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Thermodynamic analysis of an Organic Rankine Cycle for waste heat recovery from an aeroderivative intercooled gas turbine

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Abstract

This paper presents a study of an Organic Rankine Cycle combined with an intercooled gas turbine: thermodynamic analyses are carried out using four different organic fluids (toluene, benzene, cyclopentane and cyclohexane). Organic Rankine Cycle can be combined with a gas turbine through a diathermic oil circuit in order to convert gas turbine waste heat into electrical power: ORC can be a promising choice for waste heat recovery at low/medium temperatures. An intercooled gas turbine is characterized by low exhaust temperature, and the Organic Rankine Cycle, that can work with lower temperature respect to a Rankine cycle, can be an interesting solution to improve the efficiency of the power plant. In an intercooled gas turbine with high pressure ratio, waste heat can be recovered from exhaust gas and also from the intercooler: air temperature exiting from the first compressor is about 160-220°C and it is generally cooled by water. This waste heat can be recovered by an Organic Rankine Cycle to convert the low-temperature heat source into mechanical energy and increase the global power plant efficiency.

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Keywords: Organic Rankine Cycle; Intercooled Gas Turbine; Combined Cycle; Power Plant; Low-Temperature Heat Recovery.

1. Introduction

The increasing fuel costs are forcing governments and industries to increase the cycle efficiency of engines or improve combined gas turbine cycles [1,2]. The use of an Organic Rankine Cycle (ORC) is a good solution to recovery waste heat at low/medium temperatures. In fact, the low temperature heat discharged in several industrial applications can't be exploited with a traditional water Rankine Cycle. In such applications where the temperature of

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waste heat is in the 300-450°C range, they stand as a very interesting option to enhance system performance. ORC cycles can have high thermodynamic efficiency with low mechanical stresses and longer component life compared to other cycles; it's important to notice that they can have a good flexibility, especially for startup phase, and that an ORC cycle can have a light and small packaging. Waste heat recovery ORCs have been studied in a number of previous works [3, 4, 5, 6] that used simple thermodynamic models comparing different working fluids for low and high temperatures. For power production with gas turbine based combined cycles, generally bottoming cycles are traditional steam-water Rankine cycles, but the use of ORCs is an attractive alternative solution. Clemente et al. [7] studied an expander for an ORC cycle used to recovery heat from a regenerative gas turbine. Chacartegui et al. [8] showed a parametric optimization of a combined cycle with some industrial gas turbines and an ORC bottoming cycle in order to achieve better integration between these two technologies. Muñoz de Escalona et al. [9] presented a part-load analysis of a GT-ORC combined cycle. Carcasci and Ferraro [10, 11] studied a gas turbine cycle combined with an Organic Rankine Cycle. Del Turco et al. [12] introduces the industrial ORegen™ recovery cycle for gas turbines application (power range: 2-17 MW). Organic Rankine Cycles can be combined with a gas turbine through a diathermic oil circuit, but diathermic oil presents a temperature operation limit (about 360-380°C). Thus, using an Organic Rankine Cycle to recover the heat from gas turbine exhaust, relevant exergy losses are present: in fact ORC application is typical of low-temperature heat source.

To improve the performance, some modern gas turbines present high pressure ratios or particular configurations (like recuperative or intercooled cycles), with a low exhaust gas temperature. Furthermore, it's authors opinion that ORCs in the medium and large scale power generation have not been analyzed carefully previously. Thus, the use of ORC bottoming cycles coupled with gas turbines characterized by high efficiency but low exhaust temperature, can be an interesting solution. In an intercooled gas turbine (like GE® LMS100, with pressure ratio about 42 [13]) generally a water cycle is used to cool the air between the compressors: waste heat coming from the intercooler can be used in an Organic Rankine Cycle, due to air temperature level (about 200°C). The low-temperature heat can be converted into mechanical energy from the Organic Rankine Cycle and so the electric efficiency can increase. In the present paper, an ORC integrated with an intercooled gas turbine is studied; four different working fluids have been adopted to simulate the ORC: benzene, cyclopentane, cyclohexane and toluene. Two different plant configurations are studied, in which the waste heat are recovered from the exhaust gas or from the intercooler. Finally, a cycle analysis by varying the expander inlet pressure is presented.

Nomenclature

c	Specific Heat	[kJ/kgK]	T	Temperature	[°C]
L	Specific Work	[kJ/kg]	W	Power	[kW]
m	Massflow rate	[kg/s]	η	Efficiency	[-]
P	Pressure	[bar]	ρ	Density	[kg/m^3]
Q	Heat	[kW]			

Subscripts

air	Air	in	Inlet
amb	Ambient	lim	Limit
con	Condenser	max	Maximum
eco	Economizer	oil	Oil
el	Electric	out	Outlet
ev	Evaporator	pp	Pinch point
ex	Expander	pump	Pump
exh	Exhaust from Gas Turbine	rec	Recuperator
fan	Electrical Fan	sat	Saturation
fl	Organic Fluid	st	Stack
gb	Gearbox	sub	Subcooling
GT	Gas Turbine		

2. Working fluids

Generally for ORC applied to a gas turbine, fluids used are hydrocarbon with 5-6 carbon number [10, 11, 12]. Thus, four different dry working fluids have been tested: benzene, cyclopentane, cyclohexane and toluene. Many previous works showed that toluene [7, 8, 10, 11] is a good choice for recovering high-temperature heat, but the ORC performance decrease when the exhaust gas temperature is low. NIST (National Institute of Standards and Technology) software has been used to simulate the behavior of these working fluids. The maximum critical pressure is reached by benzene and the minimum by cyclohexane; at a fixed temperature, cyclopentane shows the highest saturation pressure and toluene the lowest, while benzene and cyclohexane have similar behaviors, as reported in Figure 1. Benzene and cyclopentane show the highest slope of saturation vapor curve, as seen in Figure 2, and that mean lower heat recovery in the recuperator.

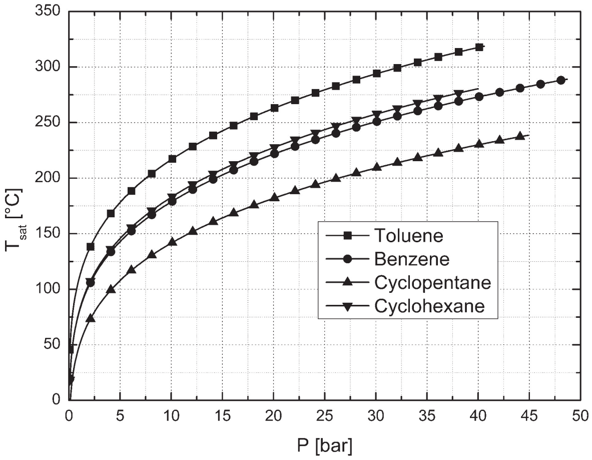

Fig. 1. Saturation curves for different fluids respect to pressure.

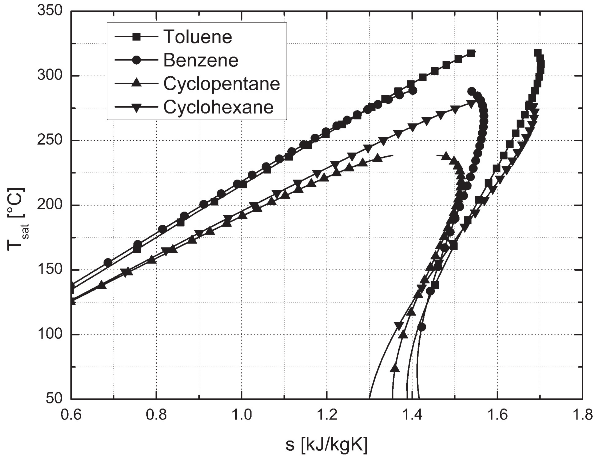

Fig. 2. Saturation curves for different fluids respect to entropy.

3. Power plant layout

The power plant considered is a combined gas turbine topping cycle and a subcritical organic Rankine bottoming cycle. Figure 3 and Figure 4 shows the two different power plant layout configurations: in Figure 3 the bottoming cycle is placed right after the power turbine, in order to recovery the exhaust heat, while in Figure 4 waste heat is recovered from the intercooler.

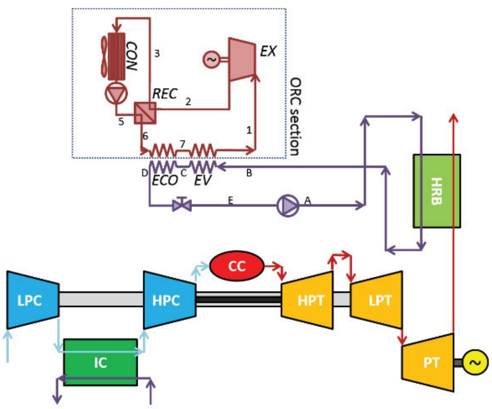

Fig. 3. Power plant layout: waste heat recovered from exhaust gas.

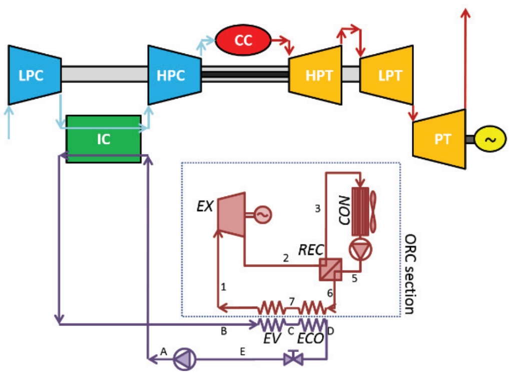

Fig. 4. Power plant layout: waste heat recovered from intercooler.

Gas turbine considered in this work is the LMS100PB^{TM} from General Electric®, which is an aero derivative three-spool turbine [13, 14]. It is one of the first turbine to develop an intercooler system aiming at reach high efficiency cycle even in a non-combined configuration. The presence of the intercooler keep low the exhaust temperature, which is good for coupling with a bottoming low-temperature cycle. Main LMS100PB^{TM} gas turbine specifications are shown in Table 1 [13, 14]. The heat transfer of the hot air from the low-pressure compressor to the organic fluid occurs through an intermediary diathermic oil circuit, interposed for safety reasons. The use of an ORC plant scheme with or without a superheater is a significant choice and depends on the selected working fluid and on the thermal source temperature. In some papers [12] the ORC cycle is shown with the presence of the superheater, while in others it's presented as a non-superheated cycle [8, 9]. Carcasci et al. [11] show that on ORC without a superheater is the best configuration using toluene, benzene and cyclohexane. They show that cyclopentane has a best behavior with superheater but temperature range analyzed in this work made this kind of choice inconvenient.

Table 1. LMS100PBTM data sheet [13, 14].

Parameter	Value	Unit
ISO Rated Power	100.0	*MW*
Heat Rate	8155	*kJ/kWh*
Electrical Efficiency	44.0	*%*
Overall Pressure Ratio	42.0	-
Exhaust Flow	220.0	*kg/s*
Exhaust Temperature	413.0	*°C*
Power Turbine Speed	3000	*RPM*
LPC Outlet Flow Rate	215.0	*kg/s*
LPC Outlet Temperature	173	*°C*
HPC Inlet Temperature	32	*°C*

Table 2. Power plant parameters used for thermodynamic analysis.

Parameter	Value	Unit	Parameter	Value	Unit
$T_{oil,max} = T_B$	380.0	*°C*	$T_{air,in}$	15.0	*°C*
$P_{ex,in} = P_1$	varied	-	$\Delta T_{air,con}$	8.0	*°C*
$T_{st,lim}$	105.0	*°C*	$\Delta T_{pp,con}$	5.0	*°C*
$\Delta T_{pp,HRB}$	5.0	*°C*	$\Delta P_{air,con}$	80	*Pa*
$\Delta T_{pp,EV}$	5.0	*°C*	$\Delta T_{pp,rec}$	15.0	*°C*
η_{ex}	0.85	-	ΔT_{sub}	30.0	*°C*
η_{gb}	0.98	-	η_{pump}	0.70	-
η_{el}	0.98	-			

As a first choice, the plant layout presents one pressure level boiler with an internal heat exchanger (recuperator REC) in order to increase the system efficiency [5]. The hot air heats the diathermic oil in the first heat recovery unit (HRB or intercooler); in the second loop, the hot oil passes through the second heat recovery unit, composed by an evaporator (EV) and an economizer (ECO), where the organic fluid is heated and enters in an expander (EX). The exhaust fluid exchanges heat in the recuperator (REC), thus it heats the condensed fluid. Finally, the organic fluid is cooled in an air condenser (CON) and pressurized in a pump. This particular type of condenser has been chosen considering the plant location a waterless area.

4. Thermodynamic and theoretical approach

When imposing the inlet expander pressure ($P_{ex;in} = P_1$), the temperature can be determined from the saturation condition: $T_1 = T_{sat}(P_1)$. On the other hand, considering the condenser, the discharge pressure at the expander exit ($P_2 = P_{con}$) can be determined: the saturated temperature of the organic fluid using ambient air temperature can be evaluated ($T_4 = T_{air,in}+\Delta T_{air,con}+\Delta T_{pp,con}$) and consequently its pressure: $P_{con} = P_{sat}(T_4)$. The fluid saturation temperature is equal for every fluid ($T_4 = 28°C$), but the condensing pressure depends on saturation curve (P_{con} = 0.15, 0.16, 0.47 and 0.06 *bar* for benzene, cyclohexane, cyclopentane and toluene, respectively). By imposing the difference of cooling air temperatures in the condenser ($\Delta T_{air,con} = T_{air,out} - T_{air,in}$), air mass flow rate is determined and imposing pressure losses ($\Delta P_{air,con}$) the power requested by the fan can be determined. Using inlet condition pressure and isentropic efficiency of expander, the specific work of the expansion can be determined. Moreover, the outlet condition of organic fluid pump can be determined. Thus, using pinch point temperature difference ($\Delta T_{air,rec}$) and energy balance in the recuperator REC, inlet conditions of condenser fluid and economizer can be determined. The no-boiling phenomena must be verified into recuperator. If this event occurs, recuperator pinch point ($\Delta T_{pp,rec}$)

must be increased. The maximum diathermic oil temperature is the minimum value between maximum oil range limit and the hot air temperature reduced of pinch point difference temperature in heat recovery boiler HRB: $T_B = min(T_{oil,lim}; T_{GT,air,in}\text{-}\Delta T_{pp,HRB})$. Thus, imposing inlet and outlet fluid condition and inlet oil temperature and using the pinch point into evaporator, the energy balance into HRSG (oil-fluid Heat Recovery Steam Generator) can be used and so the outlet diathermic oil temperature T_E and relative fluid mass flow rate (m_{fl}/m_{oil}) can be determined.

The hot air parameters (LPC temperature and flow rate, Table 1) are fixed from gas turbine performance and particularly from pressure ratio of low pressure compressor. Thus, using an energy balance into HRB, the diathermic oil mass flow rate and the exhaust air temperature can be determined. After that balance, stack temperature had to be controlled because if it's lower than the stack temperature limit, then $\Delta T_{pp,HRB}$ had to increase. In Table 2, values imposed for thermodynamic analysis are shown; it's important to notice how $T_{oil,max}$ is close to gas turbine exhaust temperature (Table 1), giving the possibility to join the two cycles together. Stack temperature is imposed limited to $105°C$ because lower values can lead to acid condensations of exhaust gases; $P_{ex,in}$ is varied in order to obtain best performances.

The authors developed an in-house code able to perform thermodynamic and design/off-design simulations of the proposed power plant. The code is developed in ANSI Standard of the *Fortran 90* programming language and the elementary energy balances were previously validated with commercial codes.

5. Results

The LMS100PB™ gas turbine presents two possibilities to recovery heat: from exhaust gas (with lower temperature than common gas turbines) and from heat exchanged by intercooler.

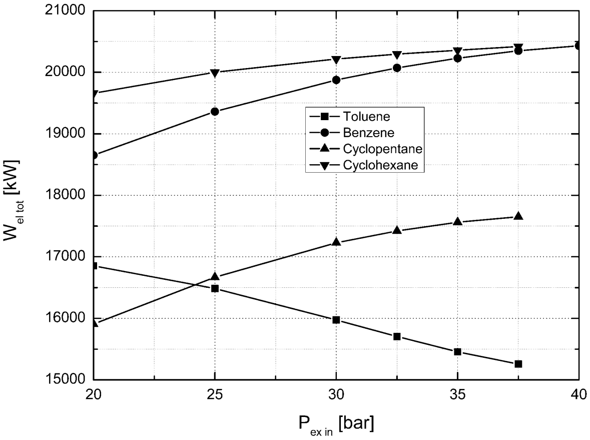

Fig. 5. Electrical power versus maximum fluid pressure.

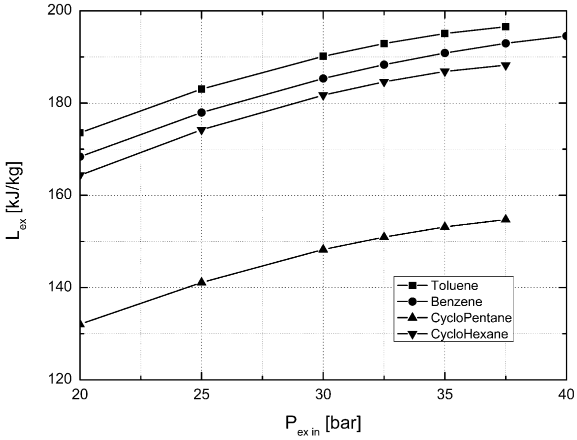

Fig. 6. Expander specific power versus maximum fluid pressure.

5.1. Heat recovery from exhaust gas

The maximum oil temperature ($T_{oil,max} = T_B = 380°C$) is imposed and it can be immediately noticed that it's a bit lower than exhaust gas temperature from gas turbine ($T_{GT,exh} = 413°C$), so the exergy losses in the heat exchange are contained. The inlet expander temperature depends on maximum fluid pressure ($T_1 = T_{sat}(P_1 = P_{ex,in})$) because the superheater is not present. The maximum fluid pressure is varied to obtain the best performance in term of output power. Figure 5 shows the net electrical power varying the maximum pressure for different organic fluids. Increasing the fluid pressure leads to a rise in output power. Benzene and cyclohexane present the maximum output power for fluid pressure at about 40 bar and 38 bar. Toluene and cyclopentane present a lower values and for toluene the trend is opposite to others fluids. The power of combined power plant is increased about 20.4 *MW* and the electrical efficiency reach 54.4%, with an increase of 10.4 percentage point respect to gas turbine simple cycle efficiency. This value is very close to that 50% indicated by Del Turco et al. [12].

The expander power is due to the contribution of the specific work of the expander and the fluid mass ow rate. The specific work of the expander increases with working fluid pressure (Figure 6), in fact the expander pressure ratio grows. The specific work of expander using cyclopentane is lower than other fluids because of highest condenser pressure (it depends on the trend of saturation pressure respect to temperature, so pressure ratio is lower).

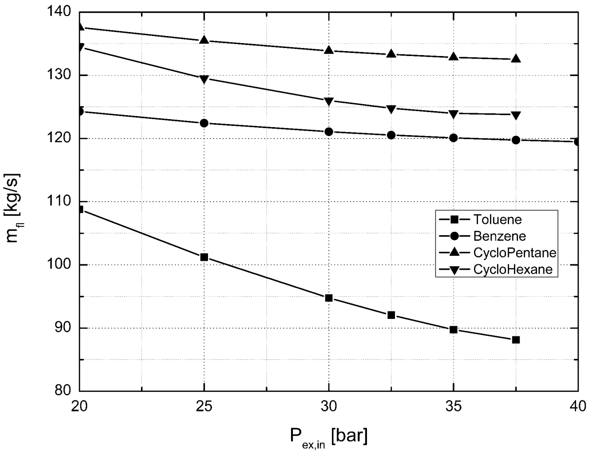

Fig. 7. Fluid mass flow rate versus maximum pressure.

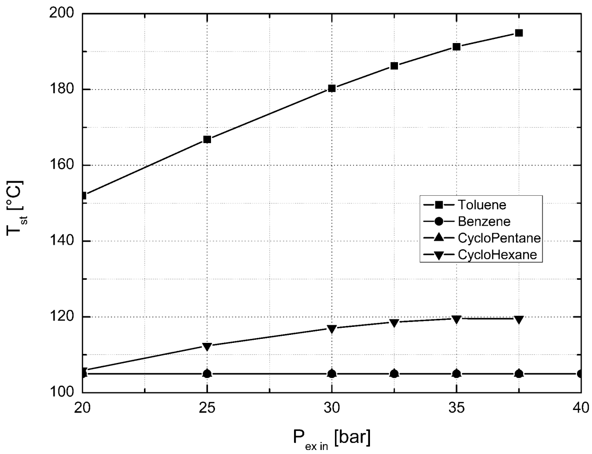

Fig. 8. Stack temperature versus maximum fluid pressure.

The organic fluid mass flow rate (Figure 7) tends to decrease with pressure. Cyclopentane presents the highest values and toluene the lowest. Imposing the pressure, saturation temperature of toluene is higher than other fluids, thus, imposing the pinch point temperature difference, the heat exchanged into evaporator is less than in the benzene case; thus the toluene mass flow rate is less than other fluids (Figure 7). In the case of toluene, the condenser pressure is greater than other fluids, so the expansion ratio of fluid into the expander is lower and its exhaust temperature is higher. Heat recovered into recuperator (REC) is higher and, consequently, the economizer inlet temperature, the return oil temperature and hot gas stack temperature are greater. In the case of benzene, stack temperature reaches the limit ($T_{st,lim} = 105°C$), so in this case, pinch point temperature difference into HRB must be increased. The stack temperature (Figure 8) is an indication of heat recovery from the GT exhaust gases. The stack temperature depends on the working conditions of the economizer, which, in turn, depends on the conditions of the recuperator. Toluene presents the highest stack temperature, so the heat is not completely recovered, while for other fluids stack temperature reach the minimum value because the return oil temperature is low and that permits to recover the maximum heat. If stack temperature limit value had been lower, than benzene might have shown better performances because of the reduced constraint values.

ORC-combined cycle performance can be compared with a traditional steam-water combined cycle: one pressure level bottoming cycle is considered for the comparison (more complex bottoming cycle can be studied, but in this case it should present same complexity of an ORC bottoming cycle). Using the same condition for the condenser, turbine and evaporator (for the steam superheater, the approach temperature difference is imposed of 40°*C*), the maximum net electric output power results in about 18.5 *MW* (for a steam pressure of 13.6 *bar*) while ORC cycle can produce 20.4 *MW* in the same conditions. A simple analysis can show that the performance of a combined cycle using ORC can be better than a one-pressure level steam-water bottoming cycle.

5.2. Heat recovery from intercooler

Another heat source available for the LMS100PB™ gas turbine is the intercooler. Intercooler permits to decrease the power requested from the compressor and it is suggested in the case of high pressure ratio. Exhaust air temperature coming from low pressure compressor is cooled before entering into the high pressure compressor. Cooling is carried out by a water circuit and then the hot water is cooled using an heat exchanger with ambient air moved by a fan. Ambient air mass flow rate can be evaluated using energy balance, obtaining $m_{air,con}$ = 3815 *kg/s*, and a requested electrical power of the fan of 303 *kW* ($W_{air,fan} = m_{air,con} \cdot (\Delta P_{air,con}/\rho_{air,amb})$).

An ORC power plant can substitute completely or partially the cooling system, as shown in Figure 4. Figure 9 shows ORC electric output power using different organic fluid and toluene shows the lowest value (toluene is confirmed as a good organic fluid for relative high temperature heat source). The curves for others fluids present a maximum value at about the same output power (about 2.2 *MW*, while electric efficiency increases of 2.2 percentage point), but with different pressure: the best inlet expander pressure for benzene and cyclohexane is about 1.7 *bar* and 4.1 *bar* for cyclopentane. The thermodynamic efficiency of ORC cycle is low (about 7.2%) due to the low sources temperature. The fluid mass flow rate decreases when the inlet expander pressure increases (Figure 10). Using toluene the mass flow rate is the lowest and using cyclopentane the fluid mass flow rate is the greatest.

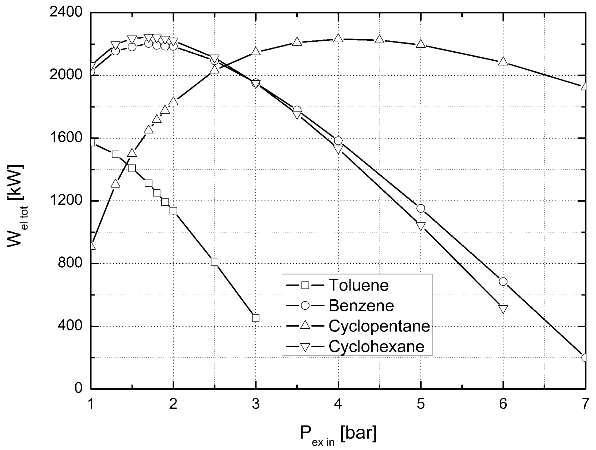

Fig. 9. Electric output power of ORC using intercooling heat source.

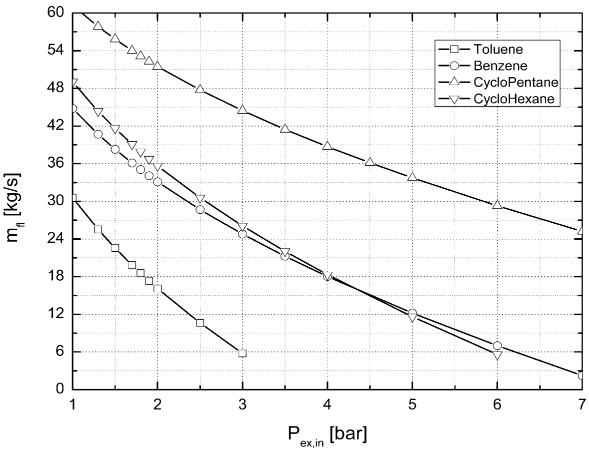

Fig. 10. Fluid mass flow rate versus maximum pressure using intercooling heat source.

In the opposite, increasing the maximum fluid pressure lead to an increase of ORC expander specific work (Figure 11) because the pressure ratio increases and so the enthalpy difference increases too. Benzene, cyclohexane and toluene are not very different, but toluene presents the lowest value in term of mass flow rate (Figure 10). The specific work using cyclopentane is the lowest (Figure 11), but the mass flow rate is the greatest (Figure 10).

Figure 12 shows the inlet HPC air temperature (the exhaust air temperature from the heat recovery boiler air-oil): it is very high for toluene; in fact, in this case the heat recovery is low and so the output power (Figure 9). Using cyclopentane, the lowest air exhaust temperature is obtained. However, considering the value corresponding to the pressure that optimize output power, the air exhaust temperature is about 90°*C* for all fluids except toluene. Therefore, a supplementary small-size intercooler is necessary to reach the target air temperature of 32°*C* (Table 2). In fact, a fraction of heat which must be removed is converted into electric power and another part is burn off into air condenser of ORC, so the small size intercooler need less ambient air and so the fan power absorbed is about 124 *kW* (versus 303 *kW* of standard layout). The recuperator can maybe be eliminated from the cycle, so the air entering the intercooler will be lower, but ORC condenser will have to exchange an increased amount of heat and the fan will have to work more than before: for that reason the final effect on the cycle will be the same.

6. Conclusions

An Organic Rankine Cycle can be a good solution for heat recovery if combined with an intercooled, high pressure ratio, gas turbine cycle. In this paper different organic fluids based on hydrocarbon are compared in two different configurations: heat recovery from gas turbine exhaust or from intercooler. Using benzene and cyclohexane, the power of combined power plant can be increased of about 20.4 *MW* and the electric efficiency of

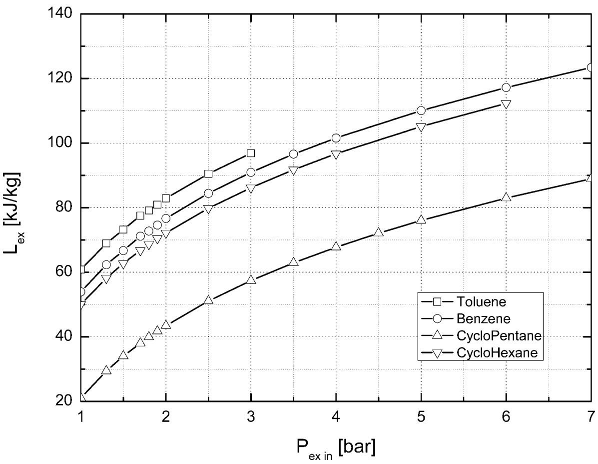

Fig. 11. Expander specific work versus maximum pressure using intercooling heat source

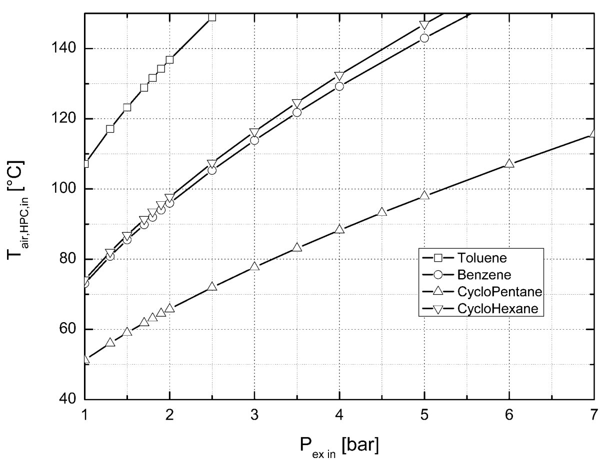

Fig. 12. Inlet High Pressure Compressor temperature versus maximum pressure using intercooling heat source

the combined plant can reach 54.4% (10.4 percentage point more than simple gas turbine cycle efficiency); results show that toluene and cyclopentane are not the right fluid choice for this plant configuration.

Using the ORC linked to the intercooler can lead to an electrical power output of 2.2 MW, and also in this layout, toluene is the worst fluid that can be used. The use of dry organic fluid doesn't allow to cool completely the air between the low pressure and high pressure compressors and so the intercooling system cannot be removed: anyway it can be reduced in the size. It's important to notice that intercooler in a gas turbine cycle can also add flexibility to part load management. While the ORCs are not a good solution for a classical gas turbine in term of thermodynamic efficiency due to high exergy losses into heat recovery boiler, they can be a good alternative to a steam-water bottoming cycle for high pressure ratio gas turbine, also with advantage in term of thermodynamic efficiency.

References

[1] C. Carcasci, F. Costanzi, B. Pacifici, Performance analysis in off-design condition of gas turbine air-bottoming combined system, Energy Procedia, Vol.45 (2014), pp. 1037-1046.

[2] M. Nadir, A. Ghenaiet, C. Carcasci, Thermo-economic optimization of heat recovery steam generator for a range of gas turbine exhaust temperatures, Applied Thermal Engineering, Vol.106 (2016), pp. 811-826.

[3] O. Badr, P. W. Ocallaghan, Rankine-cycle systems for harnessing power from low-grade energy-sources, Applied Energy, Vol. 36(4) (1990), pp.263-292.

[4] Y. P. Dai, J. F. Wang, Parametric optimization and comparative study of Organic Rankine Cycle (ORC) for low grade waste heat recovery, Energy Conversion Management, Vol. 50(3) (2009) , pp. 576-582.

[5] B. Saleh, G. Koglbauer, M. Wendland, J. Fischer, Working fluids for low-temperature ORC, Energy , Vol. 32(7) (2009) , pp. 1210-1221.

[6] C. He, C. Liu, H. Gao, H. Xie, Y. Li, S. Wu, J. Xu, The optimal evaporation temperature and working fluids for subcritical Organic Rankine Cycle, Energy, Vol. 38(1) (2012) , pp. 136-143.

[7] 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, Vol. 106 (2013), pp. 355-364.

[8] R. Chacartegui, D. Sánchez, J. M. Muñoz de Escalona, T. Sánchez, Alternative ORC bottoming cycles for combined cycle power plants, Applied Energy , Vol. 86(10) (2009) , pp. 2162-2170.

[9] J. M. Muñoz de Escalona, D. Sánchez, R. Chacartegui, T. Sánchez, Part load analysis of gas turbine & ORC combined cycles, Applied Thermal Energy, Vol. 36 (2012) , pp. 63-72.

[10] C. Carcasci, R. Ferraro, Thermodynamic optimization and off-design performance analysis of a toluene based Rankine cycle for waste heat recovery from medium size gas turbines, Proc. ASME Gas Turbine India Conference, 2012, GTIndia2012-9645.

[11] C. Carcasci, R. Ferraro, E. Miliotti, Thermodynamic analysis of an Organic Rankine Cycle for waste heat recovery from gas turbines, Energy, Vol. 65 (2014), pp. 91-100.

[12] P. Del Turco, A. Asti, A. S. Del Greco, A. Bacci, G. Landi, G. Seghi, The ORegen. waste heat recovery cycle: reducing the CO2 footprint by means of overall cycle efficiency improvement, Proc. ASME Turbo Expo 2011 , GT2011-45051.

[13] M. J. Reale, New high efficiency simple cycle gas turbine - GE's LMS100, General Electric® Report, 2004 , GER-4222A (06/04).

[14] AA. VV., LMS100. Flexible Power, General Electric® Report, 2006, 353 GEA-14355A (09/06).