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Comparing Three Methods for Design Analyses of Rankine Cycle for Waste Heat Recovery from Natural Gas Compression

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ABSTRACT

This study applies three methods for design of Rankine cycles utilizing waste heat in the oil and gas industry. The methods differ in the level of detail of heat exchanger model applied and in the flexibility with respect to heat exchanger design. All the methods involve formulating a constrained optimization problem with the objective of maximizing net power output, but vary in the constraints related to limit the size of the heat exchangers. The cycles with using two working fluids, n-butane and a n-butane/n-pentane mixture (30%/70%), are compared for each method. The methods yield different results; both the maximized net power output and the best-performing working fluid differ with respect to the method used, emphasizing the importance of method selection.

Keywords: Oil and gas industry, waste heat recovery, Rankine cycle, natural working fluids, natural fluid mixtures.

1. INTRODUCTION

The global oil and gas waste heat recovery market is forecasted to have a growth at a compound annual growth rate of 7.5% during the period 2019–2023, based on the analysis by Market Research Future (2019). Among the various sources of waste heat, natural gas export compressors provide a significant potential for recovering waste heat for power generation. The organic Rankine cycle (ORC) is a proven technology for low and medium temperature heat sources, which gives the opportunity for waste heat recovery in the oil and gas industry.

For promoting the ORC technology in low-temperature heat recovery, the heat recovery system is desired to be efficient and cost-effective. The improvement of thermal efficiency can be realized by e.g. using mixtures as working fluid (Quoilin et al., 2013). However, Andreasen et al. (2019) reviewed several studies based on the methods of thermodynamic optimization and made general conclusions that the gained performance of ORCs with using zeotropic mixtures requires heat transfer equipment with larger capacities (UA values) and heat transfer areas. The question whether the increased performance with using zeotropic mixture compensates for the larger investment in heat exchangers (HXs) was raised. In fact, some studies have found that the methods of fixing total heat transfer area (Baik et al., 2012) or economic optimization (Andreasen et al., 2016), show that using mixture did not have a significant advantage over pure fluids. As seen, these studies ended in different conclusions as cases (e.g. waste heat source), working fluids considered, and evaluation methods employed are varied. There exist extensive studies for different cases and assessment of fluids; however, the number of works comparing the different assessment methods on the same basis is limited. Andreasen et al. (2019) assessed four methods for performance comparison of pure fluids and zeoptropic mixtures in ORC with a heat source temperature at 120 °C. The methods involve evaluation of cycles based on the same minimum pinch point temperature difference, mean temperature difference, thermal capacity (UA value), and surface area considering a shell-and-tube heat exchanger. They concluded that the method of minimum pinch point temperature difference results in optimistic estimations of the benefits of using zeotropic mixtures, while the method of equal mean temperature difference or UA value result in convervative estimations. They claimed that the method of heat transfer area is too comprehensive, and they suggested to use the method of minimum pinch point temperature difference and UA value (or mean temperature difference) concurrently for preliminary fluid selection.

The present study considers a heat source at 250 °C and focuses on the assessment of the performance with given size of the heat exchangers, which is directly related to the investement cost. Three methods are employed for estimating the design performance of an ORC utilizing waste heat from natural gas compression. The first method is a pure thermodynamic analysis that estimates the thermodynamic performance potential of an ORC within a limited budget of UA value. The second method involves the use of a generic heat exchanger model capable of estimating pressure loss and heat transfer coefficients, without relying on a certain heat exchanger type. This method gives the heat transfer area as an indication of heat exchanger size, and has been presented in detail in Hagen et al. (2020). The last method considers a plate heat exchanger model requiring detailed description of plate geometry, providing the size of the heat exchanger in terms of mass and volume. Moreover, the heat exchanger geometry parameters are simultaneously optimized together with other process variables, which supplies opportunities for optimal design of the heat exchangers. Two working fluids, n-butane and a n-butane/n-pentane mixture (30%/70%), are employed as working fluids for demonstrating the three methods.

2. METHODS AND RESULTS

2.1. Case definition

The waste heat source considered is an offshore natural gas export compression train, where the process stream is compressed to 180 bar in several stages with intercooling and separation of condensate between each stage. The explored case is a novel variant with compression occuring in fewer stages, resulting in the gas reaching a temperature of 250 °C before cooling and export. The heat released by cooling the natural gas can be extracted and converted to power using an ORC. The heat source outlet temperature after the cycle is desired to be smaller than 80 °C, but there is no lower limit. For simplicity, pure methane is used to represent the gas mixture.

2.2. Cycle and components

The heat-to-power conversion is realized using a Rankine cycle, as illustrated in Figure 1. The heat is extracted from the heat source by the pressurized working fluid in the heat recovery heat exchanger (HRHE). The high pressure and temperature working fluid expands in the expander and power is generated and transferred to a generator. Then, the working fluid is cooled down and condensed in the condenser against the heat sink. After that, it is pumped back to the HRHE. In addition to these main components, a "source pump" and "sink pump" have been included to account for the pressure loss of the heat source in the HRHE, the pressure loss of the heat sink in the condenser and the pressure loss of the heat sink pipe (assumed as 0.5 bar).

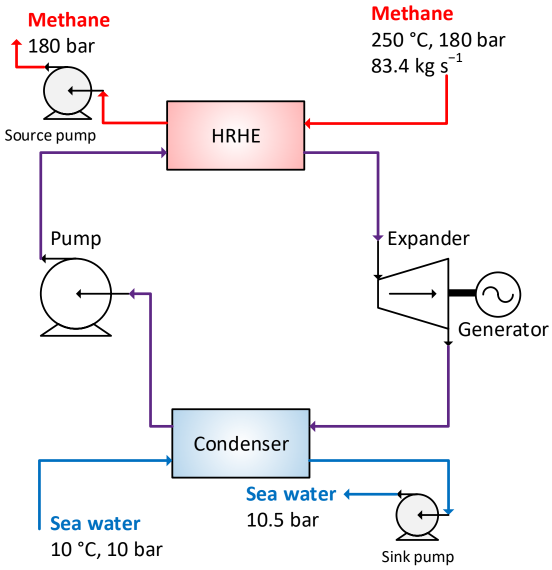


Figure 1: Layout of the heat-to-power cycle.

The expander and the pumps are modelled as an isentropic process corrected by isentropic efficiency, and power consumed or generated is corrected with mechanical efficiency. The heat source and heat sink conditions and the efficiencies of the pumps and the expander are summarized in Table 1.

Table 1: Heat source and heat sink conditions and efficiencies of pumps and expander

Heat source	Fluid	Methane
	Inlet temperature	250 °C
	Inlet pressure	180 bar
	Mass flow rate	83.4 kg s^{-1}
Heat sink	Fluid	Sea water
	Temperature	10 °C
Pumps	Isentropic efficiency	0.70
	Motor efficiency	0.95
Expander	Isentropic efficiency	0.85
	Generator efficiency	0.95

The heat exchangers are modelled with three different models assuming counter-current flows. These models differ with respect to the level of detail and represent the size of the heat exchangers in terms of UA value, area and mass, as described in Section 2.3.

2.3. Three methods of heat exchanger modelling

2.3.1. UA analysis

In the UA analysis, a pure thermodynamic heat exchanger model is used, namely the conditions of one stream (cold or hot) are specified and the capacity can be calculated directly without iteration. The unspecified conditions (e.g. outlet temperature) of the other stream are obtained from the heat balance. The pressure drops at both sides in the heat exchangers are not considered, while the pipe pressure loss for heat sink is included. The UA value for the heat exchanger is calculated from the capacity and the logarithmic mean temperature difference between the hot and the cold streams, based on division of heat exchangers into subsections of equal capacity for better estimation (Hagen et al. 2020). The total UA value is the sum of UA value of the HRHE and the condenser.

2.3.2. Generic heat exchanger (GHX) analysis

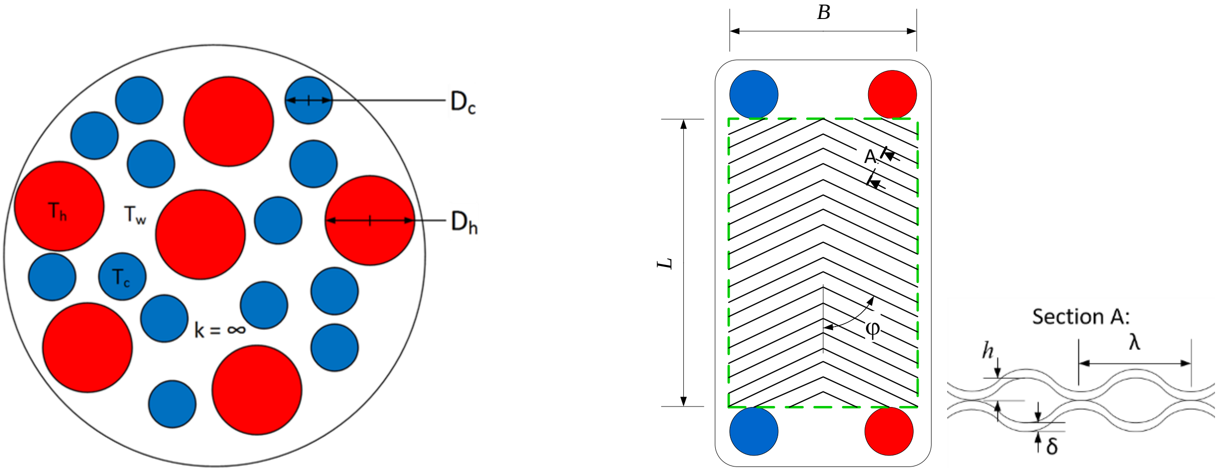
The analysis employs a generic heat exchanger model proposed by Hagen et al. (2020), which is an abstract representation of heat transfer mechanisms and is illustrated in Figure 2(a). The geometry of the generic heat exchanger is defined by five generic parameters, including hydraulic diameter and flow cross-sectional area for both streams, and flow length. Given the five parameters and fluid properties, the local heat transfer coefficient (HTC) and pressure drop can be calculated from thermal-hydraulic correlations. The generic heat exchanger model does not require specification of the heat exchanger type, but only appropriate correlations.

With given conditions of the cold and hot streams at one end of the heat exchanger, the temperature and pressure of the streams along the flow length can be solved. Meanwhile, the heat transfer area is obtained directly as the product of the perimeter and the flow length. The area of each heat exchanger is the average value of the heat transfer area of the cold and hot sides. The total heat transfer area is then the sum of the area of the HRHE and the condenser.

2.3.3. Plate heat exchanger (PHX) analysis

In the analysis, a model representing a plate heat exchanger is developed, which considers a more specific description of the heat exchanger geometries so that the mass and volume of the heat exchanger can be estimated. Figure 2(b) shows the geometry of a single, chevron-type plate. The blue and red circles show the inlet and outlet of the cold and hot streams, respectively. In the present study, only the geometries relevant to the effective heat transfer are considered, which is characterized by the plate width (B) and the plate length (L). The plates are corrugated in order to increase the turbulence, the thermal exchange surface and to provide mechanical rigidity to the exchanger. The corrugation is described by the chevron angle (φ) and the corrugation pitch (λ). The chevron angle is fixed at 60°, and the corrugation pitch is set to ensure a surfance enhancement factor of 1.25.

For modelling the plate heat exchanger, two other parameters are also included, which are the average distance between the plates, called channel height (h), and the number of plates. With the specified plate thickness (δ, currently set as 0.5 mm), these two parameters can give the height of the heat exchanger. The plate material is assumed to be stainless steel. The mass of the heat exchanger can be obtained from the plate length, width, thickness and the number of plates. The total mass is the sum of the mass of the HRHE and the condenser.



(a) (b)

Figure 2: (a) Generic heat exchanger model by Hagen et al. (2020) (b) A chevron-type plate of the plate heat exchanger model (effective heat transfer is defined to occur over the surface marked by the green box).

2.4. Optimization

The Rankine cycle is optimized to get the maximum net power output (defined by Eq. (1)) with the given constraints on the total UA-value, total area or total mass of the heat exchangers.

$$W_{net} = W_{exp} - W_{pump} - W_{source\ pump} - W_{sink\ pump} \qquad \text{Eq. (1)}$$

where W stands for the power, and the subscript "exp" represents the expander, and the terms related to pumps are corresponding to the components in Figure 1.

The optimization problem is solved by NLPQL (Schittkowski, 1986), which is a gradient-based method for solving constrained optimization problem. The variables and constraints related to the process applied for all the three methods are listed in Table 2. The only exception is that for the UA-analysis the pump outlet pressure and expander inlet pressure refer to the same variable, due to the absence of working fluid pressure loss. The constraint of no sub-atmospheric working fluid pressure is set since allowing vacuum pressure may result in air mixing with the working fluid in the case of leakage. The variables and constraints related to the heat exchangers for the three models are given in Table 3, as well as the fixed parameters for simulation.

Table 2: Common process variables, constraints, and objective function for all the three methods

Process variables	Mass flow rate of working fluid
	Mass flow rate of heat sink
	Expander inlet pressure
	Pump inlet pressure
	Pump outlet pressure
	Expander inlet temperature
Constraints (process)	Dry vapour at expander inlet
	Dry vapour at expander outlet
	No sub-atmospheric working fluid pressure
Objective function	Maximize net power output, Eq. (1)

Table 3: HX-related variables, constraints and fixed parameters

Variables	UA-value	Generic heat exchanger	Plate heat exchanger

HRHE		Cross-sectional area at working fluid side Length	No. of plates Width Channel height at working fluid side Channel height at source side
Condenser		Cross-sectional area at working fluid side Length	No. of plates Width Channel height at working fluid side Channel height at sink side
Constraints			
HXs geometry			The channel height ratio of two streams within 1.5
HXs size	Max total UA	Max total area	Max total mass
Fixed parameters	HXs pressure drop as 0	HRHE area ratio (hot/cold) as 1 HRHE source HTC 5000 W m^{-2} K^{-1} Condenser sink cross-sectional area as 0.47 m^2 Working fluid side hydraulic diameter as 10 mm (HRHE), 20mm (condenser)	Plate length-to-width ratio as 2

In the generic and plate heat exchanger analyses, the local heat transfer and pressure drop are calculated along the flow direction with the widely-used correlations, as given in Table 4. The results may slightly vary depending on the selection of heat transfer correlations considering applicable fluids and heat exchanger geometry; however, this effect is outside the scope of the present work. The thermodynamic properties of the fluids are calculated with REFPROP (Lemmon et al., 2013).

Table 4: Correlations of heat transfer coefficient and pressure drop used in generic and plate HX analysis

		Heat transfer	Pressure drop
Generic HX	**Single-phase**	Gnielinski (1976)	Selander (1978)
	Two-phase	Boyko and Kruzhilin (1967) (condensation) Bennett and Chen (1980) (evaporation)	Friedel (1979) with single-phase formulation by Selander (1978)
Plate HX	**Single-phase**	Martin (2010)	Martin (2010)
	Two-phase	Han et al. (2003) (condensation) Han et al. (2003b) (evaporation)	Han et al. (2003) (condensation) Han et al. (2003b) (evaporation)
Mixture effect	**Two-phase**	Silver (1947) and Bell and Ghaly (1973) for multicomponent condensation and evaporation	

2.5. Selection of working fluids

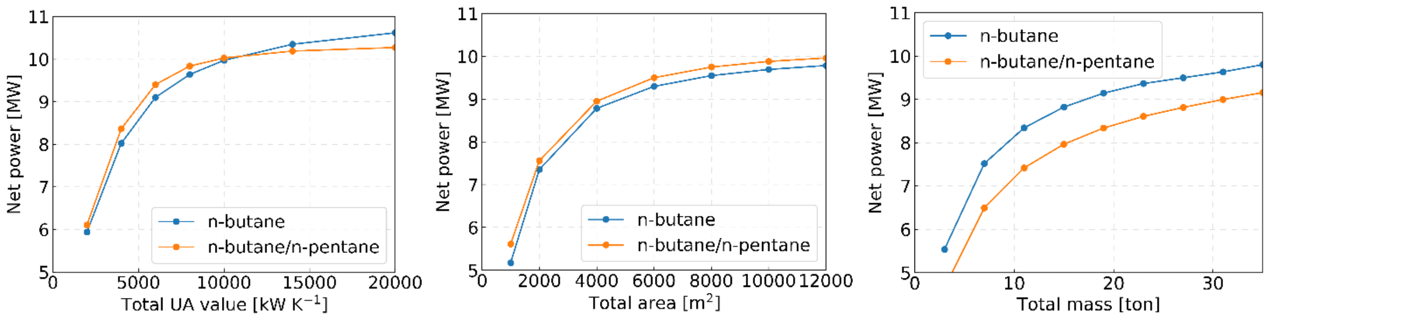
Several fluids were considered as working fluids given their low global warming and zero ozone depletion potential, which include n-butane, isopentane, isobutane, n-Pentane, propene, and propane. As the current work focuses on the comparison of methods rather than finding the best fluids, only one pure fluid and one mixture are investigated with the three different methods and presented here. With an initial screening based on UA-value method, n-butane is selected as the representative of the pure fluid, while n-butane/n-pentane with molar fraction 30%/70% is for the mixture given their better performances among the considered fluids. The R134a was also tested as a baseline. Specifically, the cycle with n-butane slighlty outperforms that with R134a over the considered range of UA value within 20000 kW K^{-1}, giving 5%–7% increase in net power output under the same UA value. The pure and mixture fluids in the following sections are referred to the two selected fluids thereafter.

2.6. Results and discussion

Figure 3 compares the net power output by using n-butane and n-butane/n-pentane mixture from the three levels of analyses. Under UA analysis in Figure 3(a), the cycle with the mixture gives higher net power output than the pure fluid when UA value is smaller than 10000 kW K^{-1}. The net power by the mixture then flattens out for UA above 10000 kW K^{-1}, while n-butane outperforms the mixture. The reason is that for the mixture, the condensation pressure reaches its lower limit of 1 bar, corresponding to a condenser outlet temperature of 19.6 °C. This restricts the pinch point temperature difference in the condenser. The mixture can achieve higher power output if the condensation pressure is allowed to be sub-atmospheric.

In the generic heat exchanger analysis illustrated in Figure 3(b), the mixture achieves slightly larger net power output over the whole range of total area considered. Moreover, the power output with both the pure and mixture fluids are smaller compared to the results from UA analysis. This can be clearly observed when plotting the total UA value against power from all the methods, as depicted in Figure 4. It is worth noting that the total UA value in UA analysis is input as an constraint, while it is calculated in the other two methods. The lower power output in the generic heat exchanger analysis compared to UA analysis is caused by additional exergy losses due to the presence of pressure loss in the heat exchangers. The comparison of these two methods indicates that pressure loss results in a larger exergy loss for n-butane than the n-butane/n-pentane mixture in the generic heat exchanger analysis.

Figure 3(c) shows the results from the plate heat exchanger analysis. N-butane performs better than the n-butane/n-pentane mixture over the whole range of mass considered. The contrary behaviours of the pure and the mixture fluids between the generic and plate heat exchanger analyses could be attributed to two main differences between the two methods. One is that the heat transfer and pressure drop correlations are different in the two models, and the other is the constraints on the heat exchanger geometries. The generic heat exchanger model is based on an abstract heat exchanger, in which there are no restrictions on the heat transfer area ratio or the cross-sectional area ratio between the cold and hot sides, except for one constraint applied on the heat transfer area ratio of the HRHE. The plate heat exchanger model represents a specific type of heat exchanger, which have the heat transfer area ratio of 1.0, and the constraint on the channel ratio within 1.5 gives the cross-sectional area ratio less than 1.5. By comparing the results from the two methods presented in Figure 4, these differences tend to have larger influences on the performance of the mixture than the pure fluid in the current case.



(a) (b) (c)

Figure 3: Net power against different (a) total UA value, (b) total area (GHX), (c) total mass (PHX).

Figure 5 depicts the overall heat transfer coefficient of the condenser in the generic and plate heat exchanger analyses. The heat transfer coefficients of the two fluids are similar at the same area in the generic heat exchanger analysis, while the heat transfer coefficient of the mixture is much smaller than n-butane at the same mass in the plate heat exchanger analysis. The overall heat transfer coefficients of the two fluids in the HRHE are found to be similar in both methods. Thus, the overall heat transfer coefficient of the condenser is one of the reasons that the mixture has the lower power output than n-butane under the same mass.

From the UA analysis to the plate heat exchanger analysis, more detailed description on the heat exchangers, and more geometrical variables and constraints have been included in the optimization model, which increases the model complexity and computational cost. The different results indicate that the selection of heat transfer correlation and the formulation of the optimization problem affect significantly the results. In particular, the generic and plate heat exchanger analyses conclude different working fluids as the best. This indicates the complexity of ORC-design analysis and that as realistic boundary conditions as possible should be considered even when the task is screening of working fluid candidates.

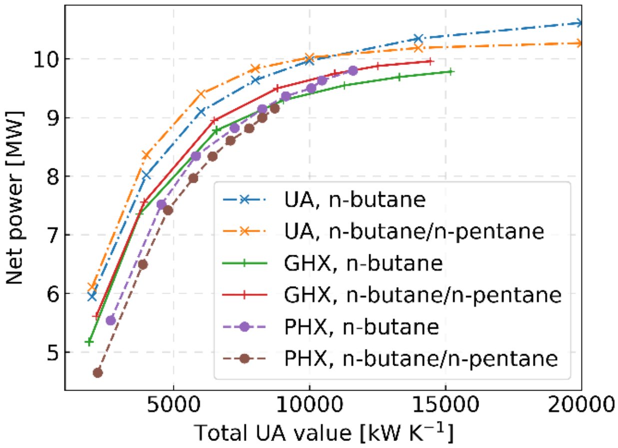
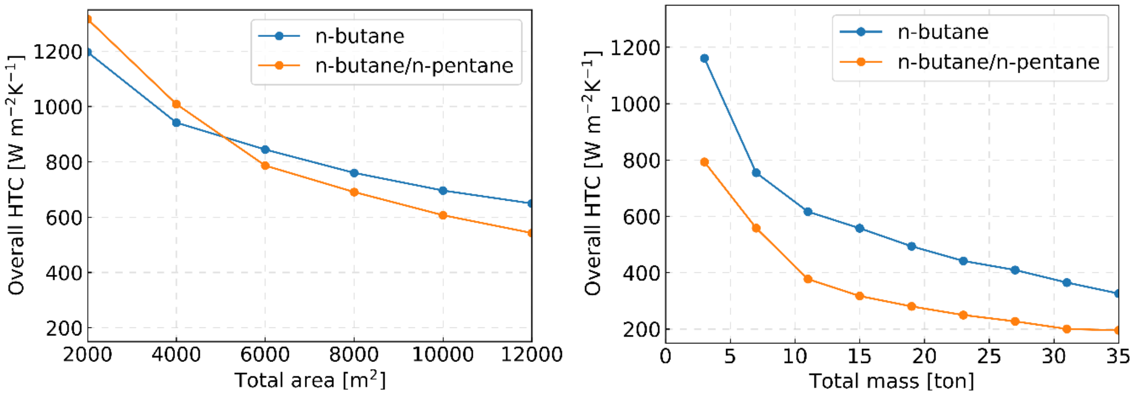


Figure 4: Net power against total UA value from the three different methods.



(a) (b)

Figure 5: Overall HTC of the condenser against (a) total area (GHX), (b) total mass (PHX).

3. CONCLUSIONS

This study applies three methods for design analysis of Rankine cycles utilizing waste heat from natural gas compression. They are 1) UA analysis, 2) generic heat exchanger analysis, and 3) plate heat exchanger analysis, which indicate the size of the heat exchagners in terms of total UA value, area and mass, respectively. All the methods involve solving a constrainted optimization problem with the objective of maximizing the net power output, but differ in the level of detail with respect to the heat excanger model and in the flexibility regarding the heat exchanger design. The methods were demonstrated considering two working fluids (n-butane and a 30%/70% n-butane/n-pentane mixture) and a wide range of heat exchanger sizes. In the UA analysis, the mixture outperforms the pure fluid for UA values below 10000 kW K^{-1}, while it is surpassed by the pure fluid for higher UA-values due to the constraint on condensation pressure. In the generic heat exchanger analysis, the mixture achieves larger power output than the pure fluid over the whole range of heat exchangers area considered. In constrast, the pure fluid performs better than the mixture over the whole range of heat exchangers mass considered in the plate heat exchanger analysis. The different results indicate the complexity of ORC-design analysis and that as realistic boundary conditions as possible should be considered even when the task is screening of working fluid candidates.

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