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Thermodynamic influences of lubricant in an ORC for waste heat recovery in propulsion systems

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Abstract

Efficiency enhancement of various propulsion systems is in the focus of technical development, with the aim of reducing both operating costs and emissions. When considering internal combustion engines, only a fraction of the supplied fuel energy can be converted to mechanical work, whereas the remaining energy is mostly being released to the environment as waste heat. Organic Rankine Cycles (ORC) enable the use of this very waste heat for increasing overall efficiency of the propulsion system. To use the full potential of the waste heat recovery system, the impacts of the single influencing factors within the ORC have to be understood, such as the working medium, process management when using a regenerator, the number and arrangement of the single heat sources, the lower temperature level of the cycle (condensation temperature) or the share of circulating lubricant within the working cycle. This paper wants to assess the thermodynamic influence of the lubricant in an ORC in order to show the maximum potential of efficiency.

Keywords: WHR system; ORC; lubricant; efficiency.

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Nomenclature

b	constant of specific heat capacity of the lubricant [kJ/(kgK)]
c_{oil}	specific heat capacity of the lubricant [kJ/(kgK)]
h_i	specific enthalpy at the thermodynamic state i [kJ/kg]
m	linear coefficient of specific heat capacity of the lubricant [kJ/(kgK ²)]
$\dot{m}_{WF}, \dot{m}_{oil}$	mass flow of working medium / lubricant
p, p_L, p_H	pressure [bar], condensation pressure [bar], high pressure [bar]
q, q'	specific heat, heat related to working medium mass [kJ/kg]
q'_{oil-I}, q'_{oil-II}	heat of lubricant related to working medium mass: supplied / released via working medium [kJ/kg]
Q_{in}, Q_{out}	supplied / released heat [kJ]
$\dot{Q}_{in-oil}, \dot{Q}_{out-oil}$	supplied / released heat flow of lubricant [kW]
$\dot{Q}_{in-WF}, \dot{Q}_{out-WF}$	supplied / released heat flow of working medium [kW]
s, s_i	specific entropy, specific entropy at the thermodynamic state i [kJ/(kgK)]
T, T_i	temperature, temperature at the thermodynamic state i [K]
T_{in}, T_{out}	temperature of heat supply / release [K]
v	specific volume [m ³ /kg]
\dot{V}_{oil}	volumetric flow of lubricant [m ³ /s]
w'_{oil}	work related to working medium mass [kJ/kg]
Δ	difference of two values
μ_{oil}	mass fraction of lubricant [-]
$\eta_{ORC}, \eta_{oORC}, \eta_{oil}$	thermodynamic efficiency of the ORC / the oORC / the lubricant [-]
$\eta_{s-i,E}, \eta_{s-i,E_{max}}$	inner isentropic efficiency, maximum inner isentropic efficiency in oORC [-]
η_{th}, η_C	thermodynamic / Carnot efficiency [-]

1. Introduction

The use of Organic Rankine Cycles (ORC) for recovering mechanical work from (waste) heat ranges from solar thermal and geothermal energy to industrial processes and waste heat recovery of internal combustion engines in stationary as well as mobile applications (Zhiwei, 2016). A distinction can be made between Rankine Cycles (RC) and ORC, whereat the first, often used in classic power plants, normally use water as a working medium. ORC, in contrast, run on organic fluids with lower boiling temperatures and different gradients of the saturation vapour curve, providing favourable characteristics when using low temperature heat. Process management can be similar to the one of the RC (Brüggemann, 2011).

The expansion engines represent one of the key components regarding the cycle efficiency; they can be categorised as machines using closed working spaces, and those without. Especially when treating with comparatively small mass flows, transient operation or operation outside the design point, machines using closed working spaces can shine (due to the influence of leakage, the possibility of using partially wet steam, etc.), whereas bigger power plants benefit from the high operating time and the high efficiency at stationary operation at the design point of steam turbines (Muhammad Imran, 2015).

The different engine types require diverse boundary conditions concerning lubrication, connection of the power output shaft or starting procedure. In particular for machines using closed working spaces (reciprocating piston expanders, axial piston expanders, vane cell expanders, screw-type or scroll expanders), a considerable effort is needed for the sealing of the moving part containing steam to the regions containing lubricant. Alternatively, permitting some inevitable mixing of the lubricant with the working medium is an option, with a downstream separation of the media (Lang, 2016).

Another approach is the deliberate and permanent mixing of the media, using specific high temperature oil being added at a certain portion to working medium. This lubrication concept does not only contribute to the simplicity of the system, but also significantly influences the thermodynamic characteristics of the system as the lubricant stays in its liquid phase while circulating in the loop. Thus, a relevant amount of heat is being transported contributing to the work release only to a limited extent. The influence of the lubricant added to the working fluid with regard to the efficiency decrease is to be investigated below.

2. Organic Rankine Cycle (ORC)

An ORC uses organic working medium circulating in a closed cycle, undergoing cyclic phase changes.

Fig. 1 shows a simple ORC (left side) with its main components, on the right side the corresponding temperature-entropy Ts diagram for idealised process operation can be seen. Starting from state 0, an isentropic pressure increase to 1s happens within the pump; 1s is part of the low temperature/high pressure domain of the cycle where the working medium is liquid. The evaporator performs an isobaric heat supply with the fluid being preheated, evaporated and superheated; the thermodynamic state 2 lies within the high temperature/high pressure domain with gaseous medium (note: for the described cycle a superheated process is assumed). Within the expansion engine the thermal energy of the steam is converted to mechanical energy via expansion, with isentropic change of state to 3s being assumed. At the end of the expansion, the state of the working medium is at waste steam temperature in the low pressure domain. Within the condenser, heat is being released isobarically to point 0 again.

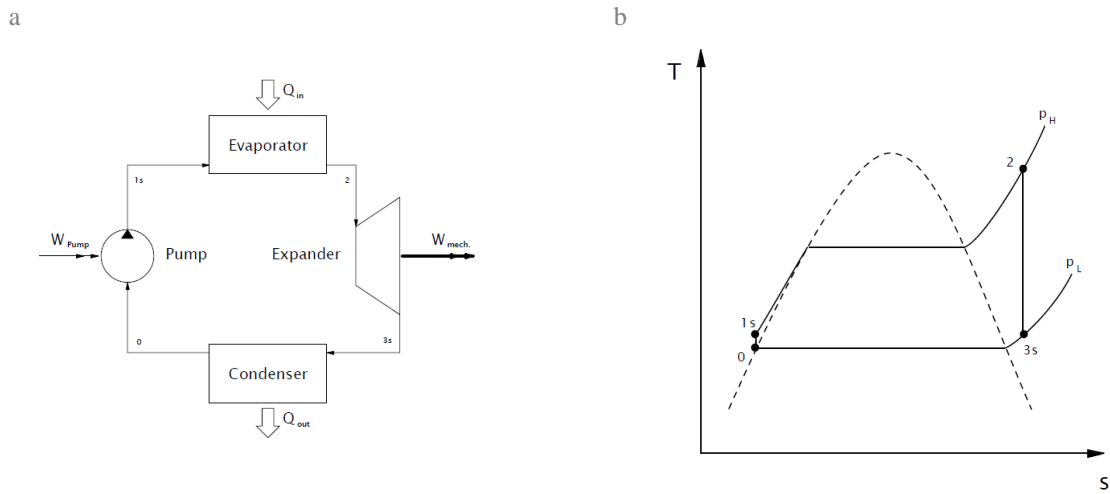


Fig. 1 (a) Scheme of a simple ORC, (b) Ts diagram of a simple ORC

2.1. Efficiency in thermodynamic cycles and in the idealised ORC

The work released in (frictionless) thermodynamic cycles results as the enclosed area between the supplied and the released heat, the thermodynamic efficiency η_{th} can be calculated as the quotient of use and expense, and thus according to equation (1) (Eichlseder, 2005).

$$\eta_{th} = \frac{Q_{in} - Q_{out}}{Q_{in}} \quad (1)$$

The upper limit of the thermal efficiency for the process of converting heat to mechanical work is defined in the Carnot efficiency η_C ; therein only the temperatures of heat supply and heat release are relevant (Eichlseder, 2005).

$$\eta_C = 1 - \frac{T_{out}}{T_{in}} \quad (2)$$

Simple ORC acc. to fig. 1 cannot reach the Carnot efficiency as heat supply cannot happen at one constant temperature. The upper limit of efficiency of such ORC, η_{ORC} , is calculated in equation (3) (Eichlseder, 2005) (Pischinger, 2009) (Vetter, 2014).

$$\eta_{ORC} = \frac{(h_2 - h_{1s}) - (h_{3s} - h_0)}{h_2 - h_{1s}} \quad (3)$$

3. Organic Rankine Cycle-Oil (oORC)

The indication oORC describes a cycle that mainly resembles the outlined idealized ORC, with the one exception of a certain fraction of lubricant being added to the working medium, circulating in the system. The definition of the upper limit of efficiency for this cycle uses the idealized assumptions of chapter 2 (no friction losses, no pressure losses, no leakage losses) as well as the following simplifications and assumptions.

3.1. Assumptions for the lubricant

The defined proportion of lubricant is the same for each point within the cycle. No decomposition whatsoever occurs, as well as no change of lubricant proportion for different load points. The lubricant stays in its liquid phase.

The lubricant is considered incompressible. The expended pump work is released to the same extent during expansion in the engine.

$$\int_0^{1s} \dot{V}_{oil} dp = - \int_2^{3s} \dot{V}_{oil} dp \quad (4)$$

The lubricant is in thermodynamic balance with the working medium throughout the whole cycle. Heat exchange with the working medium occurs corresponding to the specific heat capacity of the lubricant. The latter is taken into account as a linear function of the temperature (equation (5)), which describes reality with good approximation (T.M. Medved, 1959).

$$c_{oil} = m \cdot T + b \quad (5)$$

Heat supply to the lubricant happens from state 0 to state 2.

Heat release occurs, depending on process management, only via the working fluid (when expanding into the wet steam region) or additionally via the sensible heat out of the lubricant (when expanding only within the superheated steam region) (compare fig. 3).

3.2. Assumptions about the expansion

For the expansion process, thermodynamic equilibrium between the working medium and the lubricant is assumed at all times. For this, two systems ("working fluid" and "oil") with mutual heat exchange are considered (fig. 2 (a)). The expansion process is composed step by step according to the sketch of the Ts diagram in fig. 2 (b). The assumptions are also applicable to compression.

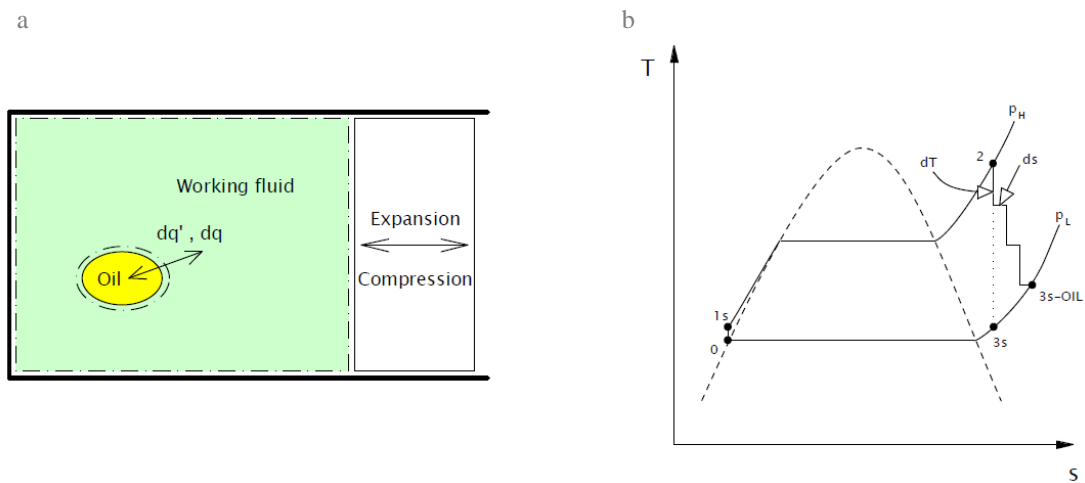


Fig. 2 (a) model sketch, (b) Ts diagram to the oORC model for stepwise expansion

For the system “working fluid”, in a specific notation (i.e. referring to 1kg working fluid), there can be written:

$$dq - pdv = du$$

$$dq = du + pdv = T \cdot ds$$

For the system “oil”, in a specific notation (referring to 1kg oil), there can be written:

$$-dq = du = c_{oil} \cdot dT$$

and in an absolute notation:

$$-d\dot{Q} = c_{oil} \cdot \dot{m}_{oil} \cdot dT$$

For the system “oil”, referring to 1kg working fluid, there can be written:

$$-dq' = c_{oil} \cdot \frac{\dot{m}_{oil}}{\dot{m}_{WF}} \cdot dT$$

and with the mass flow rate fraction according to equation (6)

$$\mu_{oil} = \frac{\dot{m}_{oil}}{\dot{m}_{WF} + \dot{m}_{oil}} \quad (6)$$

hence

$$-dq' = c_{oil} \cdot \frac{\mu_{oil}}{1 - \mu_{oil}} \cdot dT$$

The energy balance with $-dq' + dq = 0$ yields:

$$-T \cdot ds = c_{oil} \cdot \frac{\mu_{oil}}{1 - \mu_{oil}} \cdot dT$$

The equation (5) of the specific entropy of the working fluid as a function of the temperature, starting with point 2, follows from equation (5).

$$s(T) = s_2 + \frac{\mu_{oil}}{1 - \mu_{oil}} \left[m \cdot (T_2 - T) + b \cdot \ln \frac{T_2}{T} \right] \quad (7)$$

From equation (7), the temperature $T_{3s-OIL} = f(p_L, s(T))$ can be determined iteratively with a predefined condensation pressure via a fluid properties database. This determines the course and the point 3s-OIL, the efficiency calculation can thus be carried out.

3.3. Efficiency in an oORC

The heat flow supply via the working fluid is given by:

$$\dot{Q}_{in-WF} = \dot{m}_{WF} \cdot (h_2 - h_{1s})$$

The heat flow supply via the lubricant is given by:

$$\dot{Q}_{in-oil} = \dot{m}_{oil} \cdot \left[m \cdot \left(\frac{T_2^2}{2} - \frac{T_0^2}{2} \right) + b \cdot (T_2 - T_0) \right]$$

The heat flow removal via the working fluid results from:

$$\dot{Q}_{out-WF} = \dot{m}_{WF} \cdot (h_{3s-OIL} - h_0)$$

The heat flow removal via the lubricant results from:

$$\dot{Q}_{out-oil} = \dot{m}_{oil} \cdot \left[m \cdot \left(\frac{T_{3s-oil}^2}{2} - \frac{T_0^2}{2} \right) + b \cdot (T_{3s-oil} - T_0) \right]$$

Out of this, the thermodynamic efficiency for the oORC can be written by:

$$\eta_{oORC} = 1 - \frac{\dot{Q}_{out-WF} + \dot{Q}_{out-oil}}{\dot{Q}_{in-WF} + \dot{Q}_{in-oil}} \quad (8)$$

and further the decrease of the thermodynamic efficiency to the ORC with:

$$\Delta\eta_{th} = \eta_{ORC} - \eta_{oORC} \quad (9)$$

Fig. 3 (a) shows the Ts diagram with the superheated point 3 s-Oil, where a temperature difference T Diff-2 can be seen which results in a heat removal from the lubricant portion. Fig. 3 (b) shows the Ts diagram with the point 3 s-oil in the wet steam region; here no heat removal takes place via the lubricant portion because the temperature difference T Diff-2 is zero.

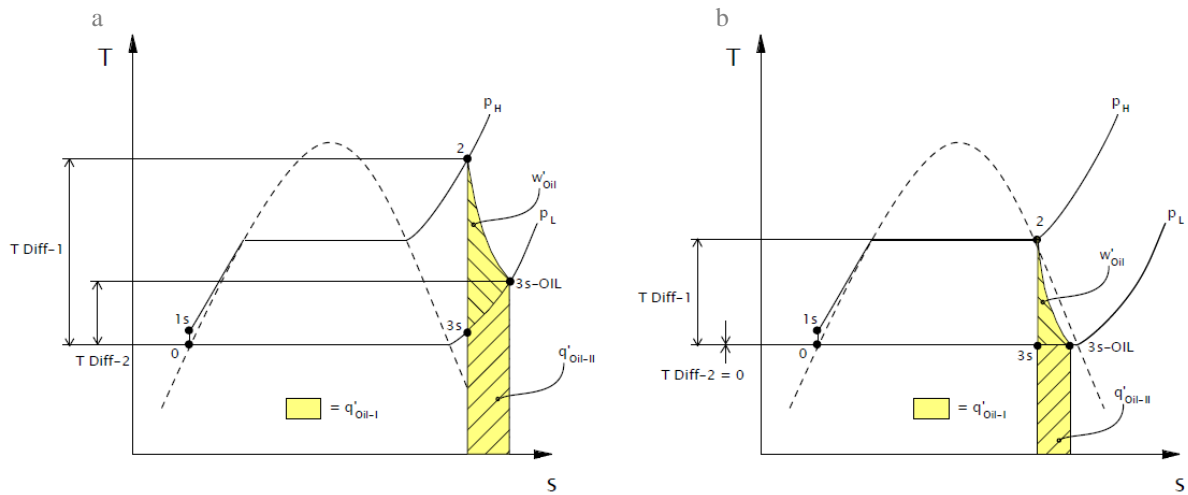


Fig. 3 (a) point 3s-OIL in the superheated region, (b) point 3s-OIL in the wet steam region

The heat supply and the heat rejection from the lubricant to the working medium are represented by the depicted surfaces in fig. 3:

$$q'_{oil-I} = \frac{\mu_{oil}}{1 - \mu_{oil}} \cdot \left[m \cdot \left(\frac{T_2^2}{2} - \frac{T_{3s-oil}^2}{2} \right) + b \cdot (T_2 - T_{3s-oil}) \right]$$

$$q'_{oil-II} = h_{3s-oil} - h_{3s}$$

From this follows the specific work that the working fluid can perform by the heat from the lubricant:

$$w'_{oil} = q'_{oil-I} - q'_{oil-II}$$

and from this the efficiency of the lubricant fraction according to equation (10):

$$\eta_{oil} = \frac{\dot{m}_{WF} \cdot w'_{oil}}{\dot{Q}_{in-oil}} \quad (10)$$

The inner isentropic efficiency $\eta_{s-i,E}$ is used to evaluate expansion machines. Therefore, the specific enthalpy difference between the input and output at the expander is related to the isentropic enthalpy difference. In the case of a lubricant fraction, a maximum isentropic efficiency cannot be exceeded given by point 3is-OIL (equation (11)).

$$\eta_{s-i,E} = \frac{h_3 - h_2}{h_{3s} - h_2} \leq \frac{h_{3s-OIL} - h_2}{h_{3s} - h_2} = \eta_{s-i,E_{max}} \quad (11)$$

3.4. Variables influencing the efficiency

The following points were taken into account in the calculation as influencing variables on the efficiency of the oORC:

- Type of working fluid
- Type of lubricant
- Proportion of lubricant, pressure and superheating temperature

3.4.1. Type of working fluid

To illustrate the effects associated with the working fluid, Ethanol was used as a wet-relaxing working fluid and Cyclopentane as a dry-relaxing working fluid. In fig. 4 the Ts diagrams of the two fluids are depicted, the characteristic points being marked where the Oil A (compare 3.4.2) with an oil fraction of 15m% is considered. In both cases, the same condensation temperature of about 90°C was applied, whereupon a condensation pressure of about 1.55bar for Ethanol and about 3.25bar for Cyclopentane was obtained. The live steam parameters were selected in each case with 20bar and 10K superheating temperature.

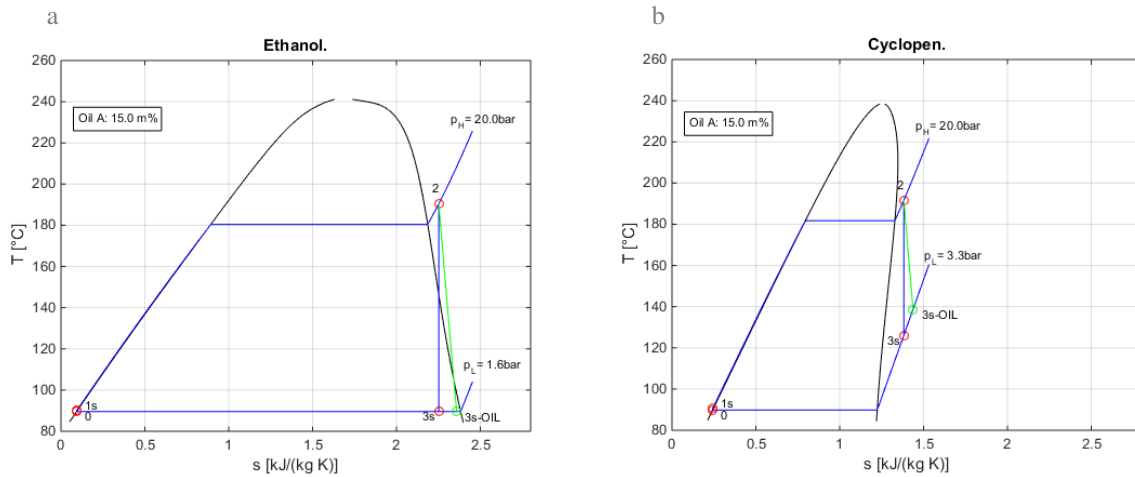


Fig. 4 (a) Ts diagram Ethanol, (b) Ts diagram Cyclopentane

3.4.2. Type of lubricant

The specific heat capacity of the applied lubricant has an effect on the transferred heat. In order to show this effect, two different oils were considered. Oil A is a mineral oil-based oil and Oil B is a synthetic one, with about 21% lower specific heat capacity at 400K. The dependency on the temperature is shown in fig. 5, the synthetic lubricant has a temperature-dependent gradient in the specific heat capacity which is about 50% lower. Fluid property data of the oils were used according to (Wagner, 2013).

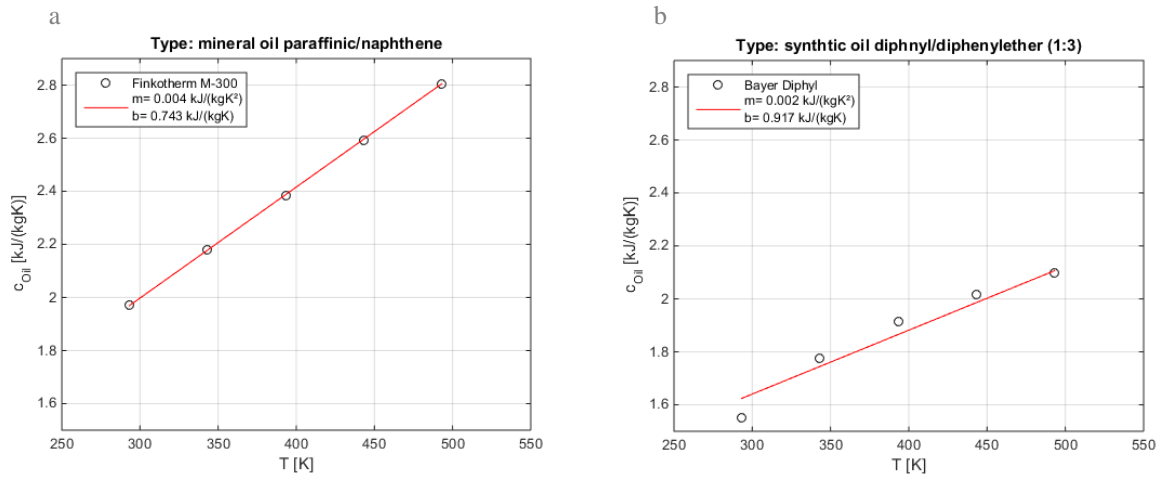


Fig. 5 (a) Oil A mineral, (b) Oil B synthetic

Table 1 shows the influence on the efficiency of the working fluids and the two lubricants in the described operating point.

Table 1. Influence of the working fluids and the lubricants on the efficiency at the boundary conditions: 15m% lubricant content, 20bar, 10K superheating, 90°C condensation temperature

Working fluid / lubricant	η_{ORC} (%)	η_{oORC} (%)	$\Delta\eta_{th}/\eta_{ORC}$ (%)	η_{Oil} (%)	$\eta_{s-i,Emax}$ (%)
Ethanol / Oil A	17,05	16,81	1,40	12,07	76,34
Cyclopentane / Oil A	15,64	14,63	6,43	4,03	72,03
Ethanol / Oil B	17,05	16,91	0,85	11,99	81,66
Cyclopentane / Oil B	15,64	14,86	4,94	4,17	77,58

The limiting efficiency η_{ORC} is about 8.8% higher for Ethanol at these boundary conditions. The advantage of Cyclopentane shows up at lower condensation temperatures when the condensation pressure for Ethanol is below the atmospheric pressure and the condensation pressure represents the limit.

Cyclopentane also shows a greater reduction in the limiting potential $\Delta\eta_{th}$ relative to η_{ORC} (6.4% for Oil A and 4.9% for Oil B). This can be attributed, on the one hand, to the proportionally higher value of the heat transported with the lubricant. The lower specific heat capacity of the synthetic oil B is found to be favourable. On the other hand, the property of a dry-relaxing fluid leads to poorer utilization of the heat from the lubricant (cf. η_{Oil}), because the expansion end is shifted further and further into the superheated region. The maximum achievable isentropic efficiency is related to the proportionally transported heat of the lubricant.

3.4.3. Proportion of lubricant, pressure and superheat temperature

The influencing factors of the lubricant content, the pressure and the superheating temperature also cause changes in the heat transport via the lubricant and the utilization of the heat from the lubricant. Both factors interact, these properties are shown in fig. 6 and fig. 7.

At higher pressures, a weaker effect on the change of $\Delta\eta_{th}/\eta_{ORC}$ is shown. At low pressures and a higher proportion of lubricant, the gradient is greater. For wet-relaxing fluids, a minimum is found in the $\Delta\eta_{th}/\eta_{ORC}$ corresponding to a certain superheating temperature. This minimum occurs at the superheating temperature which gives the point 3s-Oil close to the saturated steam line in the superheated range. In the case of dry-relaxing fluids, this behaviour is generally not given, because an increasing overheating results in efficiency degradation.

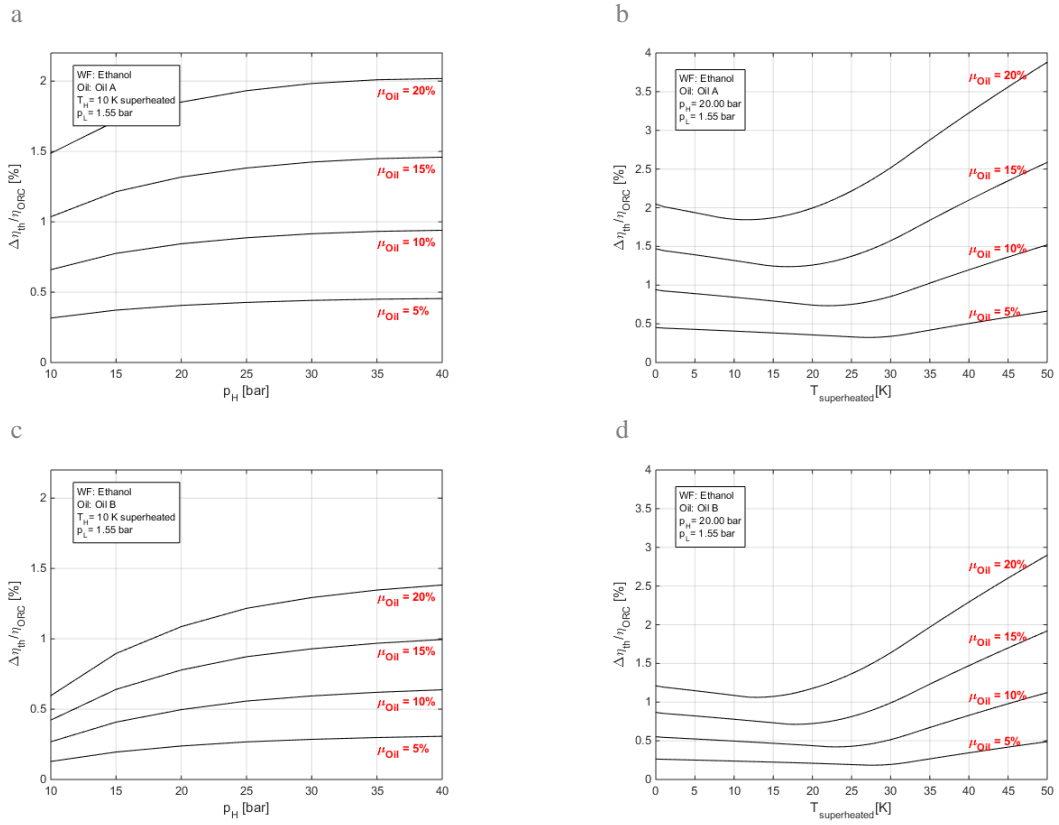


Fig. 6 Influence of lubricant content over pressure and superheat temperature: working fluid Ethanol, (a, b) Oil A, (c, d) Oil B

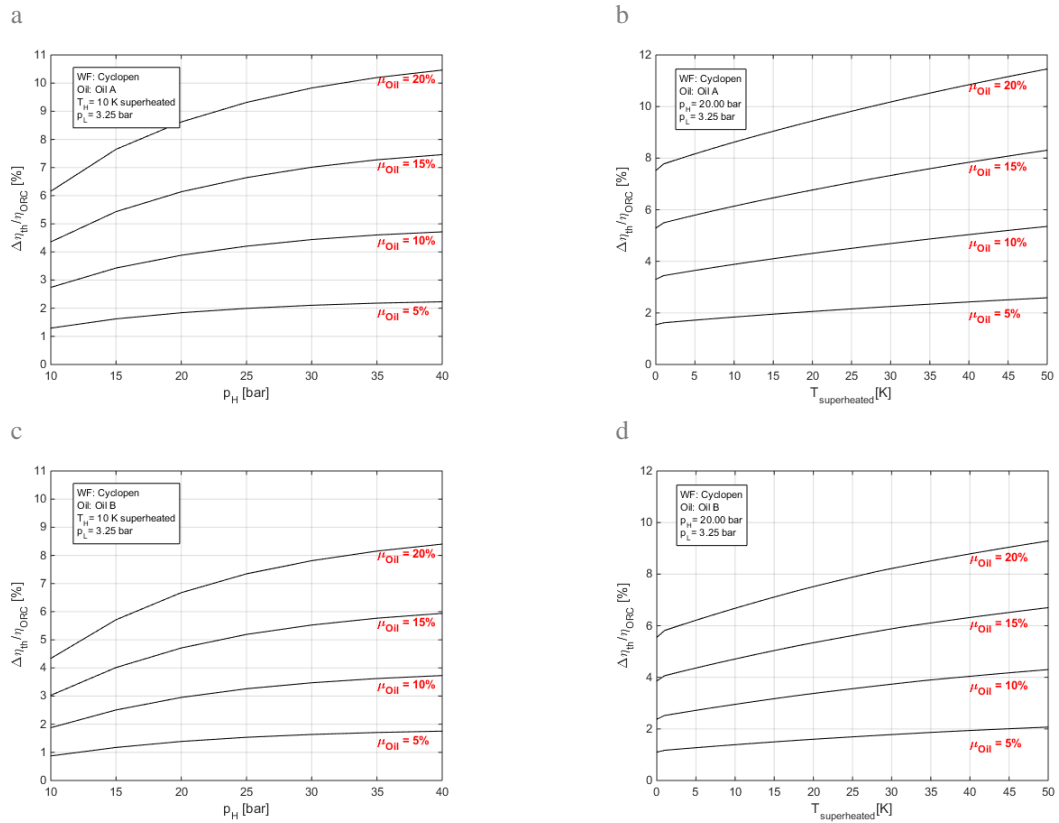


Fig. 7 Influence of lubricant content over pressure and superheating temperature: working fluid Cyclopentane, (a, b) oil A, (c, d) oil B

4. Summary and outlook

Using the methodology shown, the limit potential of an oORC can be determined. The influencing factors and their magnitude were shown here under the assumptions made. The type of working medium and lubricant as well as the lubricant content must be considered as the main influencing variables. The ratio of the absorbed heat of the lubricant to that of the working medium should be as low as possible in order to keep the influence of the oil content low, with a heat ratio of less than 2%, the efficiency reduction is less than 1%. Furthermore, in the case of wet-relaxing fluids, a specific optimum is obtained with regard to the superheating temperature, so that the efficiency reduction with respect to the ORC efficiency becomes minimal. In general, the reduction of efficiency at low pressures is smaller, but the low pressure also causes a lowering of the base, namely the ORC efficiency per se. In the real oORC, further effects occur due to the lubricant, which are very difficult or virtually impossible to determine via simulations, including:

- Influence of the lubricant on the heat transfer in the evaporator and the condenser (reduction of the heat transfer between the heat exchanger walls and the working fluid by film formation)
- Effect of non-occurring thermodynamic equilibrium (heat transfer between working fluid and lubricant, at different phases)
- Influence of the lubricant in the expansion machine, especially concerning gap sealing effects.

The actual effects of the leakage influence and the heat transfer are the subject of current test bench investigations. The determination of the real effects is relatively complex; the cycle has to be cleaned for each changed composition. The number of parameters and levels results in a correspondingly large number of variants. Furthermore, the measurements are associated with individual errors of different sizes, whereby the determination of the circulating portion of the lubricant represents the greatest uncertainty.

The current measurement results under the fluctuation range tend to show a correspondence with the described ideal processes. When looking at the circuit, the losses in the expansion engine (depending on type and operating point) were determined between 28%-76%. If the lubricant is chosen appropriately in connection with the working medium, the losses are subordinate to those from the expansion engine (Ethanol and Oil B). For the composition: Cyclopentane and Oil A, the losses due to the lubricant can take a not insignificant share.

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