THERMODYNAMIC ANALYSIS OF A TWO-STAGE ORGANIC RANKINE CYCLE WITH EJECTOR

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ABSTRACT

Results of a detailed thermodynamic analysis of a novel two-stage organic Rankine cycle with ejector are presented and compared with other ORC cycles in this contribution. The different cycle concepts were analyzed by means of the commercial power plant process simulation tool EBSILON *Professional*. This tool permits the simulation of cycle performance under the assumption of realistic component performance behaviour and fluid properties based on REFPROP data. The ejector performance was modelled on the basis of a one-dimensional gasdynamics approach, and it was implemented as sub-routine within the cycle simulation tool. It was found that the novel concept is superior to other cycles with ejectors but it is still inferior to the conventional cycle regarding thermal efficiency.

1. INTRODUCTION

It is well known that lowering the expander backpressure is beneficial for the performance of steam and organic Rankine cycles (ORC). Typically, the temperature level of the coolant medium governs the achievable backpressure level, but diffusers are also applied for lowering turbine exhaust pressure levels and to increase the turbine power output in several applications.

In the case of small and medium organic Rankine cycles it is in principle possible to achieve additional significant backpressure level reductions by means of ejectors directly placed in the turbine exhaust section. This approach was recently proposed by Li et al. (2012, 2013). In their so-called EORC concept, two independent evaporators were applied, and the vapour from the secondary evaporator worked as primary flow for the ejector to suck the exhaust from the expander so as to decrease the backpressure. That cycle concept is schematically shown in Figure 1.c, and it is compared with the simple ORC cycle (Figure 1.a) and a double organic Rankine cycle (DORC, Figure 1.b). Regarding heat source utilisation, the EORC might offer some advantages, but a thermodynamic analysis indicated a substantial exergy penalty for driving the ejector by means of an independently generated vapour. This issue has a substantial weight in the cases where the thermal efficiency of the cycle is in focus. In addition to the exergy analysis, a second heat transfer device would substantially increase the plant cost and complexity for both EORC and DORC. With a look to a second-law thermodynamic analysis, it is more obvious to drive the ejector by second-stage vapour extracted after a first expansion in a high pressure turbine stage. Then, a two-stage organic Rankine cycle with ejector (TSORCE) results without the need to implement a secondary evaporator. Furthermore, the vapour extraction lowers the low-pressure volume flow rate, and a rather compact turbine design including the ejector could be achieved.

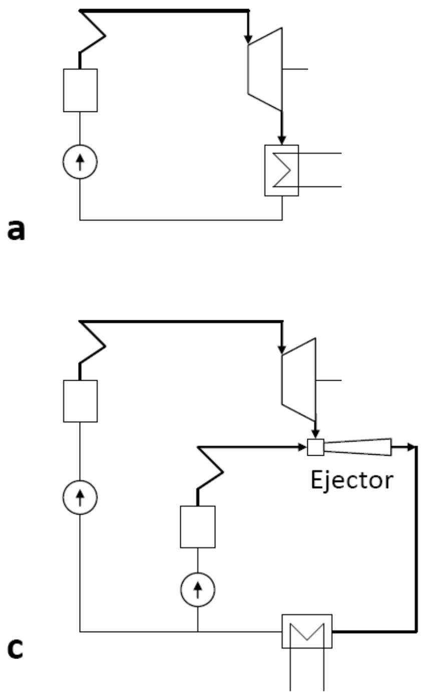

Figure 1: Comparison of ORC cycles. **a** simple ORC **b** double ORC (DORC) with two heat supply devices **c** organic Rankine cycle with ejector (EORC) as proposed by Li et al. (2012) **d** novel two-stage ORC with ejector (TSORCE). For sake of simplicity, the use of an additional recuperator between condenser and evaporator in order to increase the cycle efficiency is not shown.

This novel concept is schematically shown in Figure 1.d. For sake of simplicity, recuperators or other additional devices like control valves are not shown for all cycle concepts in Figure 1. The new TSORCE concept (Figure 1.d) does not increase the *cycle* complexity, because only a single evaporator, a single pump and a single condenser are needed as in the case of the simple ORC cycle (Figure 1.a). Only the ORC turbine device would be more complex, because two stages are required and the turbine exhaust section has to be designed as part of an ejector device. However, with a look to turbine efficiency and shock losses, it might be useful, in any case, to design the expander as a multi-stage turbine.

So far, the consideration of ejectors has been more common in refrigeration systems, as discussed by Ko and Kim (2013), Shi et al. (2015), or Zhang et al. (2016). There, the expensive compressor systems could be replaced by jet cooling due to an ejector device, see also Tashtoush et al. (2015). The background of the present study was the development of an ORC process utilising the coolant mass flow of large piston engines with a temperature level up to 140°C. Here, the main goal is to maximise the thermal efficiency of the cycle working with a low-temperature source of high capacity. The following study is hence restricted to comparable low temperature sources for a power cycle. With a look to the new concept shown in Figure 1.d, cycle optimization is not an easy task, because the involved main process parameters are strongly interacting. For instance, lowering the backpressure of the low-pressure turbine stage increases the work output of the LP turbine stage, but lowering the backpressure requires a higher mass flow rate for the primary ejector stream (and/or a higher pressure level for the extraction after the high-pressure turbine stage). In addition, it is not clear how actual turbine efficiencies would affect the cycle with ejector. In literature (Li et al. (2012)), the EORC (Figure 1.c) was theoretically analyzed with the rather questionable assumption of ideal turbine efficiencies, and a substantial deviation with experimental data was reported. In this contribution the two-stage organic Rankine cycle with ejector is analyzed by means of a professional power plant process simulation tool which permits the simulation of cycle performance with the assumption of realistic component performance behaviour and fluid properties based on accurate thermodynamic data.

2. GENERAL THERMODYNAMIC CONSIDERATION

By means of a general thermodynamic analysis based on the balance equations, it is rather easy to proof that the TSORCE concept certainly does not provide a *direct* advantage regarding thermal cycle efficiency in comparison with the simple ORC. Instead, the TSORCE concept might offer *indirect* advantages like more efficient turbine designs due to better-balanced volume flow rates through the low-pressure (LP) turbine stage.

The nomenclature used for the discussion of the expansion and ejector process is shown in Figure 2. The high-pressure (HP) turbine expansion with the full mass flow rate begins at point HP and ends at point P in the *h,s*-diagram shown in Figure 2.b. Point P represents the inlet state of the primary (or motive) flow, point S represents the inlet state of the secondary flow. This state is also the exhaust or outlet state of the LP-turbine. At the exit of the ejector, state M is achieved. This state can be obtained by means of the general balance equations for mass and energy:

$$\dot{m}_m = \dot{m}_p + \dot{m}_s = \varphi \dot{m}_m + (1-\varphi)\dot{m}_m \qquad \text{with } \varphi = \dot{m}_p / \dot{m}_m \tag{1}$$

$$\dot{m}_m h_m = \dot{m}_p h_p + \dot{m}_s h_s \Leftrightarrow h_m = \varphi h_p + (1-\varphi) h_s \tag{2}$$

In the case of an isentropic ejector, the ejector exit state (point M in Figure 2.b) obeys the entropy equation

$$\dot{m}_m s_m = \dot{m}_p s_p + \dot{m}_s s_s \Leftrightarrow s_m = \varphi s_p + (1-\varphi) s_s \tag{3}$$

Combining equation (2) with equation (3) directly shows that in the case of an isentropic (ideal) ejector, the exit state M is on the line PS combining the LP-turbine inlet and outlet states P and S. However, as discussed, for instance, by Nesselmann (1950), actual ejectors are not isentropic devices, and the actual ejector exit state, M*, is a point towards higher specific entropy. Since the energy balance determines the specific enthalpy h_m, the actual ejector exit pressure p^*_m has to be lower than the ideal ejector exit pressure p_m as illustrated by means of Figure 2.b.

In order to evaluate the thermal efficiency of the TSORCE in comparison with the simple ORC, it is sufficient to compare the specific turbine work w for the low-pressure expansion from state P to the turbine end state. For the following comparison, the condenser pressure is assumed to be constant, and in order to enable a fair comparison the condenser pressure should be identical to the exit pressure of the involved expander devices.

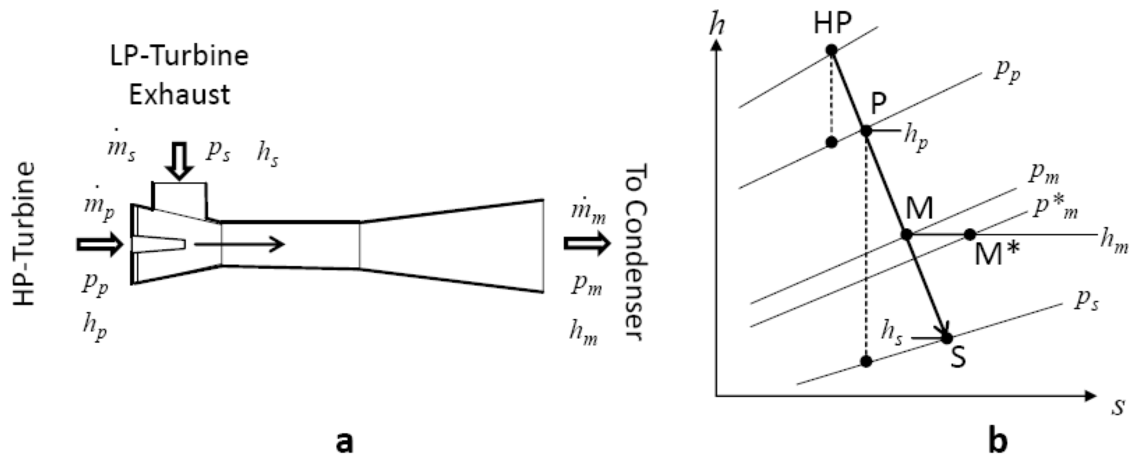

Figure 2: One-dimensional ejector model and nomenclature (**a**) and *h,s*-diagram of expansion and ejector process for the TSORCE concept (**b**). The isentropic expansion and ejector processes would follow the dotted lines but actual processes are non-isentropic as schematically shown by the thick line.

In the case of the simple ORC, this specific low-pressure expansion work is given by means of $w_{\mathrm{ORC}} = h_p - h_m$. In the case of the TSORCE concept, this work is given by $w_{\mathrm{TSORCE}} = (1 - \varphi)\,(h_p - h_s)$, because here only the mass flow rate fraction $(1 - \varphi)$ expands in the low pressure turbine (but to a lower end pressure p_s). A higher work, i. e. a higher thermal efficiency, could only be achieved by the TSORCE concept if the inequality $w_{\mathrm{ORC}} < w_{\mathrm{TSORCE}}$ would hold. But this is not possible, since the energy balance equation (2) has to be obeyed. In the ideal case, the ejector would be an isentropic device, and then both thermal efficiencies would be identical. In the case of actual ejectors, the thermal efficiency of the TSORCE is always less than the thermal efficiency of the simple ORC (if the same condenser pressure level is assumed). However, the TSORCE concept might offer some indirect advantages regarding expander design and part load behavior. Furthermore, the TSORCE concept is superior in comparison with the EORC and DORC concepts. It is therefore still useful to investigate the TSORCE performance in more detail.

3. MODELING AND SIMULATION

Thermodynamic modeling of ORC systems is preferably carried out using a modular approach, in which the overall system is obtained by the connection of suitable subsections representing the components of the system under consideration. For each subsystem or component, the governing equations have to be solved. In the following, only the main aspect of the present modeling and simulation method are briefly explained. An overview and a detailed discussion of the various mathematical and technical aspect of modeling and simulation of thermodynamic cycles, power plants and components can be found elsewhere (see, for instance, Epple et al. (2012)). An introduction to object-oriented modeling of ORC systems is given by Casella in Macchi and Astolfi (2017).
In the present study, the thermodynamic cycles were simulated by means of the professional power plant simulation tool EBSILON *Professional* developed by STEAG. EBSILON *Professional* numerically solves the thermodynamic balance equations arising from a system of thermodynamic components which can be connected by means of a graphical user interface. A comprehensive component and material data library is available, but user-defined models and routines can be implemented, too. In addition, fluid properties can be alternatively calculated using the REFPROP data base. More information about REFPROP data for ORC applications can be found by the corresponding chapter of Lemmon in Macchi and Astolfi (2017).
Although the standard component library of EBSILON *Professional* contains hundreds of different parts including also a steam ejector model for steam power plant applications, an organic vapor ejector was missing so far. Hence, the simulation tool architecture illustrated by means of Figure 3 was used for the following analysis study. For the thermodynamically cycle simulation, the tool EBSILON *Professional* was used. The required fluid properties were provided by REFPROP, and the relevant thermodynamical variables (e. g. specific enthalpy h or entropy s as function of pressure p and temperature T) were read into EBSILON. The ejector was modeled within EBSILON as a user-defined mixing component. This user-defined special mixing component obeyed the general mass flow and energy balance equations (1) and (2), but the backpressure p_s of the LP-turbine (i. e., the pressure of the secondary or suction mass flow due to the action of the primary mass flow) had to be prescribed as special constraint condition within EBSILON. The ejector variables were calculated in a separate ejector simulation tool which solved numerically the governing equations of the present ejector model. The literature on ejector modeling is extensive. Fundamentals were proposed by Schütz (1939) and Keenan and Neumann (1942); the present work employed the same gasdynamical model as described in Tashtoush et al. (2015). The required ejector model calculations were performed by means of standard mathematical software (in the present case: EES). The chosen ejector model assumes the fluid as a perfect gas. This approach is limited in the case of organic vapors, but it provided reasonable starting values for the pressure levels. These pressure values were used as input for the cycle simulation considering real gas behaviour. The overall thermodynamic cycle parameters (e. g. temperature level of the heat source or mass flow rate) and the component efficiencies (e. g. isentropic turbine efficiency) were prescribed at the beginning of each computational run. The performance (output power or thermal efficiency) were finally exported as results.

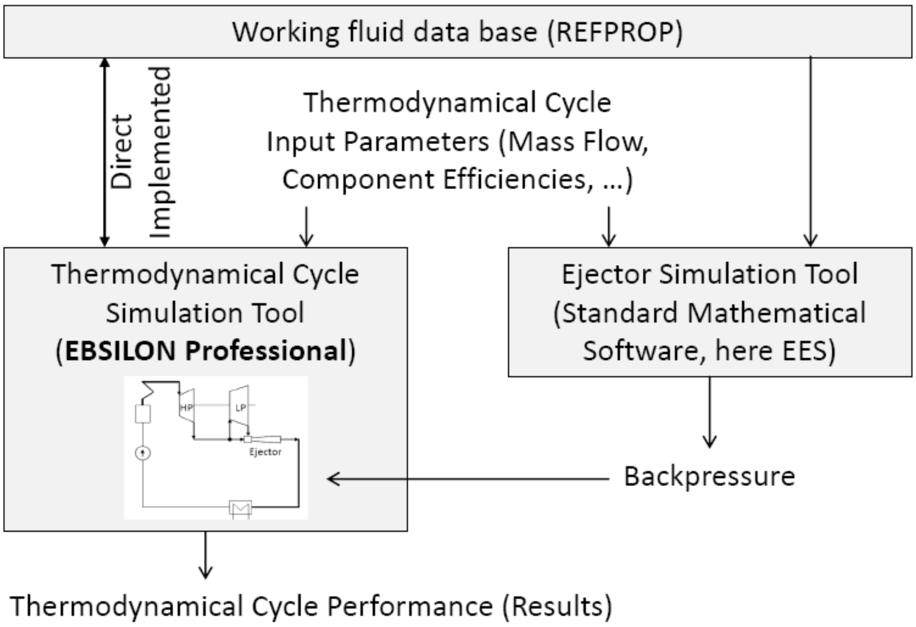

Figure 3: Simulation tool architecture for the analysis of ORC systems with ejector

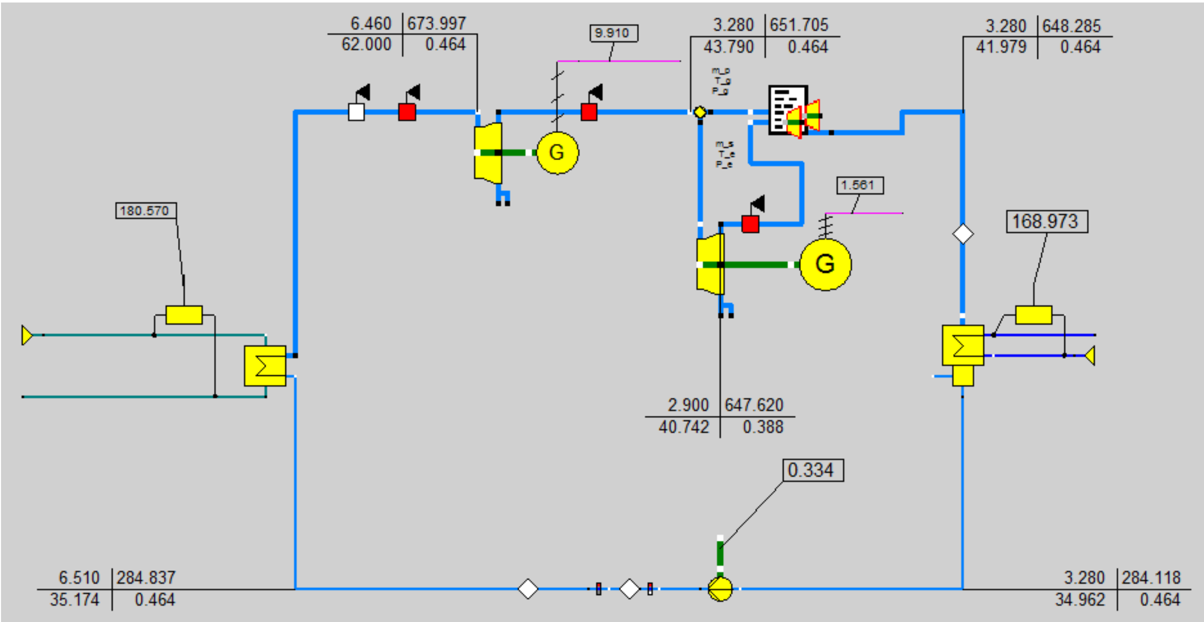

Figure 4: Simplified EBSILON *Professional* model of a two-stage ORC with ejector

In Figure 4, a simplified EBSILON *Professional* model of a two-stage ORD with ejector is shown. At characteristic sections, the process variables pressure p (in bar), specific enthalpy h (in kJ/kg), temperature T (in °C) and mass flow rate m (in kg/s) are shown as result of the cycle computation. Heat input and waste heat flow, electric power output and pump power inputs are also shown in Figure 3 (in kW). The ejector is represented by the user-defined component placed at the upper right of the diagram. It connects the HP- and LP-turbines, and it requires as external parameter the backpressure. The example shown in Figure 4 assumed a backpressure of p_s = 2.90 bar and the diffuser outlet pressure p_m was set equal to the value of the pressure of the primary jet, i. e. $p_m = p_p$ = 3.28 bar for this specific sample illustration.

4. RESULTS AND DISCUSSION

The following case study was roughly oriented at the examples discussed by Li et al. (2012). The working fluid was R600 (Butane C4H10, EBSILON *Professional*-REFPROP reference no. 1006). The corresponding cycle data and parameters of the present study are listed in Table 1. Thermal oil was assumed as heat source, and cold water was assumed as coolant for every cycle simulation. The vapour enters the HP-turbine with a small superheat level at pressure p_{HP} = 6.46 bar and t_{HP} = 62°C. The HP-turbine outlet pressure was assumed to be p_p = 3.28 bar.

In a first set of simulations, the isentropic efficiency of the involved turbines was assumed to be constant (namely 80 %). The back-pressure of the LP-turbine, p_s, was varied, and the ejector model was employed in order to provide realistic exit pressure levels, p^*_m. It should be kept in mind that the pressure level p^*_m took into account the irreversibility of the ejector process in accordance to reasonable ejector model assumptions (see Tashtoush et al. (2015)). The primary mass flow rate was fixed to 0.076 kg/s. As rated output, a total electric power of about 11.5 kW was defined as target value. This means, that due to the different ejector performance, different total mass flow rates were obtained in order to achieve this target output value. The corresponding results for the simulated cases are listed in Table 2. Further results of the cycle simulations are plotted graphically in Figure 5.

It can be observed from Table 2 and Figure 5 that with decreasing back-pressure p_s the cycle efficiency η increased since the heat supply decreased for the same total power. A smaller total mass flow rate (m_m) was also needed for lower back-pressures for the same total power, but the HP-turbine power output decreased with decreasing total mass flow rate. Since the primary mass flow rate (m_p) for the ejector was fixed, the primary mass flow rate ratio $\varphi = \dot{m}_p / \dot{m}_m$ increased with decreasing back-pressure p_s. This indicates that more fluid was relatively needed for achieving low pressure levels. However, the LP-turbine output still increased with decreasing LP-turbine mass flow rate, because the enthalpy drop $h_p - h_s$ increased stronger with decreasing back-pressure. As a result, the output levels of the HP and LP-turbines became more balanced with decreasing back-pressure. In the case of comparable high back-pressures, the HP-turbine contributed much more to the total power and the LP-turbine was of minor importance. It can be stated that the ejector is only useful in cases when a low back-pressure p_s can be achieved. Since low back-pressure levels p_s can only be achieved at comparable low ejector exit pressures p^*_m (i.e. condenser pressures), this requires the existence of a cold heat sink for cooling the condenser. And under such a condition, the conventional ORC performance would be also improved.

Table 1: Main assumed cycle input data for the present simulation study

Cycle Property	**Unit**	**Value**
Inlet Temperature of Thermal Oil (Heat Source)	°C	137
Inlet Temperature of Coolant (Heat Sink)	°C	10
HP-Turbine Inlet Vapour Temperature t_{HP}	°C	62
HP-Turbine Inlet Vapour Pressure p_{HP}	bar	6.46
HP-Turbine OutletVapour Pressure p_p	bar	3.28
Isentropic Efficiency of Pump	%	80

Table 2: Case study results (isentropic turbine efficiency 80 %, total electrical power output 11.5 kW)

Back-Pressure p_s [bar]	**Mass Flow** m [kg/s]	**Heat Supply** Q [kW$_{th}$]	**LP Turbine** P_{LP} [kW]	**Cycle Efficiency** η [%]
2.9	0.465	191	1.56	5.8
2.7	0.438	180	2.30	6.2
2.5	0.411	173	2.96	6.6
2.3	0.384	162	3.55	7.1
2.1	0.356	154	4.06	7.4
1.9	0.329	144	4.47	7.8

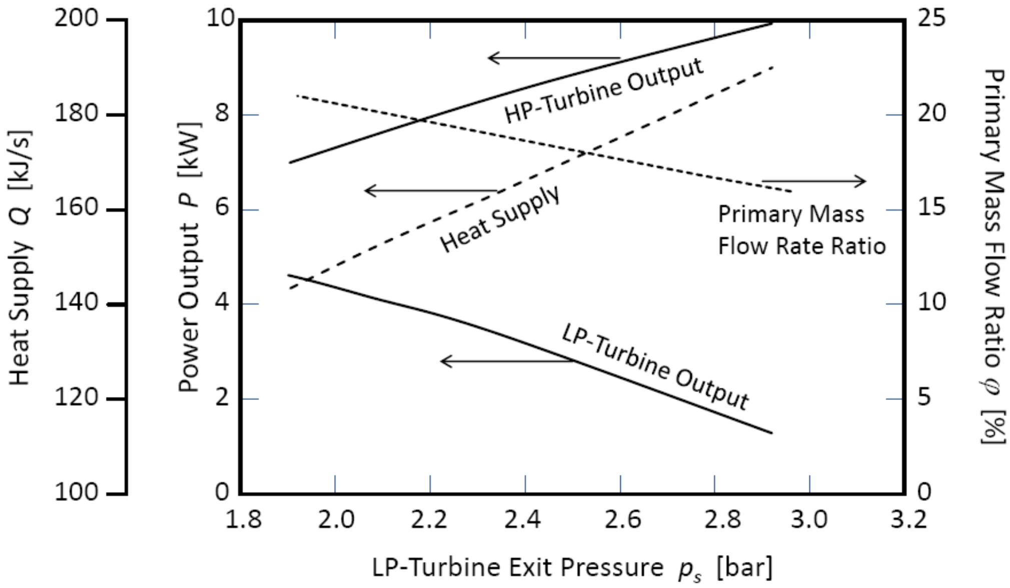

Figure 5: Case study results (isentropic turbine efficiency 80 %, total electrical power output 11.5 kW)

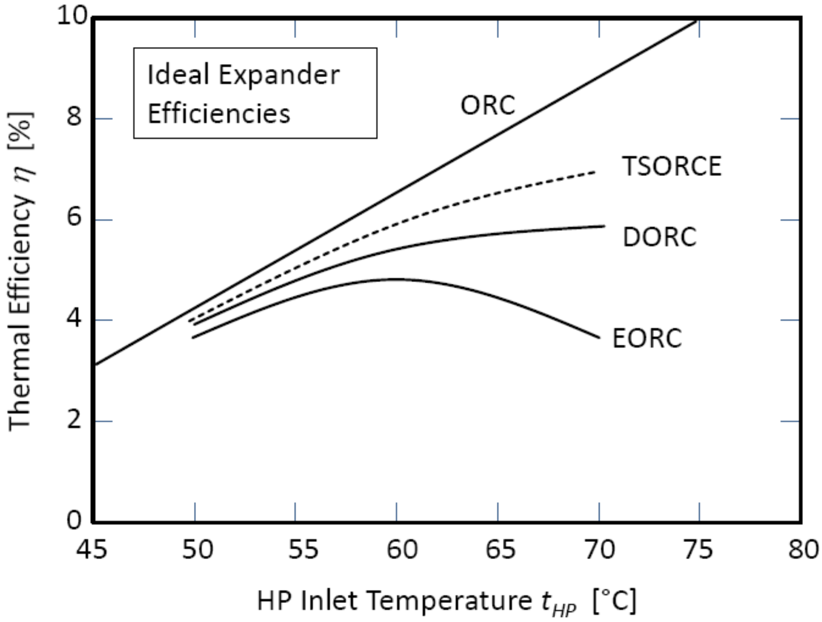

Figure 6: Comparison of thermal efficiencies for different cycle concepts considering ideal expander devices and a realistic ejector performance

Finally, the novel TSORCE concept was compared with the other ejector processes as proposed by Li et al. (2012). Assuming ideal turbine efficiencies (100 %) and identical HP inlet temperatures as done by Li et al. (2012), the four cycles of Figure 1 can be directly compared regarding their thermal efficiency. This is shown in Figure 6. As expected, the simple ORC has always the highest thermal efficiency if the same back-pressure is assumed. The EORC concept proposed by Li et al. (2012) has the lowest thermal efficiency. The double ORC concept (DORC) is between these limits. The present TSORCE concept would be superior against DORC and EORC, but the resulting cycle efficiency is still lower than the simple ORC. This is due to the inherent losses within the ejector process. Only in the case of an ideal ejector with 100 % component efficiency, the ideal TSORCE would achieve the same efficiency as the simple ORC (and the corresponding graphs would collapse in Figure 6).

5. CONCLUSION

The novel two-stage ORC with ejector (TSORCE) concept enables better thermal efficiencies than the EORC or the DORC proposed in recently in literature, but the TSORCE is not superior in comparison to the simple ORC if identical expander efficiencies would be assumed. Only secondary benefits regarding turbine design might be achieved by means of the TSORCE concept: A reduction of the LP-turbine mass flow rate might permits higher expander efficiencies, and then the TSORCE concept might provide a slight advantage against conventional ORC. However, since the TSORCE is working with a lower LP-turbine exhaust pressure levels than the ORC concept, the selection of the working fluid would be rather critical, because the lower pressure level would also lead to higher volume flow rates. Here, the disadvantage of higher specific volumes has to be balanced by the achievable enthalpy drop. Another slight advantage regarding the expander efficiencies might be achieved with a look to the polytropic and the isentropic turbine efficiency: Typically, it is more efficient to consider higher LP-turbine pressure ratios due to the inherent reheat effect during expansion.

NOMENCLATURE

h	specific enthalpy	(J/kg)
p	pressure	(Pa)

Subscript

p	primary (motive stream)
s	secondary (suction stream or back-pressure)

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