

EFFECT OF WORKING FLUID TYPE ON LOW TEMPERATURE RANKINE CYCLE OPTIMIZATION

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ABSTRACT

The characteristics of the working fluid play an important role on the performance of low temperature Rankine cycles. One important criterion categorizes fluids based on the slope sign of the saturated vapor curve in T-s diagram to distinguish between so-called wet and dry expansion. Wet expansion is preferably avoided due to the damage inflicted on turbine blades by liquid droplets. On the other hand, superheat in expander inlet and outlet increases the loads on heat exchangers and increases cost of the system. Therefore, in this paper a methodology is presented to minimize expander superheat, while maximizing net power and taking the isentropic efficiency of the expander into account. Results indicate that so-called wet fluids do not necessarily need to be excluded from the fluid candidate selection. Depending on the evaporating and condensing temperature levels, by using an expander with a specific isentropic efficiency, it is possible to decrease the superheat in the expander and in some cases close to zero. However, comparison should be between the performances of various fluids in a cycle with all external parameters and fluids should not be excluded from the screening list only based on their types.

1. INTRODUCTION

Rankine Cycles (RCs) and Vapor Compression Cycles (VCCs) are two important thermodynamic cycles for energy conversion which are widely used. RC is a heat engine that converts heat to mechanical energy and generates power in industrial applications. On the other hand, VCC such as heat pump or refrigeration cycle uses mechanical work to move heat from a low temperature level to a high temperature level used both in industrial and house-hold applications. The characteristics and properties of the refrigerant used in the cycle have significant influence on cycle performance along with the other operating conditions and constraints of the system. An important criterion which has been extensively discussed for fluid candidate selection is the type of the fluids, which categorizes refrigerants into three types of isentropic, wet and dry based on the slope sign of the saturated vapor curve in T-s diagram (Chen *et al*, 2010). If the curve slope is negative ($ds/dT < 0$), the fluid is defined as ‘wet’ and if the slope is positive ($ds/dT > 0$), the fluid is categorized as ‘dry’. In case of almost vertical saturated vapor curve ($ds/dT \approx 0$), the fluid is considered isentropic. It should be noted that this definition is regardless of the small region below the critical temperature where the slope is always negative. This definition complies with the isentropic expansion starting from saturated vapor state and the quality of the fluid along the process. As shown in Fig. 1, in case of a wet fluid, the expanded fluid would be in two-phase region, but for a dry fluid, the refrigerant would be only in superheated vapor state. For isentropic fluids, the expansion process is very close to the saturated vapor curve which makes expansion output almost

saturated vapor. As mentioned above, the small region below the critical temperature is not taken into account for dry and isentropic fluids. Otherwise, the fluid will pass two-phase region during isentropic expansion.

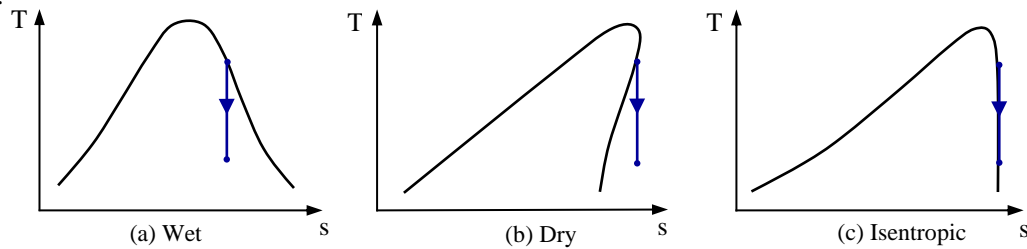


Figure 1: Three types of fluids based on traditional classification: (a) wet, (b) dry and (c) isentropic

These definitions are consistent with RC application where the fluid undergoes expansion process. However, in VCCs, the terms ‘wet’ and ‘dry’ need to be redefined based on the slope sign. If the slope is positive, the fluid undergoes wet compression and if the slope is negative, the compression is dry. Therefore, the terms ‘wet’ and ‘dry’ depend on the type of the process, the fluid undergoes and cannot be solely defined based on the saturation vapor curve. If the objective is to use a single definition for a fluid regardless of the type of the process, it should be based on the slope sign of the saturated vapor curve in T-s diagram without using the terms ‘wet’ or ‘dry’. Consequently, in this paper, ‘wet’ and ‘dry’ fluid terms will be only used together with the type of the process (expansion or compression). When referring to the specifications of the fluid, ‘NS’ (Negative Slope), ‘PS’ (Positive Slope), and ‘isentropic’ terms will be used.

Going back to the classification of the fluids by slope, traditionally, three types of fluids are defined which are used in most publications: isentropic, PS (formerly dry) and NS (formerly wet). There are few isentropic pure fluids where the saturated vapor curve is nearly vertical. However, in novel classification, new categories for fluids are also defined which are included as sub-type of isentropic fluids. For these fluids, the saturated vapor curve in T-s diagram has both a positive and a negative slope section (Györke *et al.*, 2018). At lower temperatures, the slope is negative and at higher temperatures, the slope is positive. In these fluid types, based on the temperature levels of the expansion/compression inlet and outlet, the process could be wet, dry or on saturated vapor curve. In this classification, primary and secondary characteristic points are used to distinguish the different types of fluids. Primary points are A, C, Z where C is the critical point and A and Z are the lowest temperature points on saturated liquid and vapor curves, respectively. In general, for a pure fluid, this temperature is the triple point. However, based on the application temperature range, a higher value could be chosen for the minimum temperature. NS fluids have only primary points. Secondary characteristic points are M and N which are the local maximum and minimum on the saturated vapor curve in T-s diagram, respectively where $dT/ds = 0$. Point N only exists for isentropic fluid types and as shown in Fig. 2c, by choosing a minimum temperature above point N (shown by dashed grey line), the fluid would be classified as PS type. Different fluid types and their characteristic points are shown in Fig. 2. For isentropic fluids in traditional definition, where the saturated vapor curve in T-s diagram is almost vertical, both points M and N exist but their entropies are much closer to each other which makes the slope in that region almost vertical. The classification of the fluid types could be extended to different subtypes in PS and isentropic types, based on the relative position of primary and secondary characteristic points to each other in T-s diagram. However, it would be beyond the scope of this paper to deal with sub-types.

The very important question here, is why the fluid types could be important in working fluid candidate selection. Both in RC and VCC, wet expansion/compression processes are avoided due to the risk for erosion of the blades in turbines and compressors, respectively. Therefore, the aim is avoiding the two-phase region during the expansion/compression process. Otherwise, superheat would be necessary at expander/compressor inlet to avoid wet region. On the other hand, if the expansion/compression process is very dry, high superheat would be expected at the end of the process. This resulting superheat could be beneficial for heat pump cycle, but not for refrigeration cycle and RC by putting more heat load on

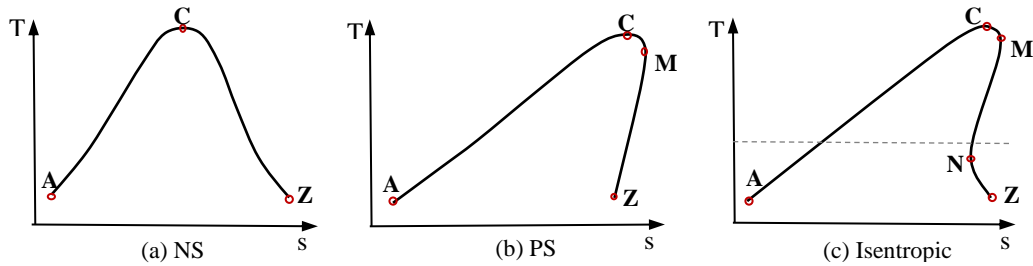


Figure 2: Novel classification of fluids with primary and secondary characteristic points in T-s diagram

the heat exchanger dissipating the heat to the environment. While the previous sections covered the relation of the fluid type to both compression and expansion processes, the rest of the paper will be only focused on the expansion process in RCs. Superheat in expander includes two parts: required superheat before the expander, and resulting superheat after the expander. In most cases, to avoid wet expansion, a combination of both of them might be present based on the fluid type, expander isentropic efficiency and evaporating and condensing temperature levels. In RC, adding superheat at expander inlet decreases the net power and the superheat both at expander inlet and outlet puts more load on the evaporator and the condenser respectively. Superheat at expander outlet could be recovered in a recuperator, but comes at a higher cost of adding another heat exchanger.

Therefore, the objective is to make the expansion process inlet and outlet close to the saturated vapor state to minimize superheat and meanwhile avoid entering the wet region. In an ideal isentropic process, it means to use isentropic fluids. However, in real processes, irreversibilities result in increased entropy in the expansion output which pushes the expander outlet point to the right side in T-s diagram. As a result, by considering expander isentropic efficiency, NS fluids could also be considered as candidates to minimize superheat. Isentropic fluids could be also used, depending on the application temperature range. The same logic could also be discussed for zeotropic mixtures. Zeotropic mixtures exhibit variable temperature phase change line or in other words, temperature glide which brings the potential for better temperature profile matching in cases where the heat source or heat sink temperature is variable. Efforts have been made to design isentropic mixtures for RC (Mondejar and Thern, 2016) and heat pump (Zheng *et al.* 2016) in the ideal case. Since different parameters might play role in RC optimization, the aim of this paper is to investigate the effects of different types of fluids on RC optimization with pure natural fluids in real expansion process and examine if expander isentropic efficiency could be used to choose fluids to minimize superheat.

2. CONCEPT OF NON-DIMENSIONAL SUPERHEAT PARAMETER

The parameter ξ is a proxy for the slope of the saturated vapor curve in T-s diagram. The parameter was originally introduced by Bertinat (1985).

$$\xi = \frac{\beta Lv}{C_p \Delta v} \quad (1)$$

Where ξ is non-dimensional superheat parameter, β isobaric coefficient at thermal expansion (1/K), L latent heat of vaporization (kJ/kg), C_p specific heat of saturated vapor (kJ/kg.K), v specific volume of saturated vapor (m^3/kg) and Δv specific volume difference of saturated liquid and vapor (m^3/kg).

By using Maxwell equation and combining it with Clausius-Clapeyron equation:

$$ds = \frac{C_p}{T} (1 - \xi) dT \quad (2)$$

s is specific entropy (kJ/kg.K) and T is temperature (K). This non-dimensional parameter ξ could be used to identify fluid types. Since the state points are far from the critical point, the term $v/\Delta v$ is close to 1 and ξ becomes mostly dependent on the ratio of L/C_p . The higher this ratio, the more negative slope and for lower ratios, the fluid has positive slope. The relation between ξ and slope of the saturated vapor curve in T-s diagram is summarized in Table 1.

Table 1: Relation between ξ and the slope of the saturated vapor curve in T-s diagram

$\xi < 1$	$\xi \approx 1$	$\xi > 1$
$(1 - \xi) > 0$	$(1 - \xi) \approx 0$	$(1 - \xi) < 0$
Positive slope	Vertical slope	Negative slope

As mentioned above, superheat in an expansion process could include either or both of required superheat at the expander inlet (in the evaporator) and resulting superheat at the expander outlet (in the condenser) which occur at different temperature levels. Superheat is mostly dealt with in terms of temperature difference, known as ‘*SuperHeat Degree*’. However, a few degrees of superheat at expander inlet might put more load on the evaporator compared to more degrees of superheat in expander outlet which puts load on the condenser. Therefore, this concept should be also dealt with in terms of heat. ‘*Specific superheat*’ is the specific enthalpy difference between the superheated and saturated vapor. On the other hand, ‘*superheat load*’ also considers the mass flow rate of the working fluid and is expressed in terms of (kW). Examples of required and resulting superheat are shown in Fig. 3. For PS and isentropic fluids, if the expansion starts below point M, only resulting superheat would be present for all values of isentropic efficiency. However, if the expansion starts from a higher pressure above point M, required superheat would be also necessary. On the other hand, for NS fluids, a combination of expected and resulting superheat might be present based on the isentropic efficiency of the expander (η).

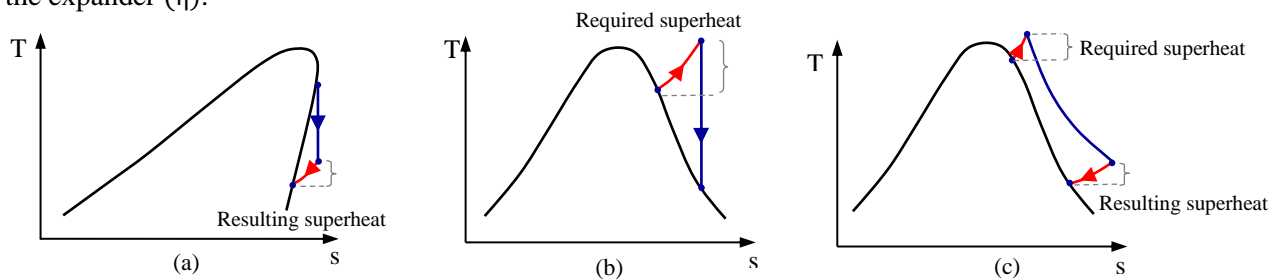


Figure 3: Superheat for: (a) PS, $\eta = 1$, (b) NS, $\eta = 1$, (c) NS, $\eta \neq 1$

3. ASSUMPTIONS AND METHODS

Pure natural refrigerants in REFPROP 10.0 database were preselected for optimizations in a sub-critical RC and 1 kg/s heat source air flow to maximize net power, which is defined by subtracting pump power from expander work. Consequently, studied fluids had zero ODP (Ozone Depletion Potential) and low GWP (Global Warming Potential). Minimum condensing pressure was set to 1 bar to avoid the risk for air suction to the cycle. Analyses were done in steady state neglecting friction losses and losses to the environment. Wet expansion was also avoided. Pump and expander were modelled with fixed isentropic efficiency and pinch point analysis was employed for heat exchanger analysis. Table 2 shows the fixed parameters for optimization. Polytropic efficiency was used to calculate the state points in between the expansion process to avoid wet expansion (Wettstein, 2015).

Table 2: List of fixed parameters in the optimizations

Parameter	Value
Heat source	Air with inlet temperature 150, 200, 250 °C
Heat sink	River water with inlet temperature 10 °C
Ambient temperature and pressure	10 °C, 101.325 kPa
Pump isentropic efficiency	0.9
Expander isentropic efficiency	0.5, 0.7, 0.9
Minimum condensing temperature and pressure	25 °C, 101.325 kPa
Pinch point temperature difference	10 °C

4. RESULTS

As it is displayed in Fig. 4, in general, NS fluids have lower critical temperatures compared to PS and isentropic fluids. However, there are a few exceptions. Condensing temperature also plays an important role in optimizations based on net power. Clearly, the higher the condensing temperature, the lower net power would be. For the fluids with critical temperature higher than 200°C, the minimum condensing temperature rises above 25°C to compensate for the minimum atmospheric pressure resulting in lower maximum net power. Therefore, fluids with much higher condensing temperature, mostly including PS and isentropic fluids (critical temperature > 240°C) are excluded from the optimizations. Besides, fluids with very low critical temperature (NS type) are also excluded to make condensation possible.

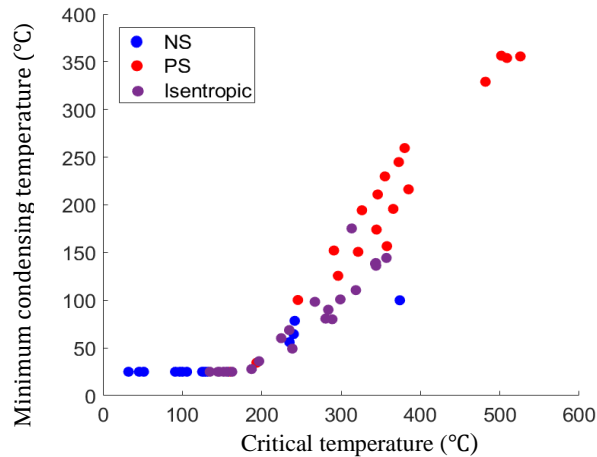


Figure 4: Minimum condensing temperature for pure natural fluids with different critical temperature and types

In Table 3, list of pure natural refrigerants is presented together with their critical temperature and type based on the color coding in Fig. 4. It should be noted that this list is just to show the trend of optimization results for different fluids and as an example, hydrogen sulfide, carbonyl sulfide and sulfur dioxide, are not considered safe refrigerants for working fluid suggestion.

Table 3: List of pure natural refrigerants with their critical temperature and type

propylene 91.1°C	propyne 129.2°C	sulfur dioxide 157.5°C	hexane 234.7°C
propane 96.7°C	isobutene 134.7	neopentane 160.6°C	acetone 235.0°C
hydrogen sulfide 100.0°C	isobutene 144.9°C	cis-butene 162.6°C	cyclopentane 238.6°C
carbonyl sulfide 105.6°C	butane 146.1°C	isopentane 187.2°C	
cyclopropane 125.2°C	butane 152.0°C	pentane 196.6°C	
dimethylether 127.2°C	trans-butene 155.5°C	isohexane 224.6°C	

Here, we conducted the study in two stages:

- Investigating the characteristics of fluids regardless of external parameters
- Investigating the whole system together with heat source, heat sink and limitations of the system.

The first approach focuses on behavior of the fluids solely, in a single cycle, while the second approach seeks the optimum fluids for given heat source and heat sink conditions. Conclusions from each perspective could be different since various parameters play the key role in each of them.

4.1 Fluid Only

In this approach fluid mass flow rate is not included and the results are only about the fluids. As mentioned in section 2, the superheat should be dealt in terms of enthalpy, not just temperature difference, since the specific heat capacity varies for different fluids and different temperature levels. Therefore, the types of the fluids are related to both required and resulting specific superheats.

Using Maxwell equation:

$$Tds = dh - vdp \quad (3)$$

Where h is specific enthalpy (kJ/kg) and P is pressure (kPa). For constant pressure, the second term will be eliminated and the specific superheat will be only a function of entropy difference and the temperature level at which superheat is added or expected. Therefore, in case of same entropy difference between superheated and saturated vapor for a PS and an NS fluid, due to the lower temperature level in PS fluid (Fig. 3a), resulting specific superheat could be less than of required specific superheat for an NS fluid (Fig. 3b). However, when comparing two fluids under similar heat source conditions, different mass flow rate of the fluids would affect the superheat load values. Thus, it is not always enough to compare fluids solely based on their types and external parameters and limitations should be also taken into account.

4.2 The Whole System

Unlike the previous approach, effect of heat source utilization, pinch point location in the evaporator and fluid mass flow rate have been taken into account and the whole system is investigated. One application of this analysis could be trade-off between expander isentropic efficiency and superheat load in NS fluids. Assuming the same working pressure levels for a specific NS fluid for a given heat source, by having an expander with lower isentropic efficiency, both net power and required superheat load decrease. The required superheat is decreased until the total superheat load in evaporator and condenser is at its minimum value. By further decreasing isentropic efficiency below this point, resulting superheat load would grow faster than the decrease in required superheat load, causing increase in total superheat load. As an example, for propane with a heat source of 200 °C and a cycle operating between 25°C and 86°C as condensing and evaporating temperatures, different values of expander isentropic efficiency are analyzed and the results are displayed in Fig. 5. Around isentropic efficiency of 0.7, total superheat load is minimum and the decrease rate of required superheat becomes lower compared to higher isentropic efficiencies. Therefore, for an NS fluid, there is an isentropic efficiency which results in lowest possible superheat load.

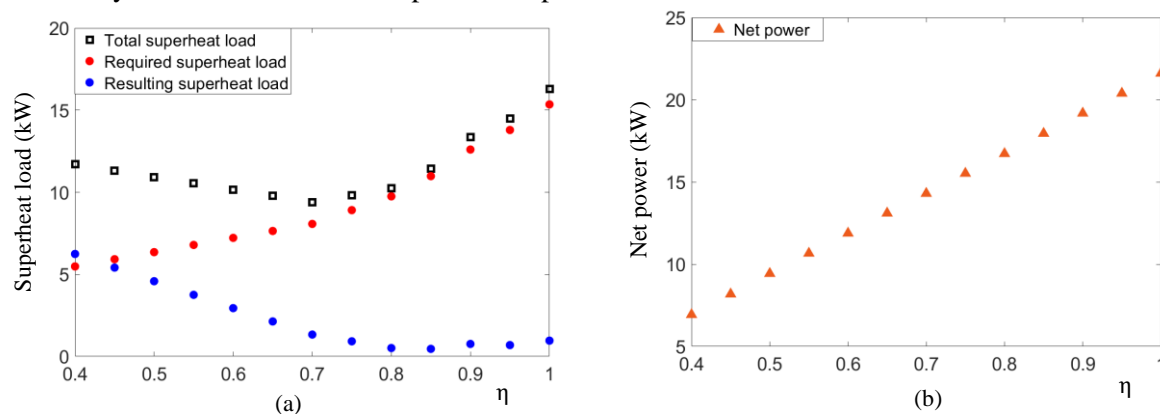


Figure 5: Effect of expander isentropic efficiency on: (a) superheat load, (b) net power for propane

For NS fluids, if the evaporating temperature is close to the critical region where the slope of the T-s diagram approaches to zero, required superheat load would be always necessary to avoid wet region. But for lower evaporating pressures, there is a possibility to reduce required superheat load close to zero, with specific expander isentropic efficiency. For PS and isentropic fluids, required superheat load would be present if the evaporator temperature is above point M (local maximum in T-s diagram). By choosing a lower temperature, required superheat load would be zero. Therefore, the evaporating temperature level plays a key role in minimizing required superheat load.

Evaporating temperature also affects the net power. The optimum evaporating temperature level for maximum net power optimizations, depends on the critical temperature of the fluid, heat source inlet temperature and pinch point temperature difference in the evaporator. Fig.6 displays net power vs. evaporating temperature for different heat source temperature levels, for neopentane with critical temperature of 160.6°C. For lower heat source temperatures, there is an optimum evaporating temperature at which the net power is maximized. However, for higher heat source values, net power

would increase by increasing the evaporator temperature, but the slope decreases gradually. Therefore, in the second case, it is important to stay away from the region close to the critical temperature to avoid or decrease inlet superheat. This behavior is valid for all the fluids regardless of their types.

As displayed in Fig. 7, the ratio of superheat load to total heat load in the evaporator and condenser is plotted for various fluid types and three cases with expander isentropic efficiency of 0.5, 0.7 and 0.9 respectively. The optimizations are based on maximum net power for a heat source temperature of 200°C. For PS and isentropic fluids, by lowering isentropic efficiency, the above mentioned ratio decreases. However, for NS fluids, it depends on the isentropic efficiency at which there exist a minimum superheat load between that specific evaporating and condensing temperatures.

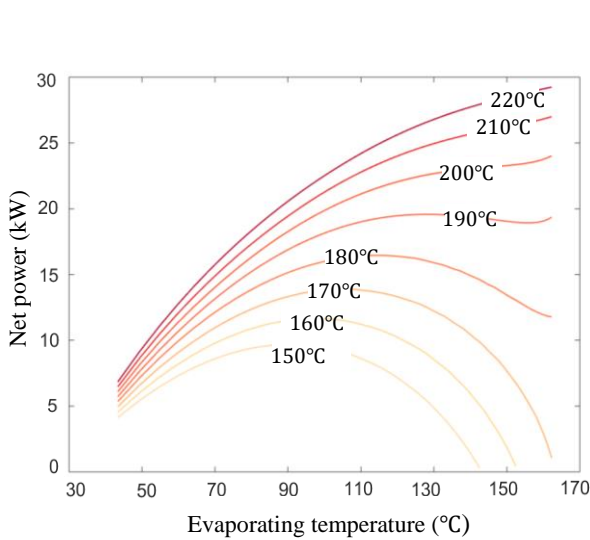


Figure 6: Net power vs. evaporator temperature at different effective heat source temperatures for neopentane

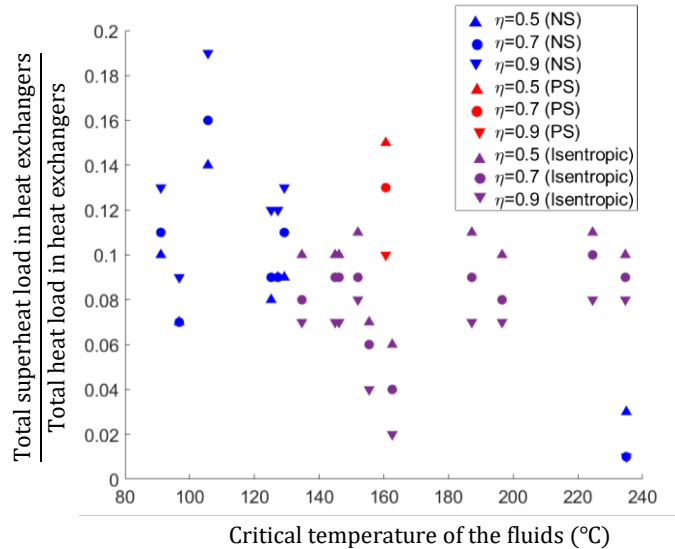


Figure 7: Ratio of total superheat load to total heat load in evaporator and condenser for various fluids at different expander isentropic efficiency

As an example, net power optimization results for a heat source temperature of 250°C with expander isentropic efficiency of 0.7 are presented in Fig. 8. Net power, required and resulting superheat loads are plotted for each fluid.

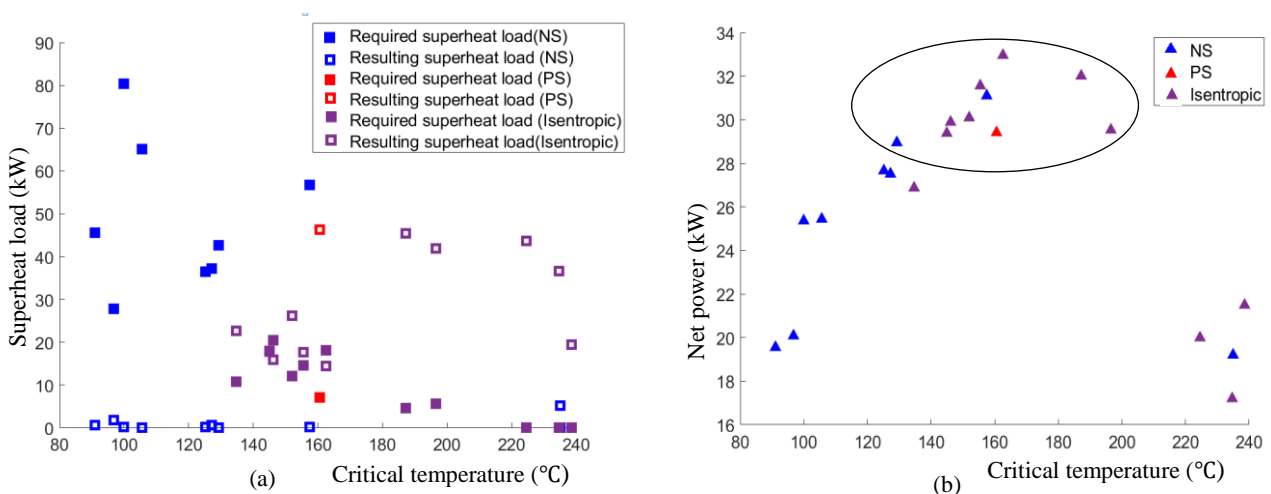


Figure 8: Results of maximum net power optimizations for different fluids: (a) required and resulting superheat load, (b) net power

As shown in Fig. 8b, fluids having high net power could be of any type of NS, PS or isentropic. By comparing the results from Fig. 8a, it is clear that required superheat load for NS fluids is higher compared to other types and there is almost zero resulting superheat. For PS and isentropic fluids, resulting superheat load is always present with required superheat load for some fluids. As it is clear, fluid type is important and optimizations should not be solely based on maximum net power. However, after having the optimum operating conditions, it is possible to further optimize each type of the fluid. For NS fluids, there is a possibility of decreasing required superheat load by adjusting the expander isentropic efficiency. For PS and isentropic fluids, required superheat load could be deleted by choosing evaporator temperature below characteristic point M.

5. CONCLUSION

The fluid type has always been discussed as an important criterion in fluid screening for RC applications. However, the results indicate that more factors need to be taken into account. The ratio of required or resulting superheat load to work is independent of fluid mass flow rate and therefore it could be linked to the type of the fluid and the steepness of the saturated vapor curve. However, the total net power could be different. Therefore, this paper suggests that the fluids should not be eliminated from the candidate list just due to their types. In cases of NS fluids, the presence of the expander isentropic efficiency could help to reduce the required superheat at the expander inlet. In this case, by using an alternative expander with lower isentropic efficiency, the superheat load on the evaporator decreases as well as the expander work. However the ratio of superheat to expander work decreases. Therefore, there could be a trade-off between the isentropic efficiency of the expander and the total superheat load. Hence, a comparison should not be made considering only ideal cases where the expansion process is isentropic. There are some characteristics of fluids which could be related to their type. However, comparison between the performances of various fluids in a cycle with all external parameters cannot be predicted just by their types. Therefore, knowing the type of the fluid helps in our approach in dealing with further optimizations for a specific fluid, but it does not necessarily help us to decide which fluid to choose.

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