Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 129 (2017) 567–574

Energy Procedia

www.elsevier.com/locate/procedia

IV International Seminar on ORC Power Systems, ORC2017
13-15 September 2017, Milano, Italy

Off-design study of a waste heat recovery ORC module in gas pipelines recompression station

Sonia L.Gómez-Aláez, Veronica Brizzi, Dario Alfani, Paolo Silva, Andrea Giostri, Marco Astolfi*

Politecnico di Milano, Energy Department, via Lambruschini 4a, 20156 Milano, Italy

Abstract

This study investigates the use of an ORC as heat recovery unit in a natural gas pipeline compression station powered by a gas turbine with the aim of increasing the process energy efficiency. A flexible Matlab® suite, able to investigate both subcritical and supercritical cycle, has been developed for the plant sizing and for the part-load simulation. The methodology to compute the system energetic performance is discussed. The ORC configuration that guarantees the maximum power output (7.22 MWe) is identified. The yearly electricity yield (42615.9 MWh) reveals good perspectives of implementing ORC with the aim of reducing the environmental impact of gas compression stations.

Peer-review under responsibility of the scientific committee of the IV International Seminar on ORC Power Systems.

*Keywords:*ORC, waste heat recovery, recompression station, off-design

1. Introduction

An emerging field of application for Organic Rankine Cycles (ORC) is the waste heat recovery (WHR) from the exhaust gases from the gas turbine (GT) which mechanically drives the compressor in gas pipeline recompression stations. The total length of gas pipelines worldwide is higher than 2.7 million km and recompression stations are

* Corresponding author. Tel.: +39 02 2399 3903.
*E-mail address:*marco.astolfi@polimi.it

1876-6102
Peer-review under responsibility of the scientific committee of the IV International Seminar on ORC Power Systems.
10.1016/j.egypro.2017.09.205

needed every 100-200 km depending on gas temperature, pipeline diameter and the gradients of natural gas (NG) demand to compensate the pressure drops. Each recompression station consists in a set of GTs (power rating from 5-35 MW) that mechanically drives the NG compressors in parallel. From the GT exhaust gases, a large amount of heat at relatively high temperatures (400-600°C) is available for heat recovery. ORC is the most suitable power system for this application as demonstrated by the fifteen ORMAT plants in North America installed since 1999 for a total electric power greater than 75 MW. On one hand, a great advantage of the installation of ORC is the high replicability of the same overall design for different recompression stations thanks to the relatively low variability of the exhaust gas thermodynamic conditions and mass flow rate through the pipeline, with clear benefits in terms of scale economies. On the other hand, the design and the operation of these systems are not trivial. During the year, they experience strong variations in both the heat input, mainly caused by NG compression power variation, and the ambient temperature since they are usually air-cooled condensed. In addition to the hourly fluctuations, seasonal pattern exists on the demand of NG mass flow. All these variations of the operating conditions, which are characterized by different time constants, affect the operation of ORC that works most of the time far from design point. With this scenario, it is crucial to design the ORC by taking into account the real operation throughout the year in order to maximize the energy production without incurring in technical and operational constraints violation.

2. Method

Considering the hourly and seasonal fluctuations of both NG volume flow rate into the pipeline and ambient temperature, it is clear that the design of a waste heat recovery ORC must rely on the results of a yearly simulation. Different plant designs, differing in heat exchangers surfaces, turbine size and nominal operating parameters would behave differently. This leads to the necessity to include, at design stage, information about the actual plant operation during a representative year and the presence of operational constraints. Using a set of opportune assumptions on plant components, the design of each component of the ORC is completely defined once the exogenous parameters (ambient temperature, flue gases mass flow rate and temperature) and the design cycle parameters (condensing temperature and turbine inlet temperature and pressure) are defined. Varying these six variables and comparing the annual results of different designs it is possible to define the optimal plant. This very general approach is not followed in the present work, and the authors preferred to define the optimal plant design in the following steps:

1. the compressor station is modelled and the trend of flue gases mass flow rate and turbine exhaust temperature are computed for one year of operation. The yearly average values of these quantities, weighted on the load of the GT connected to the ORC, are used as exogenous variables for the ORC design.
2. the design of ORC is optimized in order to maximize the net power output.
3. the plant operation is tested on one year of operation and the annual produced energy is evaluated
4. different combinations of ORC off-design parameters are tested in order to find the overall maximum of annual energy

Figure 1 illustrates the plant layout that comprises the compressor sets, which are made of two two-shaft GTs that mechanically drive NG compressors, and the ORC power block that receives flue gas from a single compressor set.

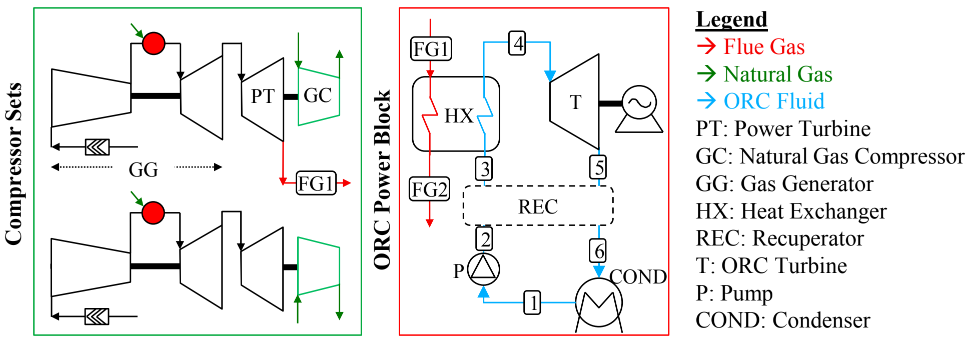

Figure 1. Layout of the plant. Recuperator (dashed line) is optional in ORC design.

2.1. Recompression station modeling

As shown in Figure 1, the ORC is bottomed up the GT that mechanical drives the NG compressor. In order to estimate the exhaust flue gases mass flow rate from the GT and their outlet temperature (the thermal input of the ORC), it is necessary to know the volume flow rate of natural gas flowing through the pipeline that determines the power required by the compressor and subsequently, the operation point of the GT.

In order to estimate the NG volume flow rate trend during the year, the authors referred to a three years experimental data set of Poggio Renatico compressor station [1]. Working on these data, monthly NG volume flow rate averages for different years have been computed. This compressor station is characterized by a maximum load in December and a minimum in July, while the whole plant is turned off during May and June as shown in Figure 4.

In autumn, the station processes a larger amount of gas than during spring, showing similar trends to other compressor stations located in different countries [2]. A simplified trend of the typical day NG flow rate has been implemented leading to an hourly profile of the demand different for each month. Finally, the total NG volume flow rate is scaled assuming to design the recompression station with two PCL801 centrifugal compressors driven by two PGT25 gas turbine, which is made up of an aero derivative gas turbine coupled with an industrial power turbine (nominal shaft power and rotational speed equal to 23.5 MW and 6500 rpm respectively) able to satisfy at full load the maximum pipeline request in December.

Under a certain flow rate threshold, identified by the maximum volumetric flow rate compressor limit, the whole compressor station demand is satisfied by a single unit (GT and NG compressor) in order to guarantee the highest efficiency. When the NG volume flow rate is larger than the maximum allowable limit by the compressor, the second unit is turned on and the load is equally split into the two units. In order to limit the additional investment cost we did not consider to install a flue gas collection system that would allow connecting the ORC to both the GTs. The primary heat exchanger (PHE) of the ORC is designed to collect the exhausts from a single GT which is operated in priority mode in order to maximize the operating hours of the ORC. It is worth underlining that the proposed configuration is not representative of all the pipeline compression stations since often GT redundancy is implemented and the GTs are operated in rotation mode. However, this last configuration would require a more complex and expensive system for the flue gases collection provided with diverters to handle GTs operation changes.

The mechanical power required by the compressor is hence computed through the PCL801 compressor operating map [3]. The compressor works at nearly constant polytropic efficiency in a wide operational range; then follows the maximum rotational speed curve until the maximum flow rate limit of 44240 m^3/h (measured at pipeline conditions), is reached.

Taking into account the mechanical efficiency, the mechanical power supplied by the GT and its relative partial load is determined. Figure 2 depicts the PGT25 performance map in terms of partial load and ambient temperature [4] and it is possible to highlight a minimum load equal to 10% of the ISO conditions power. For the calculations, the authors refer to the ambient temperature representative of Poggio Renatico (ITA). Since the monthly mean day data of this location are not available, the weather data of the closest location available on Energy plus [5] (Bologna) are taken and scaled in order to match monthly mean of Poggio Renatico.

Summing up, considering the monthly average, an hourly profile of ambient temperature and NG volume flow rate have been defined in order to be representative of a whole year. Taking into account the volume flow rate flowing through the pipeline, through the coupling mentioned before of the compressor set, the load of the GT is determined, so the thermal input of the ORC (the exhaust temperature and mass flow rate of flue gases).

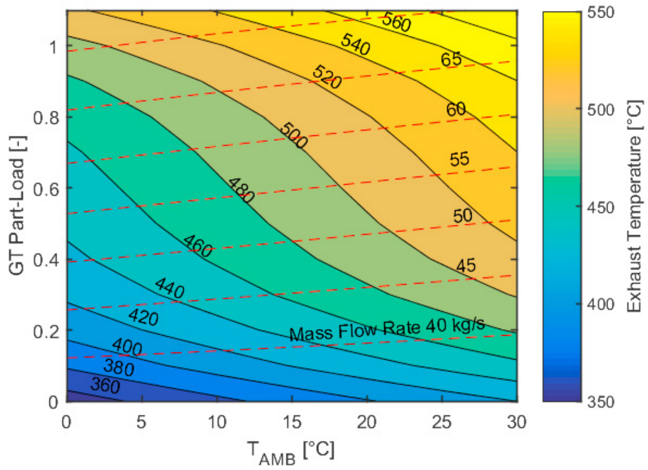

Figure 2. PGT25 performance map

2.2. Definition of the ORC design

The components of the ORC are: i) once-through PHE, ii) multistage axial turbine iii) air condenser, iv) pump and v) Recuperator (optional) (see. Figure 1). The GT+ORC model has been developed in Matlab®. The model allows to optimize both subcritical (saturated and superheated) and supercritical cycles, including the use of a recuperator. Different working fluids can be investigated thanks to the use of NIST Refprop 9.1. However, just R1233zd, that is proposed as substitute fluid of well-known R245fa because of its very low environmental impact (it is characterized by 0 ODP (Ozone Depletion Potential) and 0 GWP (Global Warming Potential)), has been chosen (T_{crit}=165.6 ºC, p_{crit}=35.709 bar). The study of ORC implementing different fluids is beyond the scope of the current research and it will be treated as future development. The exogenous parameters of the cycle are the ambient temperature, the mass flow rate of flue gases and their temperature, while the parameters of the cycle itself are the pressure and temperature at the inlet of the turbine and the condensation pressure. With the set of these parameters and the use of the assumptions reported in Table 1, all the thermodynamic state points, the working fluid mass flow rate and the cooling air mass flow rate are computed. The isentropic efficiency (η_{is}) of the 3 stages turbine is computed as function of the isentropic enthalpy head Δh_{is} and the volume ratio Vr_{is} [6]. Air cooled condenser fan consumption is evaluated with the correlation proposed in [7, 8]. A thermodynamic optimization has been carried out aiming to the ORC maximum power output using Matlab's *patternsearch* algorithm. Finally, the turbine reduced mass flow rate and heat exchanger surface are computed with the assumptions reported in Table 1.

Table 1. ORC plant assumptions

Pressure losses			General assumptions			Global heat transfer coefficients		
$\Delta p_{economizer}$	0.05	MPa	$\eta_{y,pump}$	0.8	-	$U_{PHE,liquid,EVA}$	635	W/m²K
$\Delta T_{evaporator}$	1	ºC	$\eta_{m\text{-}e,pump}$	0.97	-	$U_{PHE,vappr}$	343	W/m²K
$\Delta p_{superheating}$	$0.02 \cdot p_4$	MPa	$\eta_{gen\ m\text{-}e,turbine}$	0.97	-	$U_{condenser,2phase}$	1089	W/m²K
Δp_{PHE}	$0.05 \cdot p_4$	MPa	η_{PHE}	0.99	-	$U_{condenser,vapor}$	322	W/m²K
$\Delta p_{recuperator,\ liquid}$	0.05	MPa	$Q_{loss,recuperator}$	0.01	-	$U_{recuperator}$	732	W/m²K
$\Delta p_{recuperator,\ vapor}$	$0.02 \cdot p_6$	MPa	$\Delta T_{pp,PHE}$	25	ºC			
$\Delta p_{desuperheating}$	$0.01 \cdot p_3$	MPa	$\Delta T_{pp,recuperatot}$	1	ºC			
$\Delta T_{condenser}$	0.3	ºC	$\Delta T_{pp,condenser}$	1	ºC			
$\Delta p_{desuperheating}$	$0.01 \cdot p_1$	MPa	$\Delta T_{pp,recuperatot}$	1	ºC			
$\Delta T_{condenser}$	0.3	ºC	$\Delta T_{pp,condenser}$	1	ºC			

2.3. System Off-design and yearly simulation

Once the optimum on-design configuration has been calculated considering as exogenous parameters those obtained as a weighted average based on the GT load during the year (shown in Table 2), and the turbine and heat exchangers are completely designed the annual simulation is performed considering the part-load behavior of the whole system. Off-design routine finds iteratively the operating point of the compressor sets and ORC system. In particular, the compression station is modelled in accordance with the map-based approach described in Sec.2.1, while for the ORC the off-design operation is found solving a set of non-linear equations that characterizes each plant component while keeping the heat exchangers surface and the turbine reduce mass flow rate constant. As regards ORC turbine, a sliding pressure control is considered while the isentropic efficiency is computed by using a modifier function as function of ratios of Δh_{is} and Vr_{is} respect to their nominal values [9, 10]. The values of the convective heat transfer coefficients have been considered as function to the volumetric flow rate ratio with respect to the on-design conditions while pressure drops are computed assuming $\Delta p/p$ ratio equal to the sizing one. A threshold value of turbine volume flow rate twice than the nominal one is considered in order to account for chocking at turbine discharge section. If the computed volume flow exceeds this limit value (it could happen for very low ambient temperature and high GT load) fan speed is reduced limiting the reduction of condensing pressure with a reduction of system auxiliaries consumption. We also verified that it is always convenient to set the maximum temperature of the cycle equal to the nominal value while it could be convenient for some operation point to reduce the fan speed.

3. Results

3.1. Optimal on-design configuration

In order to define the ORC design, we first calculated the ORC plant exogenous parameters namely the mass flow rate of flue gases, the outlet temperature of the gas turbine and the ambient temperature as the yearly average quantities weighted on the load of the GT connected to the ORC. The optimal ORC configuration which maximizes the net power output is a supercritical recuperative cycle with a maximum temperature near the limit of thermal stability of the fluid (about 277 ºC, reported in Refprop as the limit of EoS use). Table 2 reports the main output of the ORC design in terms of both powers and heat exchangers surface (A) of PHE, condenser and recuperator. In addition, the corrected mass flow of ORC expander is reported together with first law (η_I) and second law efficiency (η_{II}) (minimum flue gas temperature equal to 80°C).

Table 2. Optimal configuration results

Parameters			Powers			Surfaces		
$m_{flue\ gases}$	60.51	kg/s	P_{net}	6314.79	kW	A_{PHE}	492.13	m^2
$T_{max,\ flue\ gases}$	491.01	°C	$P_{turbine}$	7173.77	kW	$A_{condenser}$	3428.24	m^2
$T_{ambient}$	10.89	°C	P_{pump}	562.78	kW	$A_{recuperator}$	651.74	m^2
Average GT load	66.12	%	P_{fan}	296.20	kW	$m_{corrected}$	0.3464	$kg/s{\cdot}K^{0.5}/Pa$
$T_{in,turbine}$	250.00	°C	$Q_{recuperator}$	7166.03	kW	η_I	23.14	%
$p_{out,condenser}$	1.26	bar	$Q_{condenser}$	17851.80	kW	η_{II}	49.60	%

3.2. Annual simulation

Before introducing the annual simulation results, it is worth to describe the part-load model capabilities with two part-load cases with extremely different thermal loads.

Figure 3 shows two extreme cases of off-design conditions; in particular, when the thermal load is maximum (GT load= 91.42% and T_{amb}=1.82 ºC), the cycle remains supercritical, but the pressure at the inlet of the turbine is higher (63.39 bar) than the design pressure (green). On the opposite, when the thermal load is minimum (GT load 11.39% and T_{amb}=20.74ºC), the maximum pressure decreases at the point that the cycle becomes subcritical (32.83 bar) (blue).

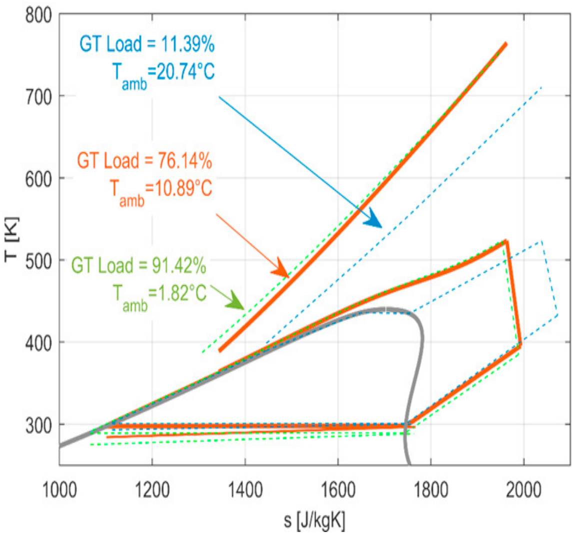

Figure 3. T-s diagram of ORC in on-design (orange) and in correspondence of maximum GT load (green) and minimum GT load (blue).

Figure 4 shows the ORC net power along the year together with the inputs of the model (i.e., NG flow rate, GT load fraction, maximum NG flow rate for a single compressor).

On the other hand, Table 3 reports the minimum and maximum power, the mean net power produced and the monthly energy produced. As expected, when the NG requested is higher, the energy produced by the ORC is also higher. So, in the coldest months, the energy production is maximized, reaching 5335 MWh in December. Thanks to the proposed arrangement the average load of the ORC ranges from 55.3% in July and 113.5% in December with an annual average load equal to 78.9%. The most penalized case shows a reduction of efficiency close to 5% points.

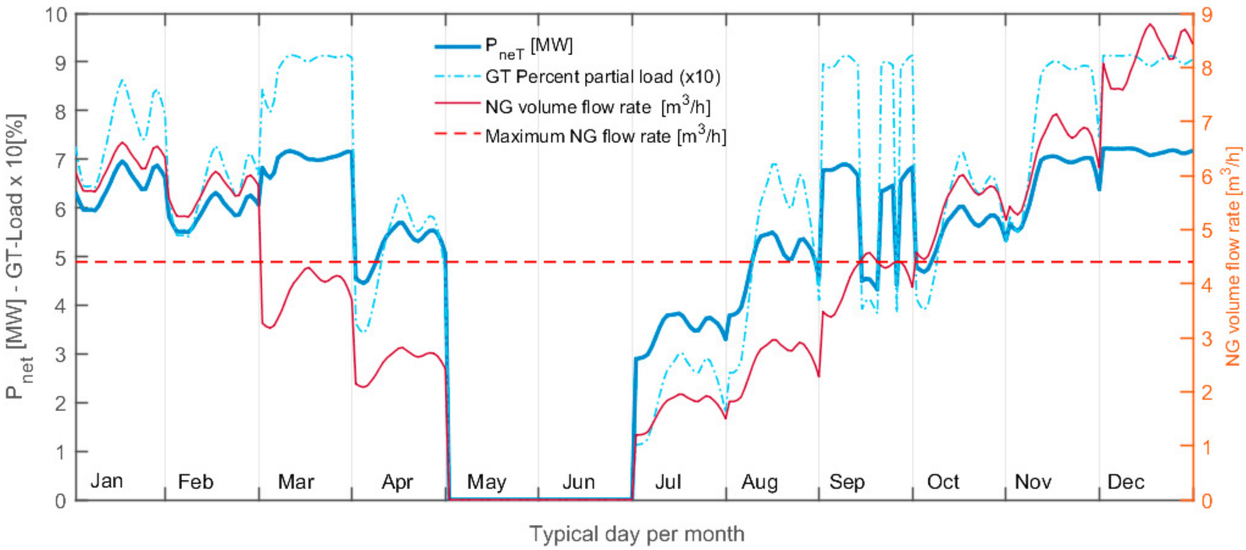

Figure 4. Annual net power profile of relevant quantities for a gas pipeline recompression station

Table 3. Monthly net power analysis, total energy produced and average recovery efficiency.

	P [kW] Monthly mean	P_{max} [kW]	P_{min} [kW]	Total monthly Energy [MWh]	η_{plant} Monthly mean
Nominal	*6314.8*				*23.14%*
January	6463.1	6957.2	5945.3	4808.6	24.27%
February	5921.1	6307.7	5499.5	3979.0	24.12%
March	7013.7	7172.2	6614.9	5218.2	23.09%
April	5219.3	5693.0	4452.9	3757.9	22.77%
May	0.0	0.0	0.0	0.0	0.0
June	0.0	0.0	0.0	0.0	0.0
July	3490.5	3830.3	2897.3	2596.9	19.44%
August	4901.1	5497.5	3787.3	3646.4	20.20%
September	6135.4	6893.6	4320.5	4417.5	20.99%
October	5510.2	6028.7	4684.4	4099.6	22.53%
November	6606.7	7063.2	5528.8	4756.8	23.45%
December	7170.7	7223.7	7080.3	5335.0	23.92%
Yearly total				42615.9	
Yearly average	4980.51				19.07%

In order to validate the proposed methodology two maps have been created to evaluate the net power produced for different ambient temperatures and percent of GT load by varying the cooling air volume flow rate and the maximum temperature at the inlet of the turbine. Figure 5.a depicts the effects of varying the air volumetric flow rate at the air cooled condenser. This parameter affects both to the condensation pressure (and thus the ORC turbine power), but also to the consumption of the fan. We found that keeping the same volumetric flow rate equal to the nominal one during all the year it is not convenient. In fact at low values of ambient temperature and GT load it is more convenient to work at lower cooling air flow rates, because the fan consumption reduction is larger than the reduction of turbine power due to the increase of condensation pressure. Figure 5.b shows the effect of different maximum temperature at the inlet of the turbine. Modifying that value becomes relevant at low temperature and low GT load. In winter, when ambient temperature is low, the NG request is higher, so the GT works at higher load; while in summer, when the ambient temperature is high, the NG request is lower, so the GT works at low partial loads. The combination of low temperature with low load is not interesting for this study because all the operating points remain far from that condition, so the maximum temperature of the cycle in off-design conditions is set at the same maximum temperature as the optimal design configuration.

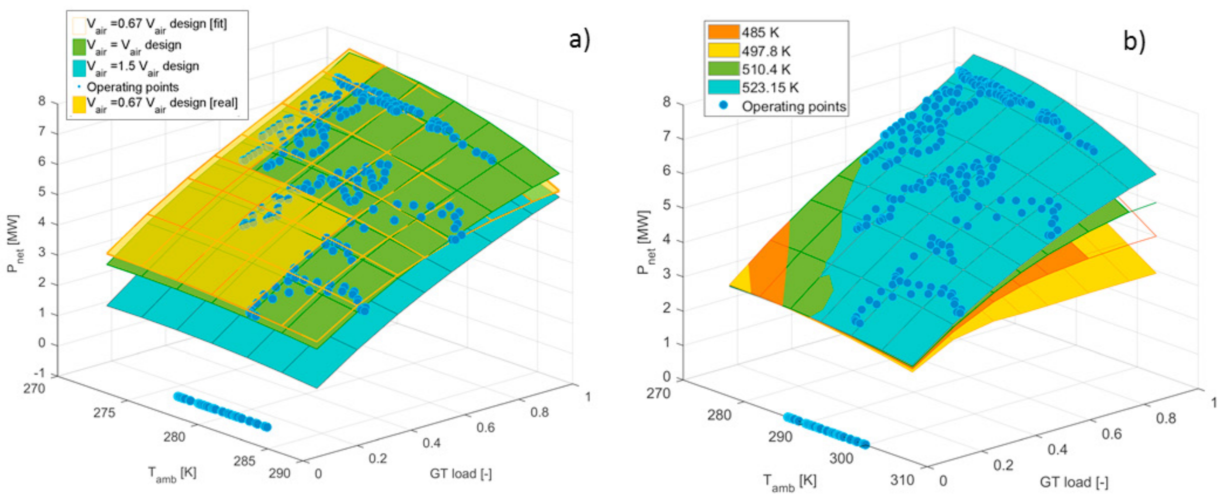

Figure 5. Sensitivity analysis on ORC power output varying a) the fan speed (air volume flow rate) b) the turbine inlet temperature against GT load and ambient temperature. Each marker represents the operating point of each hour of the yearly simulation.

For sake of completeness, it is worth to underline that, for reduced fan volume flow, models can fail to reach numerically convergence. The white surface with yellow outline is obtained by fitting the simulation results that reached convergence. It is show for sake of completeness.

4. Conclusions

This paper discusses the development of a methodology approach and simulation tool for thermodynamic and part-load assessment of ORC coupled with NG compression stations. The developed model, implemented in Matlab®, allows the optimization of ORC design for different fluids and different conditions: both subcritical (saturated and superheated) and supercritical cycles, and including or not the use of a recuperator. The optimization target is to identify the ORC design that maximizes the yearly electricity yield attainable from flue gases WHR.

The code is tested on a WHR application based on a recompression station along a gas pipeline.

An optimal on-design configuration for annual weighted average conditions has been calculated, obtaining a net power of 6.3 MW from a thermal input of 25 MW. First law efficiency (23.14 %) and second law efficiency (49.60 %) confirms that the studied solution is able to offer an efficient recovery of waste heat. With this configuration, the off-design simulation has been performed for a representative year (consisting of a whole characteristic day for each month), as a function of ambient temperature and compressor sets load. These simulations allow estimating the variation of the energy produced along the year (42615.9 MWhe/year corresponding to 6748.6 equivalent hours) and the performance of the ORC unit at different conditions. Starting from the findings of the present study, future developments could be addressed towards the development of a rigorous optimization procedure that searches the design configuration that maximize the yearly energy yield or minimize the Levelized cost of Electricity.

References

[1] Rossin, D., Silva, P. and Bischi, A. (2013). *Modellizzazione di un impianto di trigenerazione applicato alla centrale di ricompressione gas di Poggio Renatico (Fe)*. Master thesis. Politecnico di Milano.

[2] Interstate Natural Gas Association of America, (2010). *Interstate Natural Gas Pipeline Efficiency*. Washington, D.C.

[3] Nuovo Pignone, (2006). *Curve di funzionamento attese per compressore centrifugo PCL801*. Italia.

[4] Willbros Engineers, Inc., (2005). *Compressor station cost factors*. Tulsa, Oklahoma.

[5] Energy Plus database. https://energyplus.net/weather-location/europe_wmo_region_6/ITA//ITA_Bologna-Borgo.Panigale.161400_IGDG

[6] Macchi, E. and Astolfi, M. (2017). Axial flow turbines for Organic Rankine Cycle applications. In: E. Macchi and M. Astolfi, ed., *Organic Rankine Cycle (ORC) Power Systems. Technologies and Applications*, 1st ed. Woodhead Publishing, pp.302-313.

[7] Astolfi, M., Romano, M., Bombarda, P. and Macchi, E. (2014). Binary ORC (organic Rankine cycles) power plants for the exploitation of medium–low temperature geothermal sources – Part A: Thermodynamic optimization. *Energy*, 66, pp.423-434.

[8] Astolfi, M., Romano, M., Bombarda, P. and Macchi, E. (2014). Binary ORC (Organic Rankine Cycles) power plants for the exploitation of medium–low temperature geothermal sources – Part B: Techno-economic optimization. *Energy*, 66, pp.435-446.

[9] Capra, F. and Martelli, E. (2015). Numerical optimization of combined heat and power Organic Rankine Cycles – Part B: Simultaneous design & part-load optimization. *Energy*, 90, pp.329-343

[10] Ghasemi, H., Paci, M., Tizzanini, A. and Mitsos, A. (2013). Modeling and optimization of a binary geothermal power plant. *Energy*, 50, pp.412-428.