

Performance Comparison of Different Rankine Cycle Technologies Applied to Low and Medium Temperature Industrial Surplus Heat Scenarios

Goran Durakovic^{1*}, Monika Nikolaisen²

SINTEF Energy Research, Gas technology,
Trondheim, Norway

¹goran.durakovic@sintef.no

²monika.nikolaisen@sintef.no

* Corresponding Author

ABSTRACT

Energy intensive industries generate vast amounts of low-grade surplus heat, and potential for direct re-use can be limited due to insufficient local demand. Conversion of heat into electric power thus becomes an attractive option. However, heat-to-power conversion is limited by poor efficiencies for heat sources at low and medium temperatures. To optimally utilize the available energy, the choice of technology could be important.

In this work, single-stage subcritical, dual-stage subcritical and single-stage transcritical Rankine cycle technologies were compared. The investigated heat sources are an air equivalent off-gas at 150°C and 200°C and a liquid heat source at 150°C. These were all evaluated with the same available duty. The Rankine cycles were compared with equal total heat exchanger areas to enable fair comparisons of different cycles, instead of comparing cycles based on equal pinch point temperature differences as is frequently reported in literature. Six working fluids were investigated for both heat source temperatures, and cycle optimization was performed to determine the best Rankine cycle for each working fluid. This means that the optimizer could choose between single-stage and dual-stage cycle during one optimization, and determine which gave the highest net power output. The results were benchmarked against single-stage subcritical Rankine cycles.

The results showed that the cycle yielding the highest net power varied depending on the allowable total heat exchanger area. When the total heat exchanger area was large, the dual-stage and transcritical cycles performed better than the benchmark, and the improvement became larger as total area increased. For the total heat exchanger areas investigated for the gaseous heat sources, the maximum performance increase was 3 %. Below a certain total area threshold, the benchmark performed the best.

For the liquid heat source, the transcritical and dual-stage cycles increased performance by 10 % and 5 % compared to the benchmark, respectively. This larger performance increase is primarily owed to an improved heat transfer in the heat recovery heat exchanger. In fact, using a liquid heat source could improve the net power production by up to 75 % for the same total heat exchanger area compared to when using a gas heat source.

Dual-stage and transcritical Rankine cycles seem to offer only a slight improvement over the benchmark for air-equivalent gas heat sources, but only for large total heat exchanger areas. Performance improvement was found to be higher for liquid heat sources. The results indicate that the performance increase for dual-stage and transcritical Rankine cycles compared to the subcritical benchmark depends on both the total heat exchanger area and heat source composition.

1. INTRODUCTION

The United Nations has published 17 global goals to promote a sustainable future (United Nations, 2015). Three of these are related to clean energy, innovation in industry and responsible production. One way to simultaneously work towards achieving these goals is to reuse industrial waste heat for power production. There are numerous completed studies on this topic. Varga *et al.* (2012) studied recovering waste heat in similar temperature ranges as in this work for butane, isobutane and isopentane, offering insights into different working fluids. Braimakis and Karellas (2018), Lecompte *et al.* (2015), Mahmoudi *et al.* (2018) and Walraven *et al.* (2013) reviewed existing research on use of Rankine cycles to produce power from low to medium temperature heat sources. The reviews explore several cycle designs, and in this work, the focus has been on three specific technologies. The three technologies are the dual-stage cycle design where there are two evaporating pressures, the conventional single-stage subcritical Rankine cycle, as well as the transcritical Rankine cycle. Li *et al.* (2019) investigated a dual-stage organic Rankine cycle where the high-pressure stage was supercritical and the low-pressure stage was subcritical. The study only investigated R1234ze(E) as the working fluid and water as the heat source. Wang *et al.* (2018) compared the thermodynamic and economic performance of dual-stage and single-stage Rankine cycles. The study only considered isobutane as the working fluid, but evaluated a range of heat source inlet temperatures and evaporating pressures. The heat source was modelled as water. Guzović *et al.* (2014), Manente *et al.* (2017) and Franco and Villani (2009) compared optimized results for single-stage organic Rankine cycles with their dual-stage counterparts for use in geothermal power plants. Manente *et al.* (2017) and Franco and Villani (2009) both concluded that the dual-stage cycles outperformed the single-stage cycles to a higher degree with increasing heat source temperatures. Manente *et al.* (2017) investigated several working fluids but, like many others in literature, considers fixed pinch points in the heat exchangers without consideration of the heat exchanger sizes. In this work, equal heat total heat exchanger area is used as a basis for comparing different cycle designs, which leads to different Pinch point temperatures for each fluid and cycle and a fairer basis of comparison. Furthermore, first- and second-stage evaporating pressures are not pre-determined, but instead the optimizer finds the optimum pressures for each cycle. Moreover, two heat sources with the same heat duty but different compositions are considered – air and water – and the power output from these sources are compared. To the best of the authors' knowledge, no existing research on performance comparison of different Rankine cycles involves this approach.

2. METHOD

2.1 Cases

The heat source cases evaluated are described in Table 1 and heat sink specifications for all cases are described in Table 2. Case 1 and 2 both evaluate a heat source at 150°C, but Case 1 represents an air heat source and Case 2 represents a liquid heat source. Case 3 resembles Case 1, but the heat source temperature is increased to 200°C. The properties of the heat sources are calculated using local temperatures and pressures throughout the heat exchangers. In all three cases, heat source mass flows are adjusted to give 1.4 MW heat duty when the source is cooled to ambient.

Table 1: Heat source case specifications

Heat source	Unit	Case 1	Case 2	Case 3
Heat duty	MW	1.4	1.4	1.4
Fluid	-	Air	Water	Air
Inlet temperature	°C	150	150	200
Mass flow	kg/s	10	2.4	7.3
Pressure	bar	1	5	1

Table 2: Heat sink specifications

Heat sink	Unit	All cases
Fluid	-	Water
Inlet temperature	°C	10
Mass flow	kg/s	25

2.2 Cycle modelling and optimization

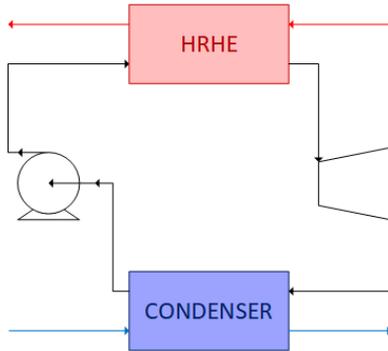


Figure 1: Process diagram for single stage cycle

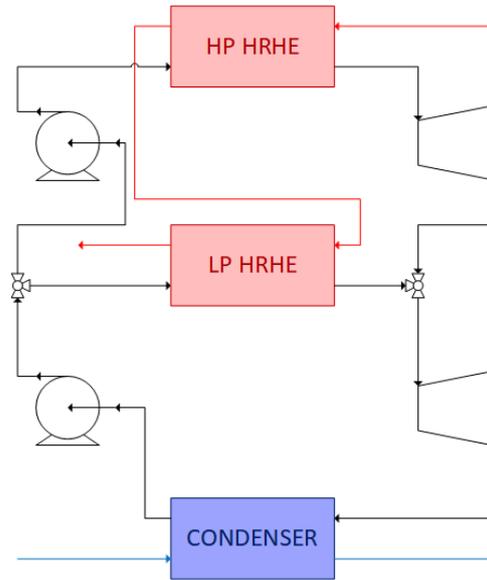


Figure 2: Process diagram for dual stage cycle

The process diagrams of the single stage and dual stage cycles are shown in Figure 1 and Figure 2 respectively. The Rankine cycles are analysed using a thermodynamic model that can model both single- and dual-stage cycle configurations, as well as subcritical or transcritical operating conditions. REFPROP 9 is used to calculate thermodynamic fluid properties. For simplicity, heat exchangers are modelled using UA-models with fixed heat transfer coefficients (HTCs) with an order of magnitude approximation similar to that described by Ho (2012); HTCs of 1,000 W/m²K for liquid heat sources and sinks, and a HTC of 100 W/m²K for the air-equivalent heat source flow evaluated in Cases 1 and 3. In reality, HTCs vary with local working fluid properties, flow and thermal conditions. Thus, this simplification is only expected to represent the difference in HTC order of magnitude between low-pressure gas flows and other flows. Based on this, HX outer areas are calculated by discretizing the HXs into n sub-section according to Equation 1 and Equation 2.

$$A_o = \sum_n \frac{Q_n}{U_o \Delta T_n} \quad (1)$$

$$U_o = \left(\frac{A_o/A_i}{h_i} + \frac{1}{h_o} \right)^{-1} \quad (2)$$

Outer to inner area ratios, A_o/A_i , in the heat recovery heat exchangers (HRHEs) recovering heat from the air heat sources are set to 15, based on the heat recovery heat exchanger designs described by Holfeld (2016). Area ratios are set to 1 for the liquid HRHEs since the HTC on both sides of the heat exchanger is the same.

Optimizing the cycles involves maximizing net power output, Equation 3, by optimizing the free variables given in Table 3. Note that the expander inlet enthalpy is a variable, meaning that it is not decided ahead of optimization whether the expander inlets will be superheated or saturated. Instead, the

optimizer finds what yields the highest net power. The maximum value of total heat exchanger area (THEA) was chosen such that the net power started to converge towards an asymptotic value. The minimum condenser pressure was chosen to avoid leakage issues. Furthermore, pressure drops in the heat exchangers and in the pipes between components are neglected in this model. Turbomachinery efficiencies are fixed and given in Table 4. During optimization, the cycles are constrained by a pre-determined total heat exchanger area according to Equation 1, allowing individual heat exchanger areas to distribute optimally. The NLPQL routine, developed for solving constrained non-linear programming problems, is used to optimize the cycles (Schittkowski, 1986).

$$W_{net} = W_{expander,1st\ stage} + W_{expander,2nd\ stage} - W_{pump,1st\ stage} - W_{pump,2nd\ stage} \quad (3)$$

In Equation 3, the "1st stage" refers to the HRHE with the lowest pressure in the dual-stage cycle, and the "2nd stage" refers to the HRHE with the highest pressure. The 2nd stage HRHE is not active in the single-stage cycles.

By allowing the fraction of working fluid mass flow entering the 2nd stage HRHE, $\dot{m}_{2nd\ stage}/\dot{m}_{1st\ stage}$, to vary between 0 and 1, the optimizer finds the split ratio yielding the highest net power output. Single-stage cycles are optimized by fixing the split ratio to 0, and can be either transcritical or subcritical depending on the optimum pressure level in the HRHEs. Only subcritical dual-stage cycles are considered.

Table 3: Free optimization variables and optimization constraints

Optimization variables	Working fluid	1 st stage HRHE pressure
		2 nd stage HRHE pressure
		HRHE split ratio
		Condenser pressure
		Expander inlet enthalpy
		Mass flow
	Heat source	Outlet temperature
Constraints		$\sum_k A_k \leq A_{tot}$
	Working fluid	$P_{cond} \geq P_{atm}$

Table 4: Turbomachinery efficiencies

Expander isentropic	0.85
Pump isentropic	0.70
Electro-mechanical	0.95

2.3 Working fluids

Six natural working fluids are investigated for each heat source temperature, Table 5. A paper by Shahrooz *et al.* (2018) that evaluates similar heat sources was used to identify promising working fluids, and a criterion by Agromayor and Nord (2017) was used to select optimal fluids based critical temperature.

Table 5: Working fluids investigated in Case 1-3

Case 1, Case 2	Propane, propene, cyclopropane, butane, isobutene, c2butene
Case 3	Propyne, butane, isobutene, c2butene, isopentane, neopentane

3. RESULTS

Figure 3 shows the net power produced by the different cycle designs in Cases 1-3 for increasing values of THEA by using the best performing working fluid in each point. The optimal working fluids are shown in Table 6. In Case 1 and 3, the net power output does not increase significantly with transcritical and dual-stage cycles compared to subcritical single-stage cycles, and the improvement is at most 3 % for a THEA of 2,000 m². For low THEAs, power production is in fact higher for the subcritical single-stage cycle. In Case 3, on the other hand, the alternative technologies improve performance to a larger extent; up to 5 % with the dual-stage cycle, and up to 10 % for the transcritical cycle. Moreover, net power output in Case 2 is significantly larger than in Case 1, even though the cases involve the same heat source duty and temperature. The net power is at most 75 % higher in Case 2 compared to Case 1 (THEA of 500 m²). The trend lines for Case 2 level off with high THEA. This is because the minimum temperature difference approaches 0 K with higher THEA, and the closer one is to this limit, the smaller the performance increase will be for a constant increase in THEA. Case 2 is thus close to this limit at THEA = 1500 m².

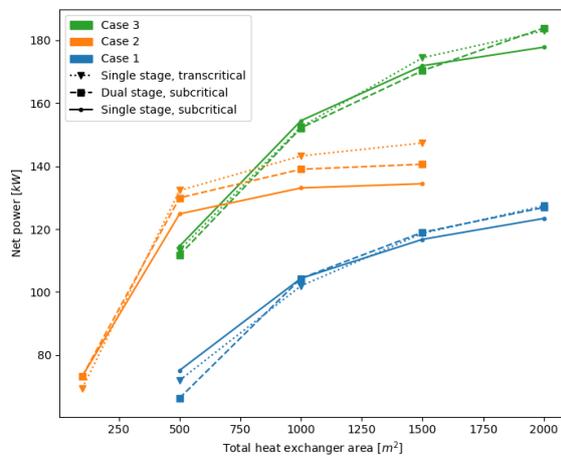


Figure 3: Net power vs. total heat exchanger area for the different cycle designs evaluated in Cases 1-3

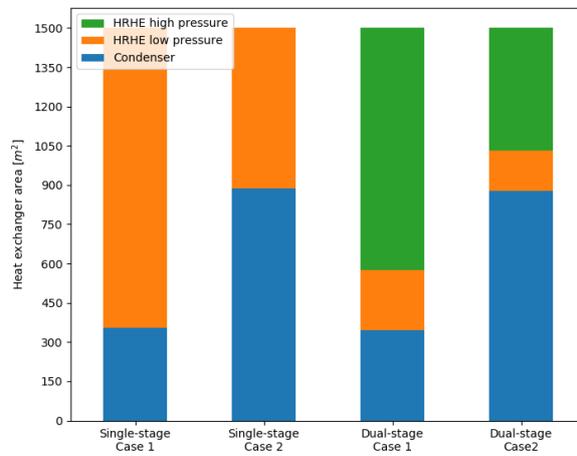


Figure 4: Breakdown of total heat exchanger area for single- and dual-stage butane cycles in Cases 1 and 2. Total heat exchanger area is 1,500 m²

Figure 4 shows a breakdown of total heat exchanger area for single- and dual-stage butane cycles in Case 1 and 2. The total heat exchanger area is 1,500 m² for all the cycles (inspect Figure 3 to see the power produced by these cases.) The figure shows that Case 2 has a much smaller fraction of HRHE area in both single- and dual-stage cycles, owing to a larger overall HTC in the HRHE.

Table 6: Optimal working fluid for each scenario

Case	Cycle	Area [m ²]				
		100	500	1000	1500	2000
1	Best simple	-	Butane	Isobutane	Butane	Butane
	Best dual	-	C2-Butene	Isobutane	Butane	Butane
	Best trans	-	Propane	Cyclopropane	Cyclopropane	Propane
2	Best simple	Butane	Butane	Butane	Butane	-
	Best dual	Butane	Butane	Butane	Butane	-
	Best trans	Cyclopropane	Cyclopropane	Cyclopropane	Cyclopropane	-
3	Best simple	-	C2-Butene	Isobutene	Butane	C2-butene
	Best dual	-	Isopentane	C2-Butene	C2-Butene	Butane
	Best trans	-	Butane	Propyne	Isobutene	Butane

4. DISCUSSION

Results show that transcritical cycles and dual-stage cycles yield at most 3 % higher net power than conventional subcritical cycles for the gaseous heat sources investigated in Case 1 and Case 3. For the liquid heat source investigated in Case 2, the equivalent increase in net power is 10 % for transcritical cycles and 5 % for dual-stage cycles. It could be difficult to motivate implementation of more complex cycles with an improvement in performance of only 3 %, since higher complexity would be expected to infer higher system costs.

There are two mechanism that may explain why the performance increase is so much higher for the alternative technologies for a liquid heat source (Case 2), as compared to when the heat source is a gas (Cases 1 and 3.) The first mechanism is that liquid heat sources transfer much more efficiently. In comparison with a gaseous heat source for the same HRHE area, the liquid heat source case would transfer more heat with smaller temperature difference to the working fluid. The first mechanism is thus the increased heat transfer, which transfers more energy to the working fluid, and the second mechanism is the reduced temperature difference, as this leads to less losses in the HRHEs. The heat transfer in the HRHE(s) is shown in Table 6 for the best simple, dual-stage and transcritical cycles for Case 1 and Case 2, and the corresponding mean temperature difference for these results are shown in Table 8.

Table 7: Comparison of heat transfer in HRHE(s) between Cases 1 and 2 for $A=1,500\text{m}^2$

	Case 1				Case 2			
	HP HRHE Q (kW)	LP HRHE Q (kW)	Total Q (kW)	Improve-ment (%)	HP HRHE Q (kW)	LP HRHE Q (kW)	Total Q (kW)	Improve-ment (%)
Best simple cycle	876.8	-	876.8	0.0	1010.1	-	1010.1	0.0
Best dual-stage cycle	779.4	149.0	928.4	5.9	913.0	188.1	1101.1	9.0
Best transcrit. cycle	906.0	-	906.0	3.3	1048.7	-	1048.7	3.8

Table 8: Comparison of mean temperature differences in HRHE(s) between Cases 1 and 2 for $A=1,500\text{m}^2$

	Case 1				Case 2			
	HP HRHE MTD (K)	LP HRHE MTD (K)	Average HRHE MTD (K)	Improve-ment (%)	HP HRHE MTD (K)	LP HRHE MTD (K)	Average HRHE MTD (K)	Improve-ment (%)
Best simple cycle	19.1	-	19.1	0.0	3.3	-	3.3	0.0
Best dual-stage cycle	21.1	16.1	18.6	2.8	3.9	2.5	3.2	3.7
Best transcrit. cycle	19.5	-	19.5	-2.0	3.0	-	3.0	8.2

It is clear how the alternative cycles do give larger improvements in both the amount of heat transferred and mean temperature differences for Case 2 (liquid heat source) compared to Case 1 (gaseous heat source). This may explain why in Figure 3, the alternative cycles yield a much higher performance increase in Case 2 as compared to Case 1. This suggests that in order for more complicated cycle designs to produce significantly more net power compared to the simple cycle design, the overall heat transfer coefficient in the HRHE must be high.

A secondary effect from the increased heat exchanger performance is that since the HRHE is more efficient per unit of surface area with a liquid heat source, the optimizer will distribute more surface area from the HRHEs to the condenser. This is seen in Figure 4. Overall, this reduces the total losses in the cycle more than just considering improved performance in the HRHEs alone.

Heat source characteristics appear to dominate system performance over cycle design. As seen in Figure 3, the liquid heat source in Case 2 yields up to 75 % higher net power output for the same THEA as the gaseous heat source in Case 1. In contrast, changing cycle design yields at most 10 % higher net power

output for the same THEA. The performance increase from gaseous to liquid heat source is owed to an order of magnitude higher heat transfer coefficient for the liquid heat source, resulting in improved heat transfer and higher net power. Moreover, increasing heat source temperature clearly improves potential, as comparison between Case 2 and Case 3 in Figure 3 shows. However, contrary to observations from cited literature, for these working fluids, the heat source temperature does not appear to significantly influence relative performance between different cycles; the maximum improvement was 3 % for both Case 2 and Case 3 using transcritical and dual-stage cycles. It should be noted that this result may not have a general validity, due to the limited number of working fluids investigated in this work.

In Case 1 and Case 3, an area ratio of 15 has been used in the HRHE to compensate for the small heat transfer coefficient on the gas side. In practice, this area ratio would have to be achieved by utilizing fins in the gaseous flow. In many industrial cases, the gas flows used for waste heat recovery contain pollutants that may lead to fouling in the heat exchanger. It has previously been made clear that high heat transfer coefficients seem to be a prerequisite for alternative cycle designs to yield a significant performance improvement. Since it is challenging to achieve a high heat transfer coefficient for dirty gases, it seems that it is not worthwhile to consider combining these more advanced cycle designs with dirty gas heat sources.

A significant simplification made in this work is the assumption of fixed heat transfer coefficients for liquid, two-phase and gaseous flows, as well as the neglect of pressure losses. Although the former partly accounts for the effect of flow characteristics on heat transfer, it does not account for the effects of fluid properties, operating conditions and heat exchanger geometry on both heat transfer and pressure loss. Such effects would probably affect the relative performance of different cycle designs, such as transcritical cycles, which typically have higher operating pressures than subcritical cycles.

5. CONCLUSION

Transcritical and dual-stage Rankine cycles offer some performance improvements compared to the subcritical cycle, but only for large total heat exchanger areas. For the air heat sources evaluated in Case 1 and 3, the transcritical and dual-stage cycles yield no more than 3 % higher net power than the subcritical cycle. For the liquid heat source evaluated in Case 2 on the other hand, the transcritical and dual-stage cycles yield up to 10 % and 5 % higher net power than the subcritical cycle, respectively. For small total heat exchanger areas in Case 1 and 3, the subcritical cycle actually results in the highest net power, but the difference is not large. Thus, this work indicates that the performance of dual-stage Rankine cycles depends on the total heat exchanger area and heat source composition, contrary to what was found by Manente *et al.* (2017) and Franco and Villani (2009).

NOMENCLATURE

A	Heat exchanger area	(m ²)
dual	Dual pressure cycle	(-)
h	Heat transfer coefficient	(W/(m ² K))
HRHE	Heat recovery heat exchanger	(-)
MTD	Mean temperature difference	(K)
P	Pressure	(kPa)
simple	Simple cycle	(-)
T	Temperature	(K)
THEA	Total heat exchanger area	(m ²)
trans	Transcritical cycle	(-)
Q	Heat duty	(kW)
U	Total heat exchange coefficient	(W/(m ² K))
W	Work	(kW)
Subscripts		
cond	Condenser	

i	Inner area of heat exchanger
k	Heat exchanger k
n	Heat exchanger element n
o	Outer area of heat exchanger
tot	Total

REFERENCES

- AGROMAYOR, R. & NORD, L. O. 2017. Fluid selection and thermodynamic optimization of organic Rankine cycles for waste heat recovery applications. *Energy Procedia*, vol. 129, p. 527-534.
- BRAIMAKIS, K. & KARELLAS, S. 2018. Energetic optimization of regenerative Organic Rankine Cycle (ORC) configurations. *Energy Conversion and Management*, vol. 159, p. 353-370.
- FRANCO, A. & VILLANI, M. 2009. Optimal design of binary cycle power plants for water-dominated, medium-temperature geothermal fields. *Geothermics*, vol. 38, p. 379-391.
- GUZOVIĆ, Z., RAŠKOVIĆ, P. & BLATARIĆ, Z. 2014. The comparison of a basic and a dual-pressure ORC (Organic Rankine Cycle): Geothermal Power Plant Velika Ciglena case study. *Energy*, vol. 76, p. 175-186.
- HO, T. 2012. *Advanced Organic Vapor Cycles for Improving Thermal Conversion Efficiency in Renewable Energy Systems*. PhD, University of California, Berkeley, 130 p.
- HOLFELD, A. 2016. *Experimental investigation of heat transfer and pressure drop in compact waste heat recovery units*. PhD, Norwegian university of science and technology, 206 p.
- LECOMPTE, S., HUISSEUNE, H., VAN DEN BROEK, M., VANSLAMBROUCK, B. & DE PAEPE, M. 2015. Review of organic Rankine cycle (ORC) architectures for waste heat recovery. *Renewable and Sustainable Energy Reviews*, vol. 47, p. 448-461.
- LI, J., GE, Z., DUAN, Y. & YANG, Z. 2019. Design and performance analyses for a novel organic Rankine cycle with supercritical-subcritical heat absorption process coupling. *Applied Energy*, vol. 235, p. 1400-1414.
- MAHMOUDI, A., FAZLI, M. & MORAD, M. R. 2018. A recent review of waste heat recovery by Organic Rankine Cycle. *Applied Thermal Engineering*, vol. 143, p. 660-675.
- MANENTE, G., LAZZARETTO, A. & BONAMICO, E. 2017. Design guidelines for the choice between single and dual pressure layouts in organic Rankine cycle (ORC) systems. *Energy*, vol. 123, p. 413-431.
- SCHITTKOWSKI, K. 1986. NLPQL: A fortran subroutine solving constrained nonlinear programming problems. *Annals of Operations Research*, vol. 5, p. 485-500.
- SHAHROOZ, M., LUNDQVIST, P. & NEKSÅ, P. 2018. Natural refrigerants for low temperature power cycles. *13th IIR Gustav Lorentzen Conference*. Valencia.
- UNITED NATIONS. 2015. *About the Sustainable Development Goals* [Online]. Available: <https://www.un.org/sustainabledevelopment/sustainable-development-goals/> [Accessed 11.02. 2019].
- VARGA, Z., RABI, I. & FARKAS, C. 2012. Waste heat recovery with organic Rankine cycle in the petroleum industry. *Chemical Engineering Transactions*, vol. 29, p. 301-306.
- WALRAVEN, D., LAENEN, B. & D'HAESELEER, W. 2013. Comparison of thermodynamic cycles for power production from low-temperature geothermal heat sources. *Energy Conversion and Management*, vol. 66, p. 220-233.
- WANG, M., CHEN, Y., LIU, Q. & YUANYUAN, Z. 2018. Thermodynamic and thermo-economic analysis of dual-pressure and single pressure evaporation organic Rankine cycles. *Energy Conversion and Management*, vol. 177, p. 718-736.

ACKNOWLEDGEMENTS

FME HighEFF, a research centre for an energy efficient and competitive industry for the future, is acknowledged for funding this research.