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Effect of Waste Heat Utilization on the Performance of Low Temperature Rankine Cycle

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ABSTRACT

Low temperature Rankine cycle is a prominent solution for power generation in Waste Heat Recovery (WHR) application. The performance of this cycle is affected by various parameters including characteristics of the heat source, working fluid and constraints in the system. In cases where the heat source has a limited mass flux and therefore variable temperature, the amount of extracted heat affects the performance of the cycle including net power and efficiency which is also related to the working fluid. This is in the paper expressed in terms of a heat utilization factor, ψ, which shows the ratio of extracted heat to the maximum possible extraction rate in the specific case. This factor affects the performance of the cycle by moving the pinch point location in the evaporator. Results indicate that this factor has great impact on the performance of the cycle and the effect varies for different working fluids.

Keywords: Rankine cycle, Waste Heat Recovery, Heat Utilization Factor

1. INTRODUCTION

The low temperature Rankine Cycle (RC) is a prominent solution for power generation in waste heat recovery application. Fig. 1 shows the basic layout of an RC cycle including pump, evaporator, expander and condenser. In a recuperative cycle, a recuperator, i.e. internal heat exchanger, is added to extract excess heat at expander outlet and feed it to the fluid before the evaporator.

A very important question is raised here. Is it beneficial to extract as much heat as possible from the heat source? In other words, what is the optimum waste heat utilization factor for a specific case? Is it feasible to extract heat in two steps for medium to high temperature heat sources? Clearly the answers to these questions depend on different factors including heat source inlet temperature, Pinch Point Temperature Difference (PPTD) and other practical limitations such as maximum allowable evaporator pressure.

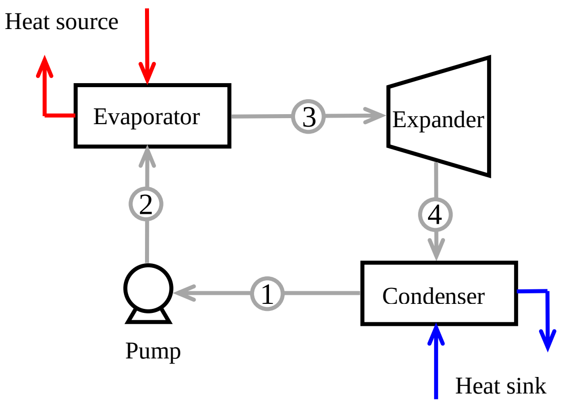

Figure 1 – Basic layout of RC

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1.1. Waste Heat Utilization Factor

The waste heat utilization factor is defined as the fraction of the extracted heat to the maximum potential in the heat source for the given boundary conditions. This maximum value depends on the heat source and the heat sink inlet temperatures, here assuming zero PPTDs and unlimited heat sink mass flow rate. It means the heat source is cooled down until it reaches the temperature of heat sink. Clearly, this is a theoretical maximum and is obviously not possible to be achieved under practical conditions. The mass flow rate of the heat sink is thus not taken into account in this definition, because it is assumed that only the heat source information is given, and heat sink mass flow rate can vary and be adapted to the specific design of the cycle.

Therefore, the waste heat utilization factor is given by:

$$\psi = \frac{h_{h,in} - h_{h,out}}{h_{h,in} - h_{c,in}} \tag{1}$$

This ratio is defined based on the difference in enthalpies of the heat source. Assuming the specific heat of the heat source is constant, this ratio could be rewritten in the forms of temperature differences:

$$\psi = \frac{T_{h,in} - T_{h,out}}{T_{h,in} - T_{c,in}} \tag{2}$$

In case of specific inlet temperatures of the heat source and the heat sink, ψ is only a linear function of the heat source outlet temperature. Therefore, effects of changes in ψ could be investigated based on variations in the heat source outlet temperature.

This definition is valid only for the basic configuration, where there exists just one cycle with only one heat source and heat sink (Fig. 2).

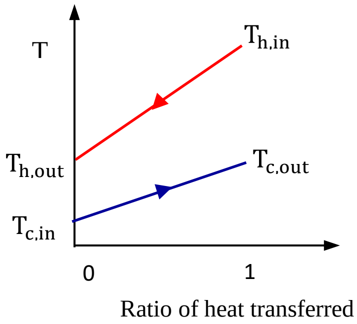

Figure 2 – Basic configuration of RC (one cycle)

In case of a two-stage configuration, this definition needs some minor changes to adapt for each configuration, which will be dealt with in the following section.

Öhman has provided another definition for waste heat utilization factor (Öhman and Lundqvist., 2013). In that definition, he takes into account inlet temperatures and the mass flow rates of both the heat source and the heat sink. Maximum heat utilization factor ($\psi = 1$) is reached when the outlet temperatures of the heat source and the heat sink are equal. This is true for cocurrent configuration of heat exchangers, and the maximum utilization factor thus changes with heat sink mass flow rate. However, in the definition in this paper, the mass flow rate of the heat sink is not given and will vary and be adapted based on the designed cycle.

The basic configuration of RC is shown in Fig. 2 with one running cycle. However, the configurations could be extended to two cycles with different arrangements of the heat sink and the heat source. There are multiple possibilities here: either of the heat source or the heat sink could be arranged in Parallel (P) or Series (S) arrangement. Therefore, there exist 4 different configurations of RC with two cycles, in total: PP, PS SP and SS. The first letter represents the arrangement for the heat source and the second letter shows the arrangement for the heat sink. Fig. 3 shows the layout of different configurations. In a parallel arrangement, the flow (either heat sink or heat source) is divided into two parallel flows and passes through the cycles. On the other hand, in the series arrangement, the output of heat flow from the first cycle enters the second cycle. The numbering

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of the cycles is based on the order in which the heat source passes through cycles. In PP configurations, the cycles are identical, and numbering does not matter and only in PS configuration, the numbering is based on the order in which heat sink passes through the cycles.

In case of having a relatively high mass flow rate, there is a potential to use the parallel arrangement. On the other hand, if the inlet temperature of the heat source is relatively high, it is possible to extract heat in two steps in series mode. Besides, if the condensing temperature of the cycle is high, it pushes up the heat source outlet temperature, which brings the possibility for the rest of the heat to be extracted in a second cycle with series heat source arrangement.

For SS configuration, there could be two different possible cases depending on the order in which the heat source and the heat sink enter cycles. In the first case, the heat source first enters cycles 1 and the heat sink first enters cycle 2. On the other hand, in the second case, both the heat source and the heat sink enter cycle 1 at first, and then pass through cycle 2. The difference between these two cycles is the potential net power distribution between case 1 and 2. Moving from case 1 to case 2, the power generation potential for cycle 1 increases, but decreases for cycle 2. The reason is due to the change in the heat sink inlet temperature for each of the cycles which affects the condensing temperature of the fluid. The higher the condensing temperature, the lower power generation potential becomes. If the condensing temperature of the cycle 1 is high, case 1 is preferred over case 2. Therefore, higher sink temperature for cycle 1 would not decrease the power potential of that cycle. Detailed explanation of condensing temperature is provided in section 3.

From a thermodynamic performance point of view, PP configuration is the same as the base case with only one cycle. Although the mass flow rate in each of the cycles is half of the value in the basic configuration and the size of components are different, net power, thermal efficiency and waste heat utilization factor are the same.

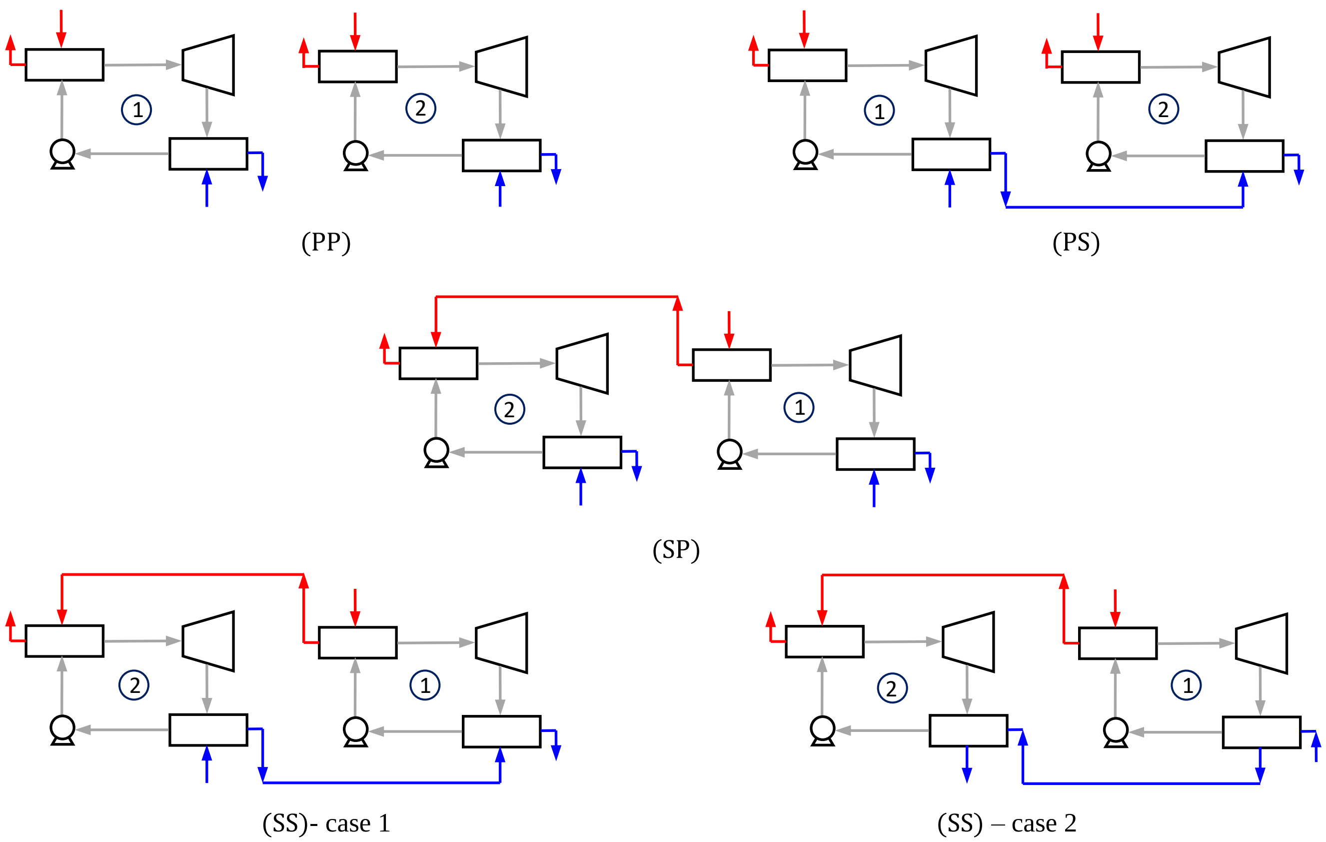

Figure 3 – Different potential 2-cycle configurations, red and blue lines show the heat source and the heat sink respectively. The cycle numbers are written inside each cycle. All heat exchangers are counterflow.

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Definition of waste heat utilization factor needs a slight change for the two-cycle configurations. The maximum potential is the same as the base case and is defined based on inlet temperatures of heat source and heat sink. The actual extracted heat is the summation of extracted heat in both cycles, either parallel or series.

2. ASSUMPTIONS

The minimum condensing pressure has been set to atmospheric pressure to avoid air suction into the system therefore for fluids with critical temperature almost higher than 200°C, the minimum condensing temperature rises above 25°C. Pure natural refrigerants in REFPROP 10.0 (Lemmon et. al, 2018) were preselected for optimizations in a sub-critical RC and 1 kg/s heat source air flow to maximize net power. Analyses were done in steady state neglecting friction losses and losses to the environment. Pump and expander were modelled with fixed isentropic efficiency and pinch point analysis was employed for heat exchanger analysis. Expander inlet conditions were at saturated vapor. The heat exchangers were in counter-current configuration. Table 1 shows the fixed parameters for optimization.

Table 1: List of fixed parameters in the optimizations

Parameter	Value
Heat source	Air with inlet temperature 150-250 °C
Heat sink	River or sea water with inlet temperature 10 °C
Ambient temperature and pressure	10 °C, 101.325 kPa
Pump isentropic efficiency	0.9
Expander isentropic efficiency	0.8
Minimum condensing temperature and pressure	25 °C, 101.325 kPa
Minimum PPTD	10 °C

3. RESULTS

As mentioned in section 2, the condensing conditions are set to satisfy minimum 25°C and atmospheric pressure. For fluids with critical temperature higher than 200°C, the condensing temperature rises above 25°C and for fluids with critical temperatures of almost higher than 240°C, condensing temperature becomes so high that it is not feasible to use the fluid, due to the sharp decrease in the net power. (Shahrooz and Lundqvist, 2019). Therefore, in this study natural refrigerants with a critical temperature up to 240°C are considered. In order to study the effects of the heat source inlet temperature and the PPTD in the evaporator, a new parameter is defined as effective heat source temperature:

$$\mathrm{T_{h,eff}} = T_{h,in} - \mathrm{PPTD_{evap.}} \tag{3}$$

This parameter takes into account the variations of both the heat source inlet temperature and the PPTD in the evaporator. In Fig. 4, net power is plotted vs. the evaporator temperature at different effective heat source temperatures for neopentane. In this graph, the optimum evaporator temperature changes with effective heat source temperature, which means both the heat source inlet temperature and the evaporator PPTD affect the optimum operating conditions. For a specific heat source inlet temperature, by increasing the PPTD in the evaporator, the effective heat source temperature decreases, which means that optimum working conditions change, and the current points are no longer optimum. The optimum point moves towards lower evaporating temperature (pressure).

Fig. 5a shows the net power vs. the heat source outlet temperature for different heat source inlet temperatures and the evaporator PPTD of 10°C. For a specific heat source inlet temperature and evaporator PPTD, not all range of the heat source outlet temperature could be covered by different working conditions (evaporator pressure). If the effective heat source temperature is less than the critical temperature of the fluid, a wide range of waste heat utilization factor is covered. In this case, the location of the pinch point is always at the saturated liquid. On the other hand, by increasing the effective heat source inlet temperature, the pinch point location moves towards the subcooled liquid section of the evaporator i.e. range of evaporator inlet up to the saturated liquid. When the effective heat source temperature is much higher than the critical temperature of the working fluid, the location of the pinch point in the evaporator becomes locked by the evaporator inlet point. Therefore, heat source outlet temperature will be locked and is not possible to be varied in a wide range.

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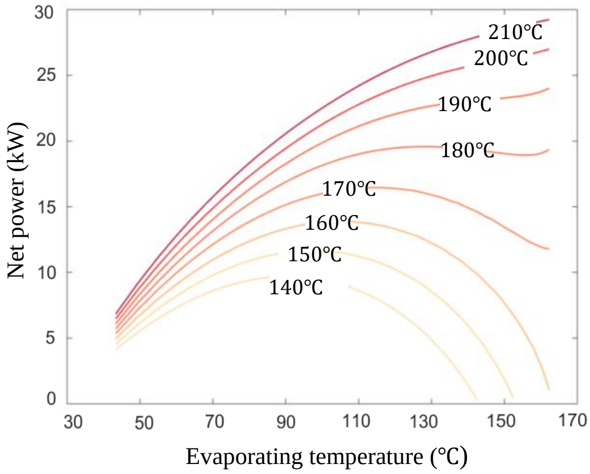
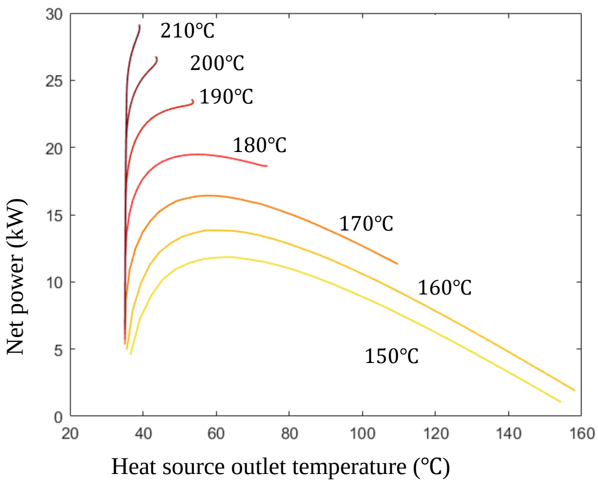
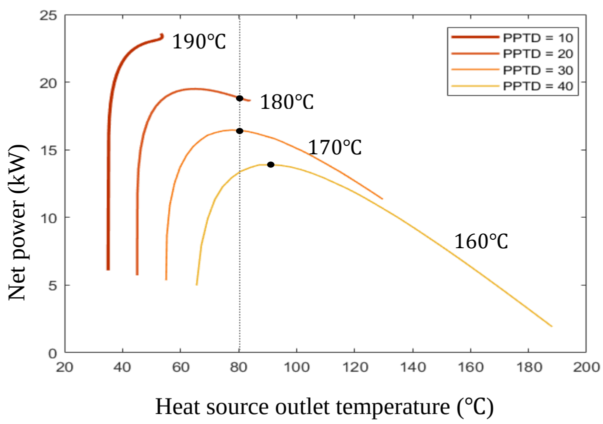


Figure 4- Net power vs. evaporator temperature at different effective heat source temperatures for neopentane in subcritical cycle. Each number on the curve shows the effective heat source temperature.

To have a variable heat source outlet temperature, different PPTD in the evaporator should be used. This is shown in Fig. 5b. For a specific heat source inlet temperature, if the evaporator PPTD increases, effective heat source temperature decreases. The optimum evaporator temperature will be as shown in Fig. 4. However, for heat source outlet temperature graph, the shape of the curve will be exactly similar to the same effective heat source temperature, but with a shift towards the right side. This shift is equal to the increase in the evaporator PPTD (Fig. 5b)



5a: Fixed evaporator PPTD of 10℃ and various heat source inlet temperatures of 160-220℃



5b: Fixed heat source inlet temperature of 200℃ and various evaporator PPTD of 10-40℃. The black points show optimum points with a heat source outlet limitation of 80 ℃

Figure 5- Graph of net power vs. heat source outlet temperature for various effective heat source temperatures for neopentane in subcritical cycle. Each number next to each curve shows the effective heat source temperature.

It is important to note that the waste heat utilization factor is not the only parameter affecting the performance of the cycle. Depending on the evaporator pressure and PPTD, and cycle type (sub-critical, trans-critical), the performance of the cycle could be totally different.

In some cases, there could be a limitation for the heat source outlet temperature when there are certain components in the heat source which will condense in case the temperature goes below the condensation temperature (Astolfi et al., 2017). In this case, the limitation should be taken into account while designing the

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cycle. Otherwise, the operating conditions are not optimum any more. If the heat source outlet temperature range for minimum evaporator PPTD is less than the minimum allowed value, the evaporator PPTD should be increased to fulfill the limitation condition, which means the effective heat source temperature is not the same any more, and as explained before, the optimum operating conditions have changed. In this case, there could be different PPTDs that satisfy this condition. However, if the PPTD is higher, the maximum net power decreases, but on the other hand, the level of the evaporator pressure decreases. If the evaporator pressure is within the allowable pressure range, the aim should be for minimum possible PPTD. Otherwise, a trade-off should be done between the evaporator pressure and the net power. This is shown in Fig. 5b. For example, with heat source inlet temperature of 200°C and minimum heat source outlet temperature of 80°C and neopentane as working fluid, the evaporator PPTD should be almost at least 20°C. The higher the PPTD, the less net power, but lower evaporator pressure.

If the heat source outlet temperature of the optimum point is high enough, it could provide the necessary heat for a second cycle in series arrangement. Fig. 6 shows the maximum potential in power generation for heat source temperature range of 80-160°C for the second cycle in series arrangement of the heat source. Below heat source inlet temperature of 120°C, different working fluids have very close net power potential. However, above this temperature propylene and propane have higher net power potential.

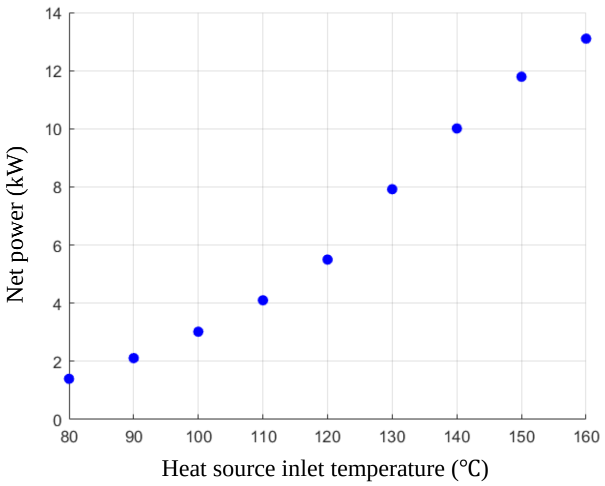

Figure 6- Maximum net power vs. heat source inlet temperature for range of 80-160°C and PPTD of 10°C in the evaporator

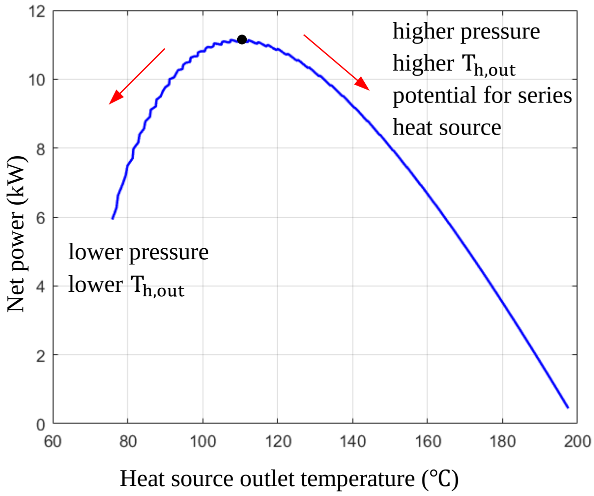

Figure 7- Net power vs. heat source outlet temperature with heat source temperature of 200℃ and PPTD of 10℃ for acetone

In subcritical cycles, there is generally no benefit of extracting heat in two steps in series mode for the heat source, unless the optimum heat source outlet temperature is high enough to provide heat for a second cycle (Fig. 7). It could also happen due to high PPTD in the evaporator of the first cycle, but is not feasible due to sharp decrease in maximum net power.

There is also another way to boost the total net power in series arrangement for the heat source, and that is a lower waste heat utilization, ψ, in the first cycle and thus providing more heat for the second cycle. It should also be noted that less waste heat utilization factor in the first cycle means higher evaporator pressure (with the same evaporator PPTD), which should not exceed maximum allowable evaporator pressure set by the designer. This is shown in Fig. 8 where a heat source of 200°C with acetone as working fluid is modelled in 2 cases: (1): with only one cycle where the optimum net power is 11.16 kW with corresponding heat source outlet temperature of 110°C, (2): including a second cycle with series heat source arrangement. As it is clear, by lowering the heat utilization in the first cycle (corresponding heat source outlet temperature of 150°C) and using it for the second cycle, there is a possibility to maximize total net power from 15.26 to 19.81 kW.

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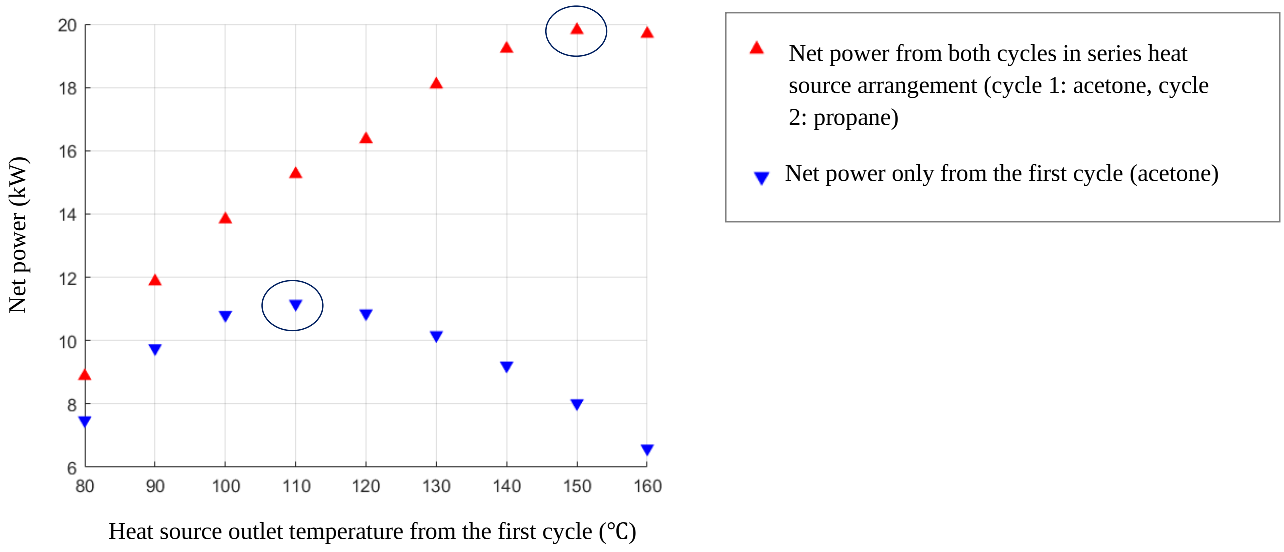

Figure 8- Net power vs. heat source outlet temperature with heat source temperature of 200℃ and PPTD of 10℃ for acetone in 2 cases: only 1 cycle with acetone, and 2 cycles with series heat source arrangement in which the first cycle is with acetone and for the second cycle with propane

In this specific case, both SP and SS- case 1 could be considered. As the condensing temperature of acetone is 56°C, no net power potential is lost due to higher heat sink inlet temperature in SS- case 1 configuration.

In case of adding recuperator to the cycle, the net power stays the same, but the necessary extracted heat from the heat source decreases. This brings the possibility for series arrangement of the heat source. On the other hand, if the working fluid, operating conditions and isentropic efficiency of the expander are chosen in a way to minimize superheat at expander outlet, the need for recuperative cycle could be disregarded (Shahrooz and Lundqvist, 2019).

4. CONCLUSIONS

In this paper, the effects of different waste heat utilization factors was studied on the performance of a low temperature Rankine cycles working with pure natural refrigerants. Clearly, the design objective should not be to extract as much heat as possible from the heat source. The objective is rather to generate maximum power with respect to limitations of the specific case i.e. maximum evaporator pressure, minimum heat source outlet temperature (if the limitation exists) and other conditions. In case the optimum heat source outlet temperature is high enough, it could also have the potential to provide heat for a second cycle in a series configuration. By varying the waste heat utilization in the first cycle and providing the rest of the heat to the second cycle, it is possible to find a point with higher total net power. Obviously economic factors must be taken into account as well for a full optimization.
A series arrangement for the heat sink is only feasible when there is a limitation on heat sink mass flow rate and/or the heat source is in series arrangement and condensing temperature in the first cycle is high.

When the effective heat source temperature is less or slightly higher than critical temperature of the fluid, it is possible to control the waste heat utilization factor. Otherwise it is not possible to control it and the only way to vary the waste heat utilization factor over a wide range would be by increasing PPTD in the evaporator, which leads to sharp decrease in net power potential. This implies that selection of the appropriate working fluid is essential.

For fluids with the critical temperature up to 200°C, it is possible to match heat source temperature with the fluid in order to have active control on the waste heat utilization factor. For fluids with the critical temperature between 200 and 240°C, the condensing temperature increases to reach the minimum atmospheric pressure. Therefore, a portion of the potential net power is lost due to higher condensing temperature. In this case, the heat source outlet temperature has the potential to provide heat for a second cycle in series arrangement (for

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the heat source) and increase the total net power from both of the cycles. For fluids with critical temperature of almost higher than 240°C, the condensing temperature rises so high that it is almost unfeasible to use as working fluid. Therefore, for effective heat source temperatures more than 240°C, it is not possible to have an active control on waste heat utilization factor.

If the heat source temperature is less than the critical temperature of the fluid, it is possible to use waste heat utilization factor as a variable and control it. However, as the heat source temperature rises, the range becomes smaller and smaller until it becomes totally fixed and is not possible to be varied.

The results also suggest that the concept of waste heat utilization should not just be dealt with in forms of a non-dimensional ratio (ψ). The temperature level of the heat source outlet temperature matters which is a determining factor if a second cycle with series heat source arrangement could be added.

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NOMENCLATURE

h	Specific enthalpy of heat source (kJ.kg^{-1}.K^{-1})	T	Temperature (°C)
$T_{h.eff}$	Effective heat source temperature (°C)	ψ	Waste heat utilization factor

Subscripts

c	Heat sink	evap.	Evaporator
h	Heat source	in	Inlet
out	Outlet		

Acronyms

P	Parallel arrangement	PPTD	Pinch Point Temperature Difference (°C)
RC	Rankine Cycle	S	Series arrangement
WHR	Waste Heat Recovery		

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