Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 123 (2017) 289–296

Energy Procedia

www.elsevier.com/locate/procedia

1st International Conference on Sustainable Energy and Resource Use in Food Chains, ICSEF 2017, 19-20 April 2017, Berkshire, UK

Low grade thermal recovery based on trilateral flash cycles using recent pure fluids and mixtures

Roberto Cipollone[a], Giuseppe Bianchi[a,b]*, Marco Di Bartolomeo[a], Davide Di Battista[a], Fabio Fatigati[a]

[a]*University of L'Aquila, Via Giovanni Gronchi 18, L'Aquila 67100, Italy*
[b]*Brunel University London, Institute of Energy Futures, Center for Sustainable Energy Use in Food Chains, Uxbridge, Middlesex UB8 3PH, UK*

Abstract

The current work presents a thermodynamic analysis of a Trilateral Flash Cycle (TFC) system for low grade heat to power conversion applications. Novel aspects of the research are the usage of rotary positive displacement expanders as prime movers of the TFC system as well as the reference to working fluids and their mixtures at the state of the art. In particular, the role of a correct built-in volume ratio of the expander with respect to the pressure ratio of the thermodynamic cycle is emphasized. In fact, a mismatching of these two quantities would lead to an isochoric expansion process which, in turn, might negatively affect the overall power recovery. With reference to a transcritical CO_2 stream at 100°C as heat source for the TFC system, parametric and screening studies were carried out using different expander built-in volume ratios and working fluids respectively. Among the fluids analyzed, results showed that pure substances such as R1234ze(E) and propane would provide a greater specific work but, on the other hand, would require built-in volume ratios (8 and 14) that are beyond the capabilities of rotary positive displacement expanders (5). The addition of CO_2 to the afore mentioned working fluids would ease the mismatching issue but, at the same time, would reduce the specific power output. Regarding the built-in volume ratio analysis, it was found that optimal values change in accordance to the working fluid and refer to an expansion process with a slight isochoric phase.

Peer-review under responsibility of the scientific committee of the 1st International Conference on Sustainable Energy and Resource Use in Food Chains.

Keywords: trilateral flash cycle; positive displacement expander; refrigeration; waste heat recovery; thermodynamic analysis

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

1876-6102 © 2017 The Authors. Published by Elsevier Ltd.
Peer-review under responsibility of the scientific committee of the 1st International Conference on Sustainable Energy and Resource Use in Food Chains.
10.1016/j.egypro.2017.07.246

Nomenclature

h	specific enthalpy [J/kg]	β	volume ratio [-]
$\dot{m}$	mass flow rate [kg/s]	**subscripts**	
p	pressure [Pa]	d	discharge
P	power [W]	id	ideal, target
v	specific volume [m^3/kg]	in	intake
$\dot{V}$	volume flow rate [m^3/s]	hs	heat source

1. Introduction

Low and medium grade heat sources demonstrate a huge energy recovery potential into mechanical and electrical forms. They are represented by heat usually wasted into atmosphere in many industrial applications as well as in the transportation and renewable power sectors. Most of the available heat recovery potential falls in a temperature range of the waste heat source below 100 °C [1]. In these conditions, conventional waste heat to power conversion systems based on bottoming Organic Rankine Cycles (ORC) are challenging to operate. In fact, the condition imposed by the pinch point at the evaporator produces a strong irreversibility during the heat transfer between heat source and working fluid, which reduces the work of the cycle and, in turn, its efficiency [2-4]. A more suitable bottoming thermodynamic cycle to recover and transform heat into mechanical energy is certainly represented by the so-called trilateral cycle, (TLC) [3].

The TLC employs the same components than an ORC. Main difference is that the working fluid does not undergo to a phase change during the heat recovery but only reaches saturated liquid conditions. On the other hand, the expansion of the working fluid from such saturation state produces a sudden phase change, referred as a flashing process: for this reason, in literature TLC is often called also Trilateral Flash Cycle (TFC). TLC or TFC have been extensively studied in literature: from a thermodynamic point of view, main benefit is the thermal matching between heat source and working fluid [6] that minimizes the irreversibility produced during the heat transfer (exergy destruction) [7]. In reference [8], a dual loop waste heat recovery circuit, based on the integration of an upper trilateral cycle and a lower organic Rankine cycle, was proposed for an internal combustion engine application: maximum recovery efficiency of 10.9% and exergy efficiency of 58.8%, larger than that of the single loop trilateral cycle, were achieved. A generalization of the TLC, named Power Flash Cycle (PFC), was investigated in reference [9]; in particular, a PFC is a cycle where the compressed liquid delivers power performing a flash expansion. An additional cycle that has been examined is the Organic Flash Cycle (OFC), whereas the working fluid is first flashed in a two-phase mixture and subsequently the saturated vapor is separated and expanded through the high pressure turbine, while the saturated liquid is throttled to the same pressure [10]. This enhancement would yield approximately 20% greater power output than an optimized ORC. Moreover, comparing the performances of TLC with ORC and Kalina Cycle system, it has been demonstrated that TLC can achieve a net power output higher than those provided by the ORC and Kalina cycle, but this is greatly affected by the expander isentropic efficiency [11].

TLC fluid selection is a key point: water is unsuitable due volume ratios required for its expansion. Organic fluids are the best candidates: a TFC with R12 as working fluid demonstrated a high efficiency, also in trans-critical operation, [7]. Different aromatic hydrocarbon and siloxanes were also proposed as working fluids for TFCs demonstrating performances comparable with those of an optimized ORC, [12]. The use of mixtures as working fluid was proposed in trilateral cycle since the temperature glide during condensation better matches the temperature profile of the cooling fluid [13].

The availability of new fluids having a reduced GWP and ODP which have been considered in the literature [14] offers a new room of investigation concerning the potentialities as pure fluids and as mixtures: this possibility remained has been not fully exploited yet and deserves further attention. CO_2, for instance, is used as working fluid in refrigeration units at supercritical state and a similar interest could be expected as working fluid, pure or mixed with more modern CFC fluids.

Moreover, the technology of the expansion represents a very important issue also for the consequences on the thermodynamic cycle [15]. Screw expanders are capable of adiabatic efficiencies greater than 70%, admitting saturated or slightly sub-cooled liquid and expelling wet vapor. The problem of the high expansion ratios requested to match maximum and minimum cycle pressures remains as a problem: attempts in literature have been studied considering multiple in series expanders whose last one is a dynamic machine able to reach (in any case) the condenser pressure [12]. In fact, when the expansion is done inside a positive displacement machine (also referred as a volumetric machine) which is more suitable for reduced mechanical power (<10 kW), a real complete adiabatic expansion cannot be sustained [16, 17]. The built-in volume ratio of the machine allows that the first part of the expansion can behave as an adiabatic transformation but the remaining part (most part, in reality) behaves like an isochoric one, with a sensible reduction of the work produced and with a significant modification of the cycle which is not anymore triangular.

In this paper, the Authors present a theoretical analysis of the TFC applied to an upper thermal source operating in a low temperature range: in the specific case, this source is represented by a transcritical stream of CO_2 exiting from a compressor of a refrigeration unit. Pure fluids at the state of the art as well as their mixtures have been considered as working fluids. Furthermore, with reference to a positive displacement machine, expansion ratios have been investigated in order to match the expander built-in volume ratio with the cycle pressure ratio. The actual two stages of the expansion process (adiabatic and isochoric) that may occur in a positive displacement machine operating in under-expansion conditions are eventually presented.

2. Cycle modeling

In this work, a transcritical flow of 1 kg/s of CO_2 at 90 bar and 100°C has been considered as heat source of the Trilateral Flash Cycle system. This stream might be representative of the cooling process that occurs in a gas cooler of a refrigeration plant. In fact, the CO_2 cooling is usually done through an air cooled heat exchanger, which brings the refrigerant temperature almost to ambient conditions. However, rather than being rejected to the environment, this thermal power could be recovered and transformed into mechanical (and electrical) forms.

The thermodynamic analysis has been carried out through the modeling platform developed in [18] that is able to calculate the maximum mechanical power recoverable for a given heat source. In particular, with reference to a condensing temperature of the heat sink (water at 40°C in the current case) and for a given working fluid, the thermodynamic procedure calculates maximum pressure, corresponding saturation temperature and working fluid mass flow rate using energy balances at the heat exchangers and isentropic efficiency definitions for pump and expander. For instance, the energy balance at the heater that is reported in Eqn.1 allows to calculate the working fluid mass flow rate for the TFC system schematized in Figure 1. Real fluid properties are provided by the coupling of the governing equations to the NIST database.

$$\dot{m} = \frac{\dot{m}_{hs}(h_{hs,in} - h_{hs,out})}{h_3 - h_2} \tag{1}$$

Outputs of the model are thermal cycle efficiency, heat exchangers efficiency and net power output. The latter quantity is calculated as in Eqn. 2.

$$P = \dot{m}\left[(h_3 - h_4) - (h_2 - h_1)\right] \tag{2}$$

Additional input parameters that have been considered for the current analysis are a pump isentropic efficiency of 80%, an expander isentropic efficiency of 70% and a pinch-point temperature of 2K at the heat exchangers.

3. Results and discussion

3.1. Parametric analysis on R1234ze(E)

Figure 2 shows the T-s diagram of the TFC system using R1234ze(E). The chart also includes heat source and heat sink curves. Specific entropy is referred to 1 kg of CO_2. Due to the shape of the isobaric heat transfer curves of water and carbon dioxide, pinch point happens inside the heater and at the outlet section (water out) of the condenser. Figure 2 applies only if the expansion fully happens inside the machine, without any fluid discharge during the expansion. From a technological point of view, this refers either to a dynamic machine or to a positive displacement one in which the pressure at end at the closed volume expansion phase is equal to the one at the condenser.

The operating point of the TFC represented in Figure 2 is the one that provides maximum power. To identify this configuration, a parametric analysis acting on cycle pressure ratio has been carried out and it is reported in Figure 3. In particular, Figure 3 shows the sensitivity to maximum cycle pressure of working fluid mass flow rate and net specific work. Specific work increases because the increase of expansion work is greater than the increase of the pumping work. On the other hand, working fluid mass flow rate decreases since higher pressures imply higher saturation temperatures and, in turn, a greater heat gain per unit mass that is however fulfilled by the same upper thermal source.

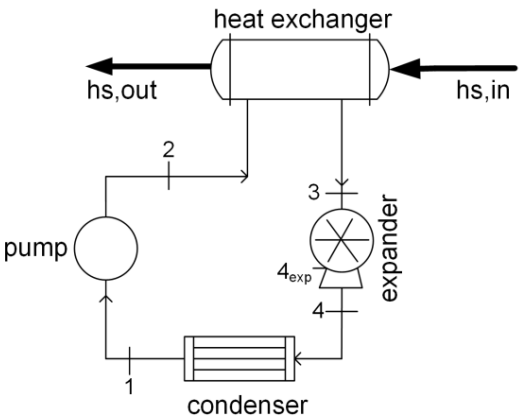

Fig. 1. TFC plant scheme

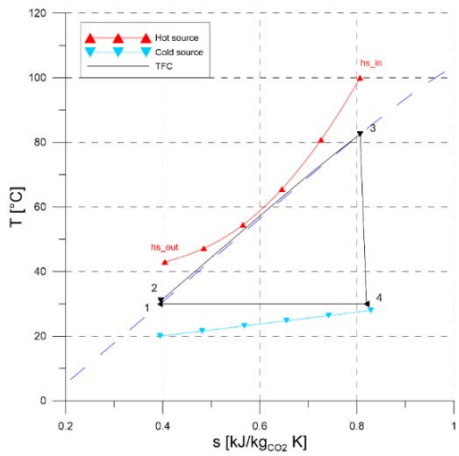

Fig. 2. T-s diagram - R1234ze(E) at maximum power configuration

3.2. Working fluid screening

A set of pure fluids and mixtures has been shortlisted after a preliminary screening procedure based on thermodynamic, environmental and operational criteria. Similarly to Figure 2, with reference to the cycle pressure ratio that maximizes the net power output, Figure 4 reports the net specific work and working fluid flow rate for the fluids and mixtures considered: except for propane, pure fluids have higher flow rates than mixtures due to a lower specific heat gain.

On the other hand, when R1234ze(E) and propane are mixed with CO_2, the net specific work of the TFC decreases. Thermal cycle efficiency, on the contrary, is sensibly higher for pure fluids (Figure 5): about 5% for pure fluids and in the range between 2-3% for mixtures.

In terms of power recovery, pure fluids range between 6.0 kW and 6.5 kW while mixtures around 2.5 kW and 3.0 kW (Figure 6): this is strongly due to the gap found in the thermodynamic efficiency of the cycle which decreases if the condensation phase is not isothermal. Figure 7 shows the situation that happens when CO_2/Propane 30% /70% mixture is considered: expansion is limited by the slope of the condensation phase reaching point 1.

Moreover, the glide related to the condensation of the mixture has a positive effect to lower the volumetric ratio of the expansion calculated as the ratio of specific volume at point 3 and the one at point 4. Figure 8 reports this expansion ratio for the fluids considered: pure fluids require higher expansion ratios (10-14) with respect to mixtures (2-4). This

is a very significant issue, because when a positive displacement machine is chosen as expander, the so-called built-in volume ratio sets the nature of the expansion.

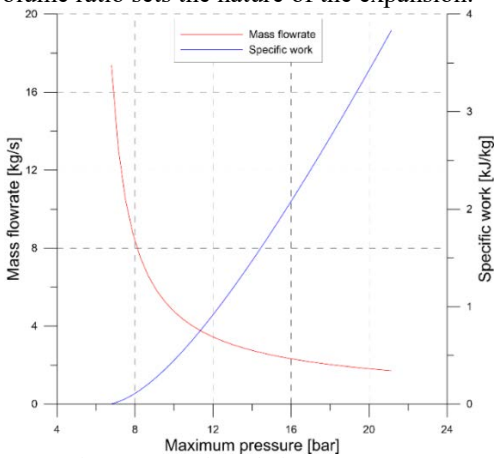

Fig. 3. Parametric analysis on R1234ze(E)

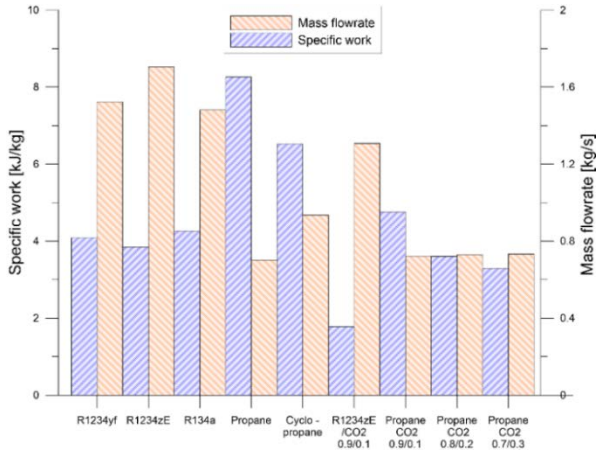

Fig. 4. Specific power output and working fluid flow rate at maximum power conditions

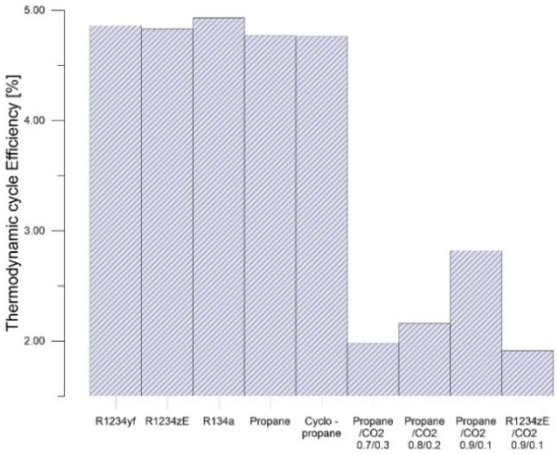

Fig. 5. Cycle thermal efficiency

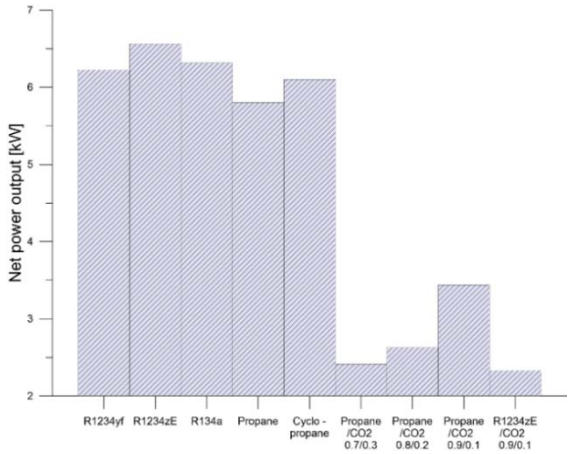

Fig. 6. Maximum power output

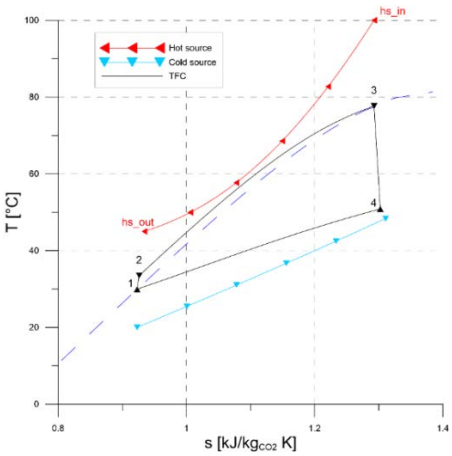

Fig. 7. TFC using a mixture as working fluid (CO_2/Propane 30/70%)

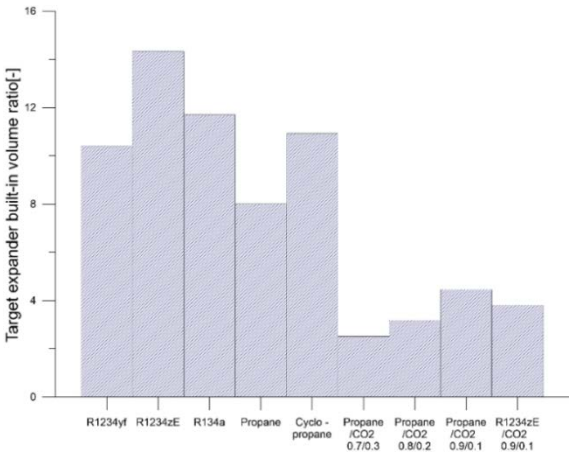

Fig. 8. Target expander built-in volume ratio to match the TFC pressure ratio

3.3. Real expansion of a positive displacement expander

The consideration that the expansion in a TFC completely happens in the liquid/vapor equilibrium (with a greater presence of the liquid with respect to the vapor) invites to consider with greater attention volumetric machines instead of dynamic ones. Volumetric machines have an inherent disadvantage represented by the built-in volume ratio which influences the nature of the expansion. It has been already observed that as long as the fluid remains inside the machine the transformation can be considered as a real adiabatic followed by an isochoric transformation until the matching with the condenser pressure. On the other hand, they have a greater flexibility to accept liquid-vapor mixtures as well as the advantage to match with mechanical power less than 10 kW and rotational speeds that cope with low cost and reliable electrical generators.

Considering these aspects mandatory, the problem of the maximum achievable built-in volume ratio deserves some attention. Pure fluids would require higher built-in volume ratios with respect to mixtures. Propane and R1234ze(E) require volume ratios close to 8 and 14 with associated power recovered equal to 6 kW and 7 kW respectively. In the case of mixtures, the target built-in volume ratios decrease: for Propane/CO_2 (70%-30%) and for Propane/CO_2 (90%-10%), the maximum ratios are 2.52 and 4.46 with a power recovered equal to 2.76 kW and 3.68 kW, respectively. Considering that the built-in volume ratio depends on the machine geometry, it cannot be taken for granted that the values calculated can be achieved inside a machine due to geometrical constraints. When this happens, the pressure inside the cell just before the discharge port opening is greater than the pressure at the condenser: a sudden outflow is produced which modifies immediately the pressure inside the cell. Specific volumes inside can be calculated according to an adiabatic expansion till to the condenser pressure and this allows the calculation of the mass remaining inside the vane before the forced discharge occurs. The forced discharge proceeds at constant pressure, matching the vane's volume reduction. The residual mass inside the vane is subsequently compressed when the discharge port is closed: its mixing with the incoming fluid when the inlet port opens restores the initial point from which the expansion proceeded.

A power reduction with respect to Eqn. 2 is produced and it results as in Eqn. 3:

$$\Delta P = \int_{\dot{V}_d}^{\dot{V}_{d,id}} \left(p - p_1\right) d\dot{V} \tag{3}$$

being $\dot{V}_d$ the volume flow rate when the discharge port opens and $\dot{V}_{d,id}$ the one that would be needed following the real expansion till to the condenser pressure p_1. Assuming an ideal volumetric efficiency, the intake volume flow rate that corresponds to a given mass flow rate $\dot{m}$ can be calculated according to Eqn. (4). On the other hand, the actual discharge volume flow rate depends on the built-in volume ratio and on the volume flow rate at the beginning of the expansion, as in Eqn (5). Finally, the ideal discharge volume flow rate depends on the inlet volume flow rate and the target built-in volume ratio (Figure 8); alternatively, one could use the mass flow rate and the specific volume v at point 4, as in Eqn (6).

$$\dot{V}_{in} = \dot{m} v_3 \tag{4}$$

$$\dot{V}_d = \beta \dot{V}_{in} \tag{5}$$

$$\dot{V}_{d,id} = \beta_{id} \dot{V}_{in} = \dot{m} v_4 \tag{6}$$

In Figure 9 and 10 the power is reported vs. the built-in volume ratio: the first one refers to mixtures, the second one to pure fluids. It is evident how when using mixtures, the fluid can fully expand inside a closed vane, reaching the maximum net power output. Indeed, in these cases the target volume built-in ratio would be lower than 4.5, i.e. compatible with positive displacement machines. For pure fluids, on the contrary, this is not possible: the highest values of the power would require built-in volume ratio greater than 8 and this is achievable only with multiple machines in series. If maximum reachable built-in volume ratio is 4.5, the maximum power recoverable is of the order of 4 kW if propane is used while close to 3 kW when R1234yf or R134a are considered. In this situation, the mixture propane/CO2 (0.9/0.1) gives the same net power, while others mixtures allow a net power close to 2.5 kW with lower more suitable built in volume ratios.

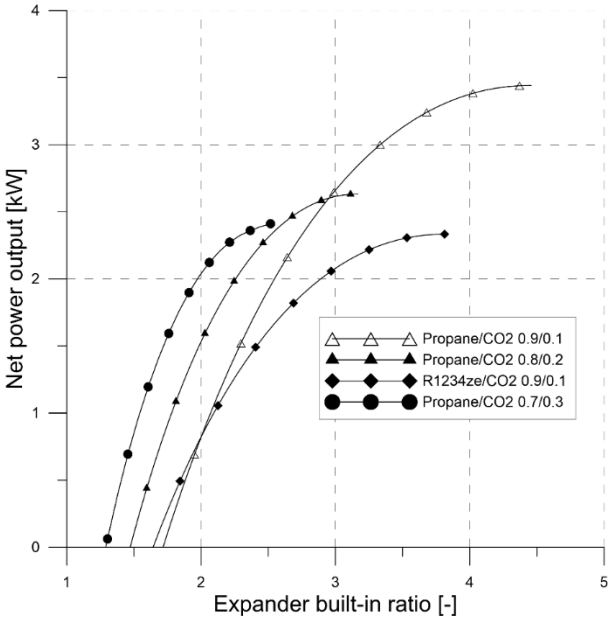

Fig. 9. Mechanical power vs expander built-in volume ratio (mixtures)

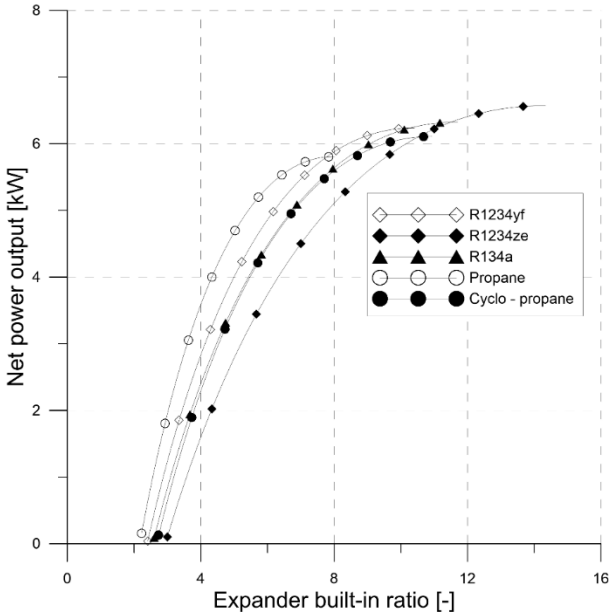

Fig. 10. Net power output vs expander built-in volume ratio (pure fluids)

4. Conclusions

TFC-based power units represent a viable way to recover low grade thermal energy into mechanical work when the higher temperature of the available thermal source is below 100°C. Considering the working fluids potentially usable pure or in mixtures, the feasibility of the power unit strongly depends on the expander technology which limits the power recovered and also affects the shape of the cycle, from triangular (originally conceived) to a trapezoidal type. In fact, in the mechanical power range lower than 10 kW, only volumetric machines are allowed. In this case, the expansion (usually considered as a real adiabatic) performs according to two thermodynamic transformation: the first part (happening inside the machine when the cell is a closed volume) is a real adiabatic, the second part is isochoric (happening suddenly inside the vane when it opens toward the discharge port). The study considered a thermal source represented by a trans-critical CO_2 whose temperature decreases from 100 °C to 40 °C. Considering as pure working fluids R1234yf, R1234zE, R134a, Propane, Cyclo-propane and as mixtures propane/CO_2 having 70 %, 80%, 90 % in propane as well as R1234ze(E) (90 %) with CO_2,when the cooling fluid is water at an inlet temperature equal to 20 °C, the following conclusions apply:

a) Most interesting fluid is represented by R1234ze(E) which guarantees a greater power output equal to 6.5 kW per unit mass flow rate of CO_2; maximum pressure is 21.2 bar when saturated temperature is 82.5 °C. This datum refers to a real adiabatic expansion from 21.2 bar to 5.8 bar; the correspondent volume ratio is 14.3 greater than the maximum allowable built in volume ratio inside existing rotary volumetric machines. Mixtures produce a significantly lower mechanical power recovered per unit mass flow rate of CO_2 (2.5-3.5 kW): the corresponding required built-in volume ratios are lower (3.5-4.0) fully compatible with the volumetric rotary machines;
b) Considering the limits imposed by the built in volume ratios (4.5), when a pure fluid is used, a maximum power equal to 4.5 kW can be achieved with propane, while if R1234yf is considered, the power reduces to 3 kW. The use of the mixtures considered does not produce a power increase; they appear more suitable for rotary volumetric machines having lower built in volume ratios.

The reference made a trans-critical CO_2 is related to the opportunity to recover the energy instead of considering a traditional gas cooler in the refrigeration units. A net electrical saving of the order of 2.5-3.0 % can be achieved.

Acknowledgement

EPTA Refrigeration Group is highly acknowledged for having funded this research.

References

[1] Clemens Forman, Ibrahim Kolawole Muritala, Robert Pardemann, Bernd Meyer, Estimating the global waste heat potential, Renewable and Sustainable Energy Reviews, Volume 57, May 2016, Pages 1568-1579, ISSN 1364-0321, http://dx.doi.org/10.1016/j.rser.2015.12.192.

[2] Sylvain Quoilin, Martijn Van Den Broek, Sébastien Declaye, Pierre Dewallef, Vincent Lemort, Techno-economic survey of Organic Rankine Cycle (ORC) systems, Renewable and Sustainable Energy Reviews, Volume 22, June 2013, Pages 168-186, ISSN 1364-0321, DOI: 10.1016/j.rser.2013.01.028.

[3] D. Di Battista, M. Mauriello, R. Cipollone, Waste heat recovery of an ORC-based power unit in a turbocharged diesel engine propelling a light duty vehicle, Applied Energy, Volume 152, 15 August 2015, Pages 109-120, ISSN 0306-2619, DOI: 10.1016/j.apenergy.2015.04.088.

[4] R. Cipollone et al., Development of an Organic Rankine Cycle system for exhaust energy recovery in internal combustion engines, Journal of Physics: Conference Series, Volume 655, 2015, pp. 012015, DOI:10.1088/1742-6596/655/1/012015

[5] H.A. Ajimotokan, I. Sher, Thermodynamic performance simulation and design optimisation of trilateral-cycle engines for waste heat recovery-to-power generation, Applied Energy, Volume 154, 15 September 2015, Pages 26-34, ISSN 0306-2619, DOI: 10.1016/j.apenergy.2015.04.095.

[6] Zamfirescu C, Dincer I. Thermodynamic analysis of a novel ammonia–water trilateral Rankine cycle. Thermo-chimica Acta 2008;477:7–15. DOI: 10.1016/j.tca.2008.08.002.

[7] Smith,I.,Stosic,N.,Colin,A. TRILATERAL FLASH CYCLE SYSTEM A HIGH EFFICIENCY POWER PLANT FOR LIQUID RESOURCES. City University, London, OHB, U.K.

[8] Byung Chul Choi, Young Min Kim, Thermodynamic analysis of a dual loop heat recovery system with trilateral cycle applied to exhaust gases of internal combustion engine for propulsion of the 6800 TEU container ship, Energy, Volume 58, 1 September 2013, Pages 404-416, ISSN 0360-5442, DOI: 10.1016/j.energy.2013.05.017.

[9] Ngoc Anh Lai, Johann Fischer, Efficiencies of power flash cycles, Energy, Volume 44, Issue 1, August 2012, Pages 1017-1027, ISSN 0360-5442, DOI: 10.1016/j.energy.2012.04.046.

[10] Smith,I.,K. Development of the Trilateral Flash Cycle System: Part 1: Fundamental Considerations. Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy. 207-3-179-194-1993. 10.1243/PIME_PROC_1993_207_032_02

[11] M. Yari, A.S. Mehr, V. Zare, S.M.S. Mahmoudi, M.A. Rosen, Exergoeconomic comparison of TLC (trilateral Rankine cycle), ORC (organic Rankine cycle) and Kalina cycle using a low grade heat source, Energy, Volume 83, 1 April 2015, Pages 712-722, ISSN 0360-5442, DOI: 10.1016/j.energy.2015.02.080.

[12] Tony Ho, Samuel S. Mao, Ralph Greif, Comparison of the Organic Flash Cycle (OFC) to other advanced vapor cycles for intermediate and high temperature waste heat reclamation and solar thermal energy, Energy, Volume 42, Issue 1, June 2012, Pages 213-223, ISSN 0360-5442, DOI: 10.1016/j.energy.2012.03.067.

[13] Smith, I., K., da Silva, R Pitanga Marques. Development of the Trilateral Flash Cycle System Part 2: Increasing Power Output with Working Fluid Mixtures. 1994/05/01. Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy. 135-208-2- SAGE Publications. DOI: 10.1243/PIME_PROC_1994_208_022_02

[14] Davide Di Battista, Roberto Cipollone, High efficiency air conditioning model based analysis for the automotive sector, International Journal of Refrigeration, Volume 64, April 2016, Pages 108-122, ISSN 0140-7007, DOI: 10.1016/j.ijrefrig.2015.12.014

[15] Johann Fischer, Comparison of trilateral cycles and organic Rankine cycles, Energy, Volume 36, Issue 10, October 2011, Pages 6208-6219, ISSN 0360-5442, DOI: 10.1016/j.energy.2011.07.041.

[16] Cipollone R, Bianchi G, Murgia S, Contaldi G. Energy recovery using sliding vane rotary expanders. In: Proceedings of the 8th International Conference on Compressors and their Systems at London. ISBN 978-1-78242-169-6; 2013, p. 183ţ194, DOI: 10.1533/9781782421702.3.183

[17] Roberto Cipollone, Giuseppe Bianchi, Davide Di Battista, Giulio Contaldi, Stefano Murgia, Mechanical Energy Recovery from Low Grade Thermal Energy Sources, Energy Procedia, Volume 45, 2014, Pages 121-130, ISSN 1876-6102, DOI: 10.1016/j.egypro.2014.01.014

[18] D. Di Battista, R. Cipollone, C. Villante, C. Fornari, M. Mauriello, The Potential of Mixtures of Pure Fluids in ORC-based Power Units fed by Exhaust Gases in Internal Combustion Engines, Energy Procedia, Volume 101, 2016, Pages 1264-1271, ISSN 1876-6102, DOI: 10.1016/j.egypro.2016.11.142.