are annually wasted at temperatures below 100°C, namely, the 51% of the total EU28 potential in industry [1]. The industrial sectors that are mostly characterised by low grade waste heat rejections are the chemical and petrochemical ones as well as food, tobacco, and paper, pulp and print [1,2]. Assuming an electric conversion with a 5% overall efficiency, around 16.5 $MtCO_2$ could be avoided every year from the low-grade energy waste.

Heat to power conversion is a topic widely discussed in the literature. Among the technologies, the Trilateral Flash Cycle (TFC) appears to be a promising candidate for low-grade applications. Besides waste heat recovery, low temperature heat to power conversion units could contribute to lower duty and size of heat sinks, such as cooling towers and dry coolers. This would not only lower capital and operational expenditures but also generate additional revenue streams for the final user.

The state of the art on TFC technology is mostly consists of academic studies related to theoretical performance comparison between TFC and other power cycles or about the screening of suitable working fluids [1] or mixtures [3]. In particular, a comparison between Organic Rankine Cycle (ORC), TFC and Organic Flash cycles showed that TFC might achieve the largest net power output, thermal efficiency and exergy efficiency [4]. With reference to automotive applications, the simulations showed that the TFC performed best for the charge air cooler [5,6]. Read et. al, [7] considered the usage of TFC in geothermal applications while in the work by Cipollone et.al [3] the heat source was a transcritical CO_2 refrigeration system.

The analysis carried out by Yari et. al [8] showed that although the TFC can achieve a higher net output power compared with the ORC and Kalina systems, its product cost is greatly affected by the expander isentropic efficiency. This concern would be more significant should the TFC system operate at off-design conditions. In order to quantify this impact, a comprehensive simulation platform has been developed in this study. The major element of novelty in the approach pursued was to account for the dynamic behaviour of the plate heat exchangers and consider performance maps for the centrifugal pump and the twin-screw expander. The results show how temperature and mass flow rate of the heat source as well as pump and expander revolution speeds affect the TFC and expander performance.

2. Methodology

The model herein presented resembles a heat to power conversion TFC unit designed for low thermal grade Waste Heat Recovery (WHR) applications. The waste heat is convoyed by a hot water stream, which flowing into a plate heat exchanger, the heater, transfers the heat to the working fluid of the system, the refrigerant R245fa. In particular, with reference to Figure 1, the refrigerant is pressurised by the pump and is heated up in the heater at constant pressure until its liquid saturation point. The heat gain therefore is achieved without any substantial phase change of the organic working fluid, other than a small vapour quality at the heater outlet. After the heating stage, the refrigerant flow is split in two parallel streams which are separately expanded in two twin screw machines. At the expander outlet the two-phase mixture is condensed in a set of three plate heat exchangers using water as cooling medium. Moreover, a receiver is placed downstream to the condensers to prevent pump cavitation and absorb thermal expansions of the working fluid potentially occurring during start-up operations or large transients.

The model of the TFC unit afore described has been developed in the commercial software platform GT-SUITE^{TM}. Each component employed in the considered plant configuration has been implemented with a dedicated template available in the software. For each template, accurate input data are of paramount relevance for the overall implementation and accuracy of the approach, and they can result from experimental measurements or more complex model results. For the heater and the condensers, these inputs have been retrieved through an online tool provided by the heat exchangers manufacturer [9]. The full modelling methodology has been presented in reference [10].

Once obtained, the data have been eventually inserted into the plate heat exchanger template of GT-SUITE^{TM} to compute the best fitting coefficients of Nusselt-Reynolds correlations used in the heat transfer calculation along the equivalent one-dimensional passages with which the heat exchanger channels are approximated [11]. The software predicts the vapor formation inside each sub-volume of the heat exchanger following the Rayleigh-Plesset equation [12] and computes the extension of the two-phase region, if any. Then, based on these predictions, it applies different correlations according to the fluid phase. In particular, Dittus-Boelter correlation is used in single phase heat transfer [11], while in the two-phase region, the correlation from Yao et al. has been considered for the condenser [13] while the one from Donowsky and Kandlikar is used for the heater [14]. The thermal inertia of the heat exchangers is considered by setting their geometry and their material properties.

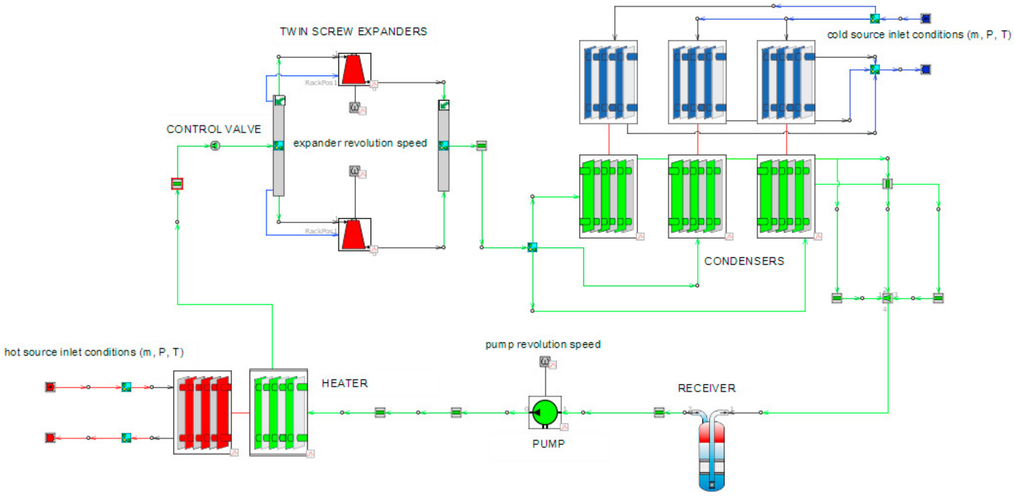

Fig. 1. TFC system configuration

Similarly, the pump performance curves provided by the manufacturer in a range of revolution speeds between 2000 RPM and 3500 RPM have been employed. The pump performance at lower or higher speeds, are extrapolated using a linear method.

The expander template is also map based, and the input are retrieved by a more complex model detailed in reference [15]. In particular four performance maps have been calculated for different refrigerant qualities at the expander inlet in a range between 0.0 and 0.4. For each of these, a set of values of mass flow rates, adiabatic and volumetric efficiencies have been inserted as a function of the revolution speed and expansion ratio across the machine. Therefore, during the simulation, the software interpolates the performance maps depending on the refrigerant quality at the outlet of the heater. To control this variable, which eventually affects the performance of the expanders, a control valve has been considered upstream of the two machines (Fig. 1) and it has been modelled as an orifice with a variable opening. The flow passage is equal to the cross section of the adjacent pipes. The minimum opening of the valve is considered as 9% of the nominal cross section area.

Finally, the receiver size has been set to 0.18 m^3 (almost the 25% of the system capacity) and connections between the several devices are made through piping sub-models, which neglect heat and distributed pressure losses. Localised pressure drops due to bends or t-split junctions are instead taken in account, while the electric machines connected to pump and expanders are not modelled, meaning that the power quantities considered are purely mechanical.

The boundary conditions imposed in the simulations are revolution speeds of pump and the expander as well as inlet temperatures and flow rates of hot and cold sources. The equations are solved with an implicit numerical method that approximates the system of algebraic differential equations to a system of nonlinear algebraic ones, which are eventually solved iteratively. The solution values at the next time step are simultaneously provided to all the sub-volumes of a given model (e.g. pipes divisions, heat exchangers channels etc.) [11] and the thermodynamic properties of the working fluid are interfaced with the solver through a dynamic-link library of the NIST Refprop database [16]. The validity of the modelling approach used is general and can be applied also to other heat to power conversion technologies (i.e. ORC, Kalina, sCO_2 etc.), and it is described in greater detail in reference [17].

3. Results and discussion

3.1. Off-design simulations

To analyse the TFC unit performance in off-design, some boundary conditions of the model have been varied with respect to the system reference point presented in Table 1. For each variable changed, the remaining ones have been maintained constant and equal to their reference values. The effect of these variations have been assessed from a system perspective, i.e. the thermal efficiency and the net power output, and for the expander, the isentropic efficiency, power output and the refrigerant quality at the inlet of the machine are presented. In all the simulations, the effect of the control valve is discarded (valve fully opened).

Table 1. Reference operating conditions.

Cycle variables	Hot water	R245fa	Cold water
Mass flow rate [kg/s]	7.84	24.65	130.30
Inlet/Max pres [bar]	4.0	6.4	3.0
Outlet/Min pres [bar]	3.9	1.1	2.7
Inlet/Min temp [°C]	85	18	12
Outlet/Max temp [°C]	25	63	17
System performance and setup			
Net power output [kW]		86	
Thermal efficiency [%]		4.3	
Expander efficiency [%]		74	
Valve opening [%]		100	

Table 2. Simulation matrix for the off-design analysis of the system.

Hot source		Min.	Design	Max.
Mass flow rate	kg/s	5.49	7.84	10.19
Inlet temperature	°C	75	85	95
Inlet pressure	bar		4	
Cold source				
Mass flow rate	kg/s		130.3	
Inlet temperature	°C		12	
Inlet pressure	bar		3	
Pump and expander				
Pump speed	RPM	2500	3000	3500
Expander speed	RPM	3000	4500	6000

Figure 2 shows the effect of expander speed variations on the system and the machine performance. It is possible to notice that the expander efficiency is greatly affected by its revolution speed, which consequently impacts the overall system efficiency and net power output, since the pump power consumption is fixed by the constant pump speed. In fact, when the expander speed is increased from 4500 RPM to 6000 RPM, the expander efficiency decreases from the optimal value of 74% to the minimum of 44%, leading to a drop in the system power output and overall cycle efficiency from 86 kW to 42 kW and from 4.3% to 2.1% respectively (Fig. 2a). A similar trend is shown if the expander speed is decreased. The refrigerant quality at the expander inlet is also slightly affected and it shows a maximum value of 0.12 for a revolution speed of 4825 RPM.

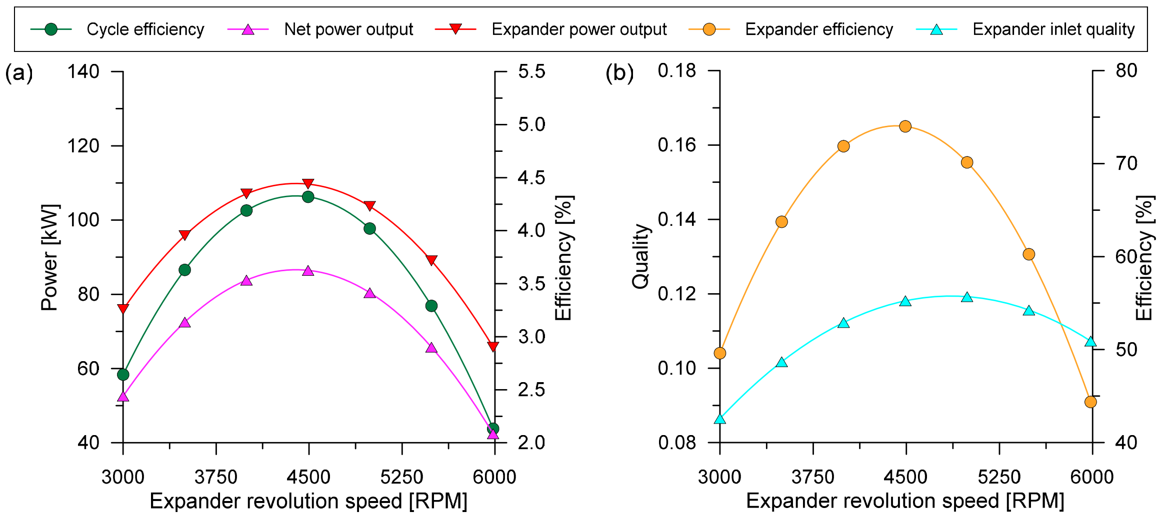

Fig. 2. Results obtained by the off-design model simulations varying the expander revolution speed:
(a) Thermal efficiency, net and expander power output; (b) expander efficiency and refrigerant quality at the inlet of the machine

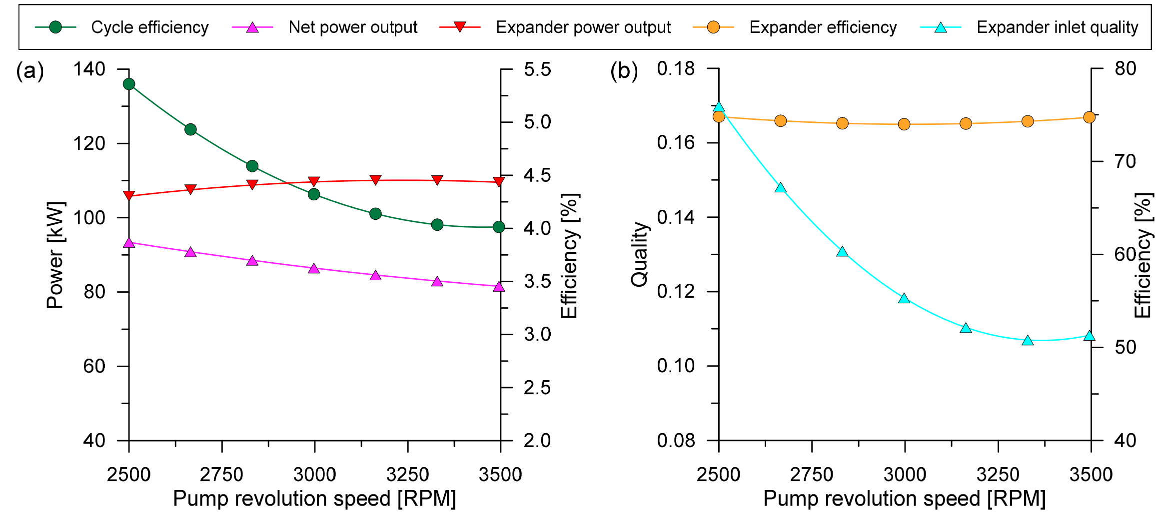

Fig. 3. Results obtained by the off-design model simulations varying the pump revolution speed:
(a) Thermal efficiency, net and expander power output; (b) expander efficiency and refrigerant quality at the inlet of the machine

An increase of the system performance can be noticed instead when the hot source inlet temperature and mass flow rate are increased. In the temperature case, the rise is steeper (Fig. 4), while for the mass flow rate, smoother trends can be observed (Fig. 5). In fact, an increment of the hot source inlet temperature from 75°C to 95°C increases the net power output from 60 kW up to 110 kW and the system thermal efficiency from 3.6% up to 5.2% (Fig. 4). For the same percentage change of the hot source mass flow rate, the power output and the efficiency of the TFC unit rise from 62 kW to 98 kW and from 4.2% up to 4.8% respectively (Fig. 5). This performance increase is not due to an augmented expansion ratio across the expander but rather to an increase of the refrigerant quality at the inlet of the machine, which rises from 0.10 up to 0.16 and from 0.11 up to 0.14 when the temperature and mass flow rate of the hot source are increased from 75°C to 95°C and from 5.84 kg/s to 10.19 kg/s respectively.

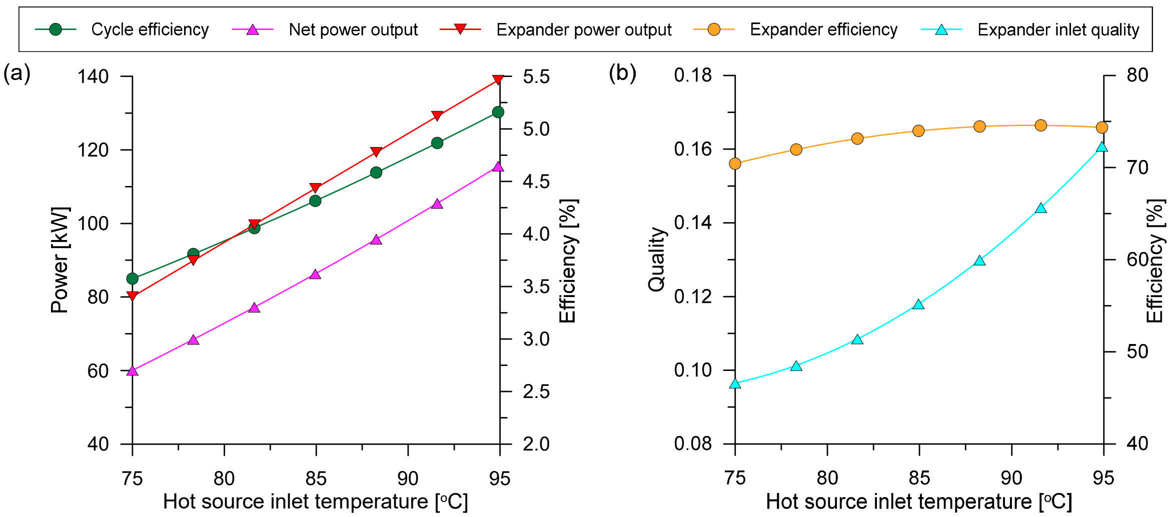

Fig. 4. Results obtained by the off-design model simulations varying the hot source inlet temperature:
(a) Thermal efficiency, net and expander power output; (b) expander efficiency and refrigerant quality at the inlet of the machine

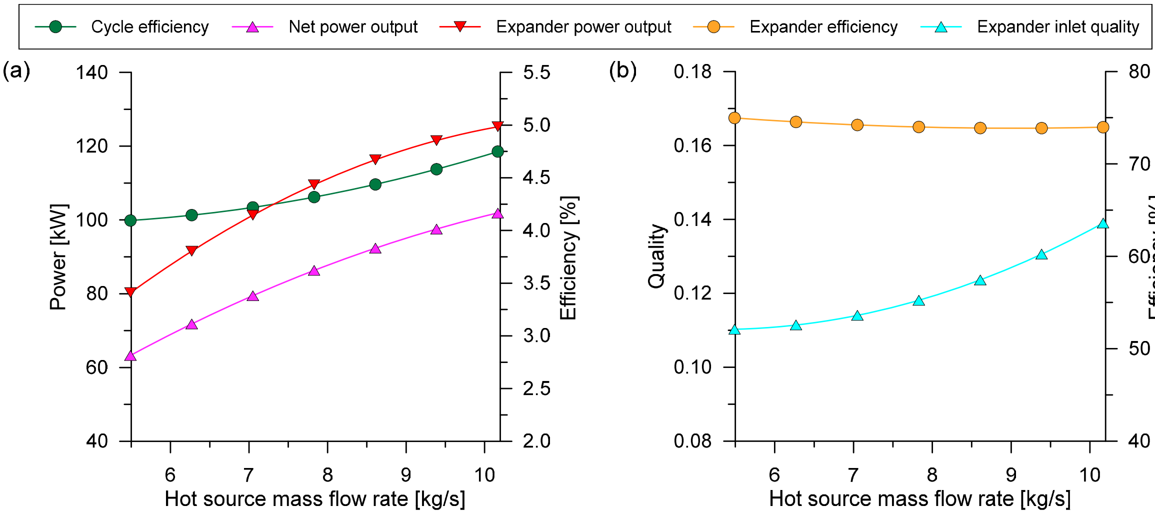

Fig. 5. Results obtained by the off-design model simulations varying the hot source mass flow rate:
(a) Thermal efficiency, net and expander power output; (b) expander efficiency and refrigerant quality at the inlet of the machine

3.2. Control valve operation

After the off-design analysis, a series of simulations have been carried out changing the valve opening area from the 9% (valve fully closed) to 100% (valve fully open) of the adjacent pipes flow area, while the other variables of the system were set at their reference value (Table 2). The results of the analysis are shown in Figure 6. It is possible to notice that the closing of the valve does not produce any sensible effect on the expander efficiency while it affects the working fluid quality at the expander inlet, which goes from 0.11 when the valve is fully opened up to 0.16 (Fig. 6b) when it is closed to its maximum position (9% control valve opening).

The increment of refrigerant quality is beneficial for the power generated by the expander since it goes from 110 kW up to 123 kW. This output increment consequently leads also to a higher net power output and the thermal efficiency of the system, which increase from 86 kW to 101 kW and from 4.3% to 5.2% respectively when the valve is closed. (Fig. 6a).

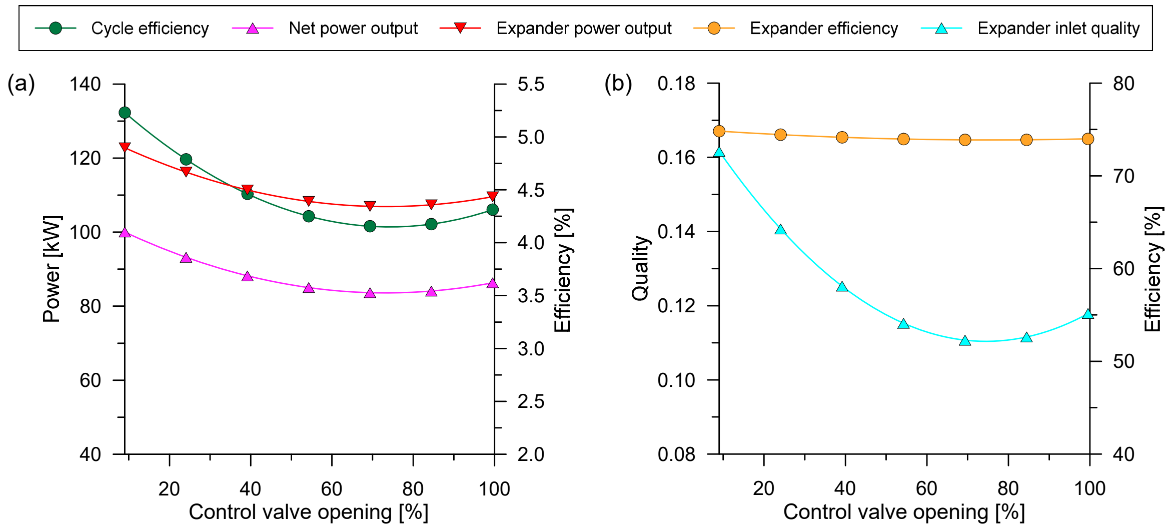

Fig. 6. Results obtained by the off-design model simulations varying the control valve opening area:
(a) Thermal efficiency, net and expander power output; (b) expander efficiency and refrigerant quality at the inlet of the machine

3.3. Sensitivity analysis

Finally, for control and optimisation purposes, a sensitivity study has been carried out to assess which main variable among the ones considered affects in a more significant way the system power output and the refrigerant quality at the expander inlet, which has been shown to play an important role in the expander performance. Therefore, the parameter considered in the previous analyses have been changed from a minimum to a maximum case. For the hot source inlet conditions and the revolution speed of the machines, the minimum, the reference and the maximum case are summarised in Table 2. For the control valve opening, a flow area of the 9%, 67.5% and 100% are considered for the minimum, reference and maximum case respectively.

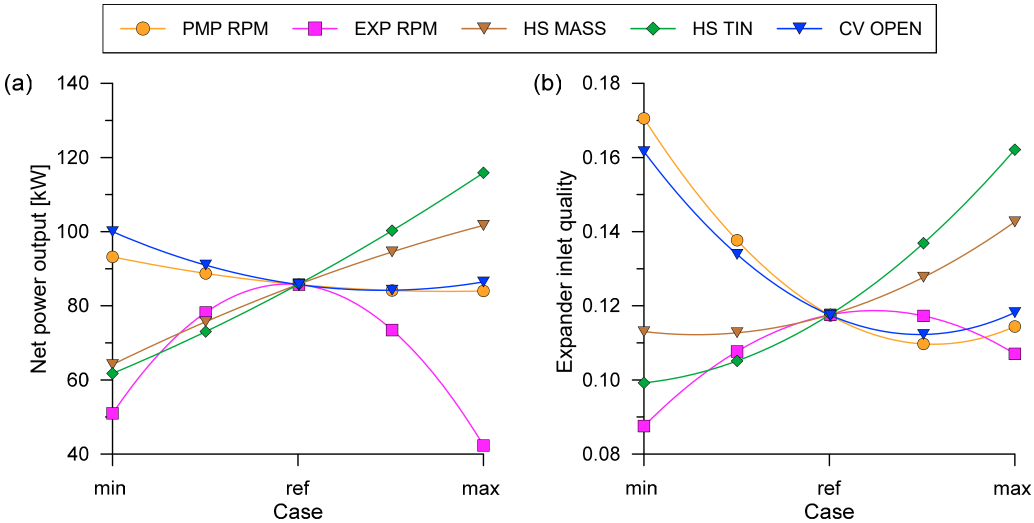

Fig. 7. Sensitivity analysis: (a) net power output, (b) refrigerant quality at the expander inlet

Figure 7 shows that the expander revolution speed and the hot source inlet temperature have the greatest impact on the system net power output. In fact, when both variables are switched from the minimum to the reference value, the net outcome of the unit goes from 61 kW to 85 kW when the hot source temperature is changed, while it rises from 50 kW up to 84 kW when the expander speed is varied. A smoother trend is shown for the same hot source mass flow rate relative variation, which allows to increase the power output from 65 kW up to 85 kW. Any sensible effect can instead be attributed on the control valve opening and the pump revolution speed variations.

On the contrary, the pump revolution speed greatly affects the refrigerant quality at the expander inlet, which is decreased from 0.17 to 0.11 when the pump speed set from the minimum (2500 RPM) to the maximum case (3500 RPM). An important effect is also shown from the control valve, which if closed can lead to an increase of the refrigerant quality up to 0.16. Noticeable effects are also shown by the hot source inlet conditions and the expander speed. These results suggest that the revolution speed of the pump and the control valve are suitable to control and optimise the refrigerant quality at the expander inlet rather than the inlet conditions of the hot source, which in rare occasion can be controlled or optimised in WHR applications since they depend on the process of the topping facility. The expander speed should be rather controlled in a narrow range close to its optimal working condition.

4. Conclusions

The modelling approach herein considered is versatile and general, and it can be applied even to different heat to power conversion technologies

An off-design analysis is carried out by varying the main system parameters to show their effect on the expander and overall unit performance. It has been found that the expander revolution speed should be varied within a narrow range, since if changed with respect to the optimal value of 4500 RPM it can provoke a drop in both the efficiency and the net outcome of the unit up to 52% respect to the target values (for a 33% speed variation). On the contrary, the pump revolution speed could be reduced to optimise the system performance since if it is decreased from 3000 RPM to 2500 RPM the power output and the overall efficiency of the system could increase from 86 kW to 93 kW and from 4.3% to 5.4% respectively. The increase of the hot source inlet temperature and mass flow rate also showed beneficial effects, which can mainly be attributed to the increase of the refrigerant quality at the expander inlet rather than the increment of the expansion ratio across the machine. However, unless an intermediate heating loop is designed to convoy the waste heat towards the unit, in WHR applications is rather complex to control or optimise these parameters since they depend on the operating condition of the topping facility.

The control valve and the pump revolution speed allow to change considerably the refrigerant quality at the expander inlet, since it increases from 0.12 to 0.16 and from 0.11 to 0.17 when the opening area of the valve is fully closed and the pump speed is decreased from 3500 RPM to 2500 RPM respectively. The control of both these variables might allow additional benefits from a system performance perspective.

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