IRREVERSIBILITY IN THE ORGANIC RANKINE CYCLE FOR LOW-GRADE THERMAL ENERGY CONVERSION SYSTEM

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ABSTRACT

Organic Rankine cycles (ORCs) are used in power generation applications for low-temperature heat sources such as waste heat recovery from plants, geothermal binary, hot springs, and ocean thermals. Meanwhile, ORC equivalent technologies are applied in ocean thermal energy conversion (OTEC), which uses the temperature difference between the surface and the depths. ORCs with such low-grade thermals must consider various forms of irreversibility, not just the performance of the turbine/generator and heat exchangers. Typical ORCs are mainly composed of a turbine/generator, a working fluid circulation pump, and heat exchangers. A new performance evaluation concept for low-grade thermal energy conversion (LTEC) is proposed that normalizes the thermal efficiency considering heat leak and equilibrium state as a dead state based on finite-time thermodynamics. In addition, considering the trade-off between the heat transfer performance and the pressure drop due to the flow, a unique performance evaluation method based on the maximization of the net obtainable energy per heat transfer is initiated on a plate-type heat exchanger in LTEC. Here, to keep the cost of systems down, ease of examination, and proper design of the equipment, standardization and calculation tools are useful. In this research, theoretical maximization of the available power employing the irreversibility of LTEC system is conducted employing the parameter analysis of simple Rankine cycle to arise the coefficient of irreversibility. In the results, the coefficient of irreversibility is clarified on working fluids (HCFC245fa and HCFO1224yd(z)), turbine efficiency, and working fluid circulation pump efficiency, respectively.

1. INTRODUCTION

Low-grade thermal energy including waste heat has abundant potential and can contribute to the achievement of sustainable development goals (SDGs). Indeed, low-grade thermal energy sources <150°C, so-called "low enthalpy energy," have the estimated potential to produce >1,400 TWh/year from low-temperature geothermal (Chandrassekharam and Bundschuh, 2016) and 10,000 TWh/year from ocean thermal energy (IEA-OES, 2007) around the world. The production of electric power from low-grade thermal energy can be applied in a heat engine that utilizes working fluid in closed cycles. The cycles use a low-boiling-point working fluid of ammonia, an ammonia/water mixture, or organic materials to avoid the high vacuumed condition in the condenser in a case where water is the working fluid. The utilization of organic materials called the organic Rankine cycles (ORCs) has long been applied in biomass heat recovery, waste heat, and low-enthalpy geothermal systems (Tartiere and Astolfi, 2017).

The ORCs experimental data trends have been summarized by Park et al. (2016), showing typical systems and working fluids. To evaluate the ORC systems, exergy analysis is conducted that shows the exergy losses in each component in the system (Abam et al., 2018). Zhu et al. (2014) introduced the concept of entransy that goes beyond conventional energy and exergic parameters to evaluate energy conversion systems. Li et al. (2017) proposed an evaluation method for ORC based on the trapezoidal model to evaluate the cycles' irreversibility. It proposed a performance evaluation method for low-grade thermal energy conversion (LTEC) to normalize the thermal efficiency considering heat leak to the environment and defined the equilibrium state as the dead state based on the concept of finite-time thermodynamics (FTT) (Yasunaga and Ikegami, 2017). In addition, considering the trade-off between the heat transfer performance and pressure drop due to the flow, the unique performance evaluation method based on the maximization of the net obtainable energy per heat transfer was initiated on plate-type heat exchangers in LTEC (Yasunaga et al., 2018). Although Bejan (1998) and many other researchers such as Ibrahim et al. (1991) and Ikegami and Bejan (1998) proposed the irreversibility of the energy conversion, the concept's application in engineering remains limited.

Since there are various organic working fluids, Saloux et al. (2018) and Stijepovic et al. (2017) proposed a thermodynamic model and proposed a new procedure to apply in ORC, respectively. In addition, Eyerer et al. (2019) experimentally compared performances to apply low global warming potential (GWP) working fluids for ORC systems and show the comparability of HCFO1244yd(z) and HCFO1233zd(e) as the widely used conventional working fluids of HCF245fa.

In contrast, for securing energy, people examine low-grade thermal energy applications that are low exergy heat sources. For the cost down of the systems and ease and adequate availability of the power generation, the standardization and calculation tools are useful. However, it is difficult to clarify the effect of the constituent equipment's performance on the irreversibility of the theory for both the system's total performance and to consider each component's performance because the power output is the balance of each item, which is complicated. Therefore, based on Finite-time Thermodynamics (FTT), this research formulates the theoretical maximization of the power considering the irreversibility of a heat engine and focuses on the irreversibility of typical components such as working fluids, turbines, working-fluid circulation pumps, and the impact of each component on the coefficient of irreversibility is investigated by comparing the theoretical formulas and parameter analysis of the Rankine cycle. The method is applied practically to show the comparability of the quite low GWP working fluid HCFO1244yz as an alternative to the conventional HCF245fa.

2. EQUATIONS AND EVALUATION METHOD

2.1 Maximum Work and Irreversibility

2.1.1 Concept of power generation system: When utilizing a finite-heat source temperature and flow rate such as low-grade thermal energy, the system's performance is the result of the balance of the constituent elements. By balancing each component's performance, the system can reach its theoretical maximum power condition. Therefore, the evaluation of the power generation system must consider the characteristics of the performance including the internal and external irreversibility to compare, develop, and improve the system. Figure 1(a) shows the concept of the power generation system using heat engines driven by low-grade thermal energy and the cooling system. The power generation device is an adiabatic system and then no heat leak happens. Figure 1(b) shows the schematic flow of the Rankine cycle. The system can generally utilize any type of heat from low enthalpy heat source flows such as exhaust gases, hot water in geothermal systems, or effluent from plants. However, this paper applies hot and cold water as typical heat sources of power generation systems.

2.1.2 Maximum work from low-grade thermal energy: In accordance with energy conservation, the available work from a heat engine W and the thermal energies Q_W, Q_C that are proportional to the temperature change in the heat source, are expressed:

$$W = Q_H - Q_L \tag{1}$$

$$Q_H = \left(mc_p\right)_H \left(T_H - T_{H,O}\right) = (UA\Delta T_m)_E \tag{2}$$

$$Q_L = \left(mc_p\right)_L \left(T_{L,O} - T_L\right) = (UA\Delta T_m)_C \tag{3}$$

where m is the mass flow rate, c_p is the specific heat of a heat source assumed constant in the system, T is the temperature, U is the overall heat transfer coefficient, ΔT_m is the logarithmic mean temperature difference, and the subscription of H is a high-temperature heat source, L is a low-temperature heat source, E is an evaporator, and C is a condenser.

Herein, the internal irreversibility coefficient ϕ is introduced in heat engines:

$$\frac{Q_H}{T_H} - \phi \frac{Q_L}{T_L} = 0,\ \frac{T_L}{T_H} \le \phi \le 1 \tag{4}$$

ϕ is the overall internal irreversibility that includes the turbine efficiency, working fluid circulation pump efficiency, pressure drop by piping and valves, and the characteristics of the working fluid's thermal properties.

From Eqs. (1)–(4), the work output will be 1 degree of freedom from the high or low source outlet temperature of the heat exchanger and the work output maxima W_m from the heat engine can then theoretically be derived as (Ibrahim et al., 1991):

$$W_m = \frac{\emptyset (mc_p)_L (mc_p)_H \varepsilon_C \varepsilon_E \Delta T_{HS,\phi}}{\emptyset (mc_p)_L \varepsilon_L + (mc_p)_H \varepsilon_H} \tag{5}$$

$$\varepsilon_E = 1 - e^{-NTU_E}\ ,\ \ \varepsilon_C = 1 - e^{-NTU_C} \tag{6}$$

$$NTU = \frac{UA}{mc_p} \tag{7}$$

$$\Delta T_{HS,\phi} = \left(\sqrt{T_H} - \sqrt{T_L/\phi}\right)^2 \tag{8}$$

where ε is the effectiveness of the heat transfer performance and *NTU* is the net transfer unit that shows the performance of heat exchangers that produce external irreversibility. Although the ε-*NTU* method is typically utilized in the design of heat exchangers with a single phase fluid, this is applied to the analysis of the power generation system to indicate the performance of heat exchangers with constraint (*UA*) (Bejan,1998, Ibrahim, 1991)). Under Eq. (5), if there are no limitations to using high- and low-temperature heat sources, W_m will be maximized in the case of $(mc_p)_L = (mc_p)_H$.

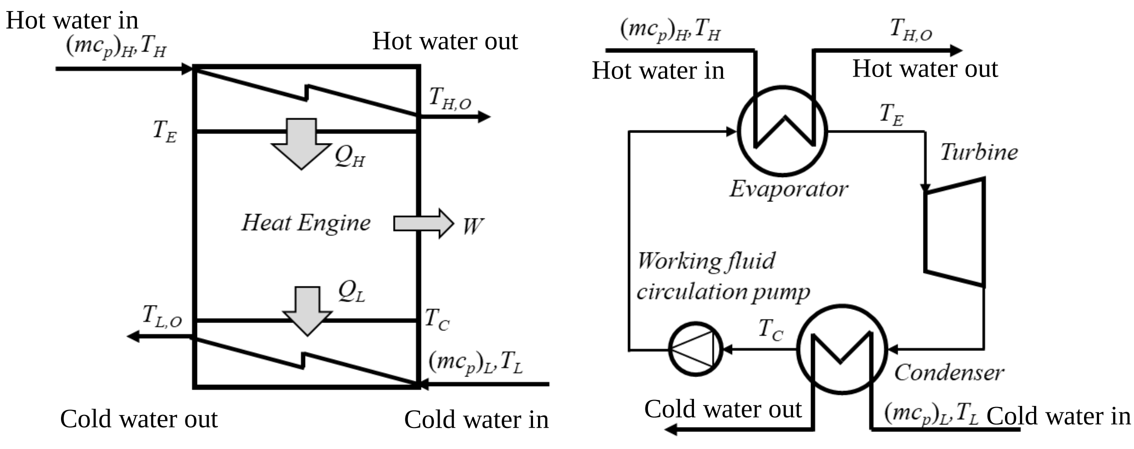


Figure 1: (a) the concept and (b) schematic flow of the LTEC power generation system.

2.1.3 Turbine and working fluid circulation pump efficiency: The efficiency of the turbine η_T and the working fluid circulation pump are defined as:

$$\eta_T = \frac{h_{T,I} - h_{T,O}}{h_{T,I} - h'_{T,O}} \tag{9}$$

$$\eta_{WFP} = \frac{h_{WFO,I} - h'_{WFP,O}}{h_{WFP,I} - h_{WFP,O}} \tag{10}$$

2.1.4 Heat source systems: The required power for the heat source system is defined as the backwork. The backwork ratio, which is the backwork over the turbine power, is particularly important in a low-grade thermal energy system. However, since the heat source units depend on the case of the project and—in this study—focus on the system's basic and unavoidable irreversibility, the backwork caused by the heat source system is neglected.

2.2 Evaluation Method

In low-grade thermal energy, the thermal efficiency η_{th} that is applied to show the efficiency of the energy conversion is expressed as $\eta_{th} = W/Q$. However, FTT shows the contradiction between the thermal efficiency maxima and maximum work (Novikov, 1958 and Curzon and Ahlborn, 1975). Yasunaga and Ikegami (2017) applied the idea of FTT to low-grade thermal energy and proposed the thermal energy of a heat source and normalized thermal efficiency of energy conversion. Considering the first law of thermodynamics Eq. (1), the proposed available energy of a high-temperature heat source Q_{HS} can be modified as follows:

$$Q_{HS} = C_{HS} r(1-r)(T_H - T_L) \tag{11}$$

where C_{HS} and r are the total heat capacity flow rate of a heat source and the ratio of a high-temperature heat source heat transfer rate, which are respectively expressed as:

$$C_{HS} = (mc_p)_L + (mc_p)_H \,,\ r = \frac{(mc_p)_H}{(mc_p)_L + (mc_p)_H} \tag{12}$$

Referring to FTT, the exergy of the OTEC, which is one of the lowest-grade thermal energies in nature, is implemented (Yasunaga, et al., 2018). In the case of the OTEC, the available flow rate and the balance of heat sources is identified flexibly because of the vast quantity of seawater in the ocean. However, in low-grade thermal energy such as geothermal, hot-springs, and discharged heat definitely have a finite flow rate. Then, the exergy is expressed as follows:

$$E_x = C_{HS} T_H \left[r + (1-r)\left(\frac{T_L}{T_H}\right) - \left(\frac{T_L}{T_H}\right)^{1-r} \right] \tag{13}$$

Therefore, the normalized thermal efficiency of the energy conversion η_{ec} and the exergy efficiency η_{ex} are defined as follows:

$$\eta_{ec} = \frac{W}{C_{HS} r(1-r)(T_H - T_L)} \tag{14}$$

$$\eta_{ex} = \frac{W}{C_{HS} T_H \left[r + (1-r)\left(\frac{T_L}{T_H}\right) - \left(\frac{T_L}{T_H}\right)^{1-r} \right]} \tag{15}$$

2.3 Parameter Analysis Condition

2.3.1 Analysis method: The conventional Rankine cycle (Figure 1 (b)) was applied to find out the basic characteristics of the irreversibility, focused on the effect of (1) the working fluids, (2) the turbine efficiency, and (3) the working fluid circulation pump efficiency in the system. In particular, the working fluids were selected for typical representative pieces considering advancement in the present; HCF245fa ($CH_3CH_2CHF_2$) is widely used in conventional LTEC systems, and low-GWP organic working fluids have been released and investigated recently. In this research, considering the sustainability and the lowest GWP in conventional ORC research (Eyerer et al., 2019), the newly developed working fluid HCFO1224yd(z) ($CF_3CF{=}CHCl$) was applied as a replacement for HCF245fa as the typical low-GWP working fluid in ORC. HCFO1224yd(z) has similar characteristics to the thermal properties of HCF245fa, but is harmless and has a low GWP (<1). The analysis based on heat and mass balance in each state point in the adiabatic system, the pressure drop in piping and the valves and levels of equipment are neglected.

2.3.2 Analysis condition: Parameter analysis is conducted by the condition shown in Table 1 and the turbine and working fluid circulation pump efficiencies are varied in the range 10–100%. The thermodynamic properties are calculated by REFPROP ver.10.0 (Lemmon et al., 2018).

Table 1: Parameter analysis condition

High-temperature heat source (Water)		**Low-temperature heat source (Water)**		**NTU for heat exchangers**
Temperature [K]	Flow rate [t/h]	Temperature [K]	Flow rate [t/h]	Evaporator/Condenser
343.15–365.15	50	283.15–303.15	50	1, 2, 3

3. RESULTS AND DISCUSSION

3.1 Maximum Available Power

The internal irreversibility is analyzed at the maximum power condition to divide the effect of each ingredient on the performance. Figure 2 shows the typical power output from the Rankine cycle as a function of the working fluid mass flow rate. The diagrams have black lines for HCF245fa and red lines for HCFO1224yd(z). Based on the theory in Section 2.1.2, the maximum power can be confirmed. Comparing the two working fluids, the power output, the normalized thermal efficiency of energy conversion, and the exergy are almost same at the peak point and the working fluid flow rate at the peak point of HCFO1224yd(z) is slightly bigger than that of HCF245fa.

To normalize the overall irreversibility coefficient, ϕ is modified as follows:

$$\emptyset^{*} = 1 - \frac{1-\phi}{1-\dfrac{T_L}{T_H}} \tag{16}$$

The system's ideal design or operational condition is the peak of the power output; however, the discrepancy of the expected performance between the design and the actual performance or the degradation of the performance for some reasons will change the best operational condition. To identify the internal irreversibility, the argument on the maximum available power is the most important since Eq. (5) shows the work output maxima. Therefore, in the analysis of internal irreversibility, a good understanding of the maximum available power is useful for engineers.

3.2 Internal Irreversibility

3.2.1 Working fluid thermal property: First, the effect of working fluid on the irreversibility coefficient ϕ_{wf} will be confirmed. The difference between the two fluid working conditions is determined in Figure 3 to illustrate the *T*–*s* and *P*–*h* diagrams showing a simple Rankine cycle without superheated vapor and subcooled liquid to simplify it for the analysis of their irreversibility. Although the actual system needed superheated vapor to prevent the turbine wet condition and subcooled liquid to avoid cavitation in the working fluid circulation pump. According to Figure 3, the working temperature of evaporation and condensation remains the same at the maximum power condition while the handling pressure differs. Figure 4 shows the dependency on the normalized internal irreversibility of the working fluid $\phi_{WF}{}^{*}$. According to Figure 5, the dependency of the internal irreversibility coefficient of a working fluid is roughly proportional to the heat source temperature ratio. Since both working fluids have characteristics of becoming dry at the turbine outlet even if the turbine could make an isenthalpic change, the bigger temperature difference between evaporation and condensation creates relatively small irreversibility in the dry condition.

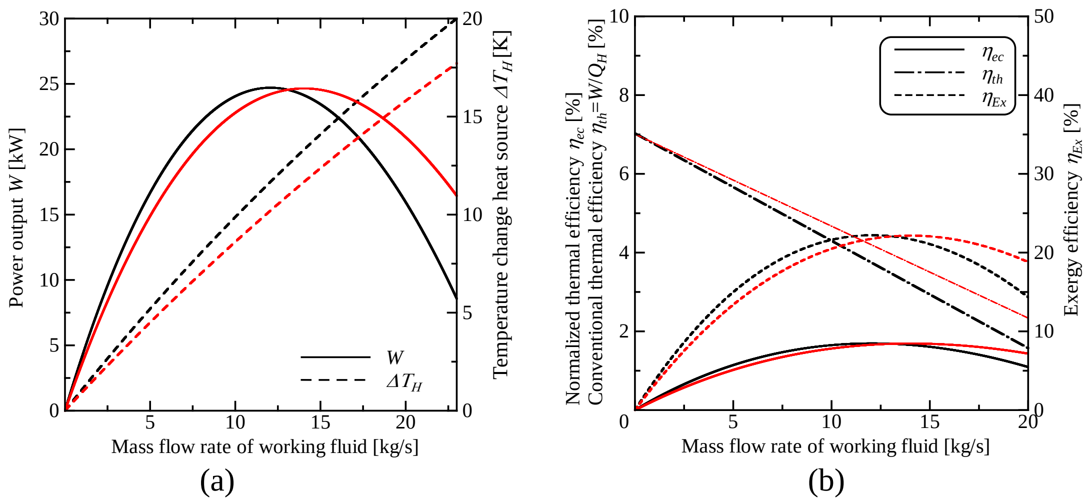
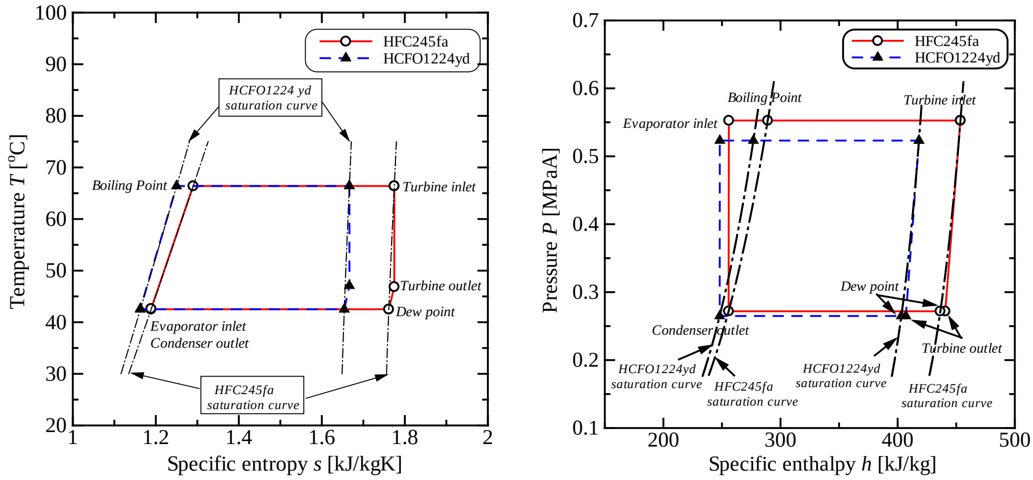


Figure 2: (a) W and $\Delta T_H = (T_H - T_L)$ and (b) normalized thermal efficiency, exergy efficiency and conventional thermal efficiency as a function of the mass flow rate of the working fluid where: T_H = 80°C T_L = 30°C; $m_H = m_L$ = 50 t/h; $NTU_E = NTU_C$ = 2.0; η_T = 60%; η_{WFP} = 60%. The black lines are HFC245fa and the red lines are HCFO1224yd(z).

3.2.2 Turbine and working fluid circulation pump efficiency: The irreversibility coefficient of the turbine and working fluid circulation pump efficiencies are calculated after normalization by the working fluid irreversibility coefficient (Figures 5 and 6). The turbine efficiency affects the internal irreversibility almost directly, while the working fluid performance has a tiny influence beyond the efficiency of >40%.



(a) *T*–*s* diagram (b) *P*–*h* diagram

Figure 3: Comparison between HFC245fa and HCFO1224yd(z) as the working fluid at the maximum power output points. (a) *T*–*s* diagram, (b) *P*–*h* diagram when T_H = 80°C; T_L = 30°C; $m_H = m_L$ = 50 t/h; $NTU_E = NTU_C$ = 1.0; η_T = 100%; η_{WFP} = 100%.

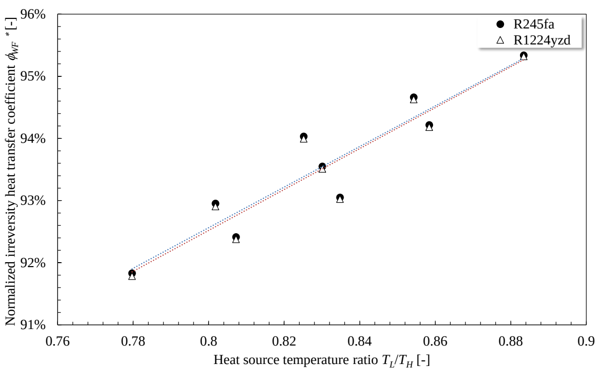


Figure 5: Dependency of the internal irreversibility coefficient of the working fluid thermal properties on the heat source temperature. Here, T_H = 70, 80, 90°C; T_L = 10, 20, 30°C; $m_H = m_L$ = 50 t/h; $NTU_E = NTU_C$ = 2.0; $\eta_T = \eta_{WFP}$ = 100%.

4. CONCLUSIONS

This research describes the effect of irreversibility on the performance in low-grade thermal energy conversion systems focused on sensible heat based on FTT and parameter analysis. The results clarify the following items:

- The theoretical maximum work formula is summarized and the effect of the coefficient of irreversibility is incorporated;
- Parameter analysis of the Rankine cycle is conducted with HFC245fa and HCFO1224yd(z) and the two working fluids' performances are clarified and the peak performance is found to be equivalent;
- The internal irreversibility of the working fluid, turbine efficiency, and working fluid circulation pump are divided and the effect of each component on the internal irreversibility coefficient is visualized.

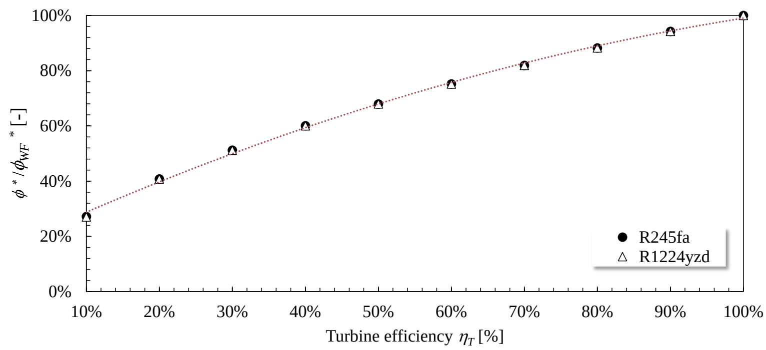


Figure 6: Dependence of the turbine efficiency on the internal irreversibility where T_H = 80°C T_L = 30°C; $m_H = m_L$ = 50 t/h; $NTU_E = NTU_C$ = 2.0; η_{WFP} = 100%.

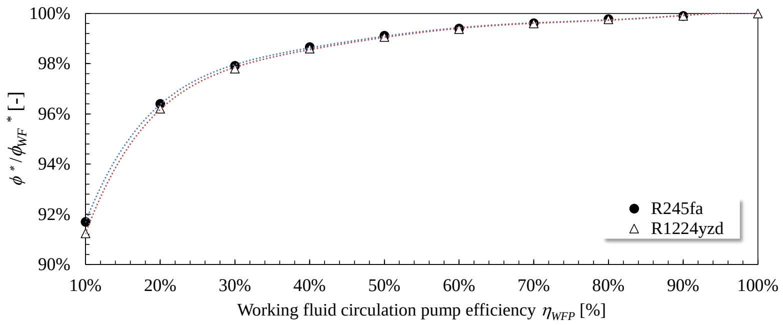


Figure 7: Dependence of the working fluid circulation pump efficiency on the internal irreversibility where T_H = 80°C T_L = 30°C; $m_H = m_L$ = 50 t/h; $NTU_E = NTU_C$ = 2.0; η_T = 100%.

NOMENCLATURE

A	heat transfer area of heat exchanger	(m^2)
C	heat capacity	(kW/K)
c_p	specific heat	(kJ/kgK)
h	specific enthalpy	(kJ/kg)
m	mass flow rate	(kg/s, t/h)
NTU	number of transfer unit	(-)
P	pressure	(kPaA)
r	ratio of high temperature heat source capacity	(-)
s	specific entropy	(kJ/kgK)
T	temperature	(K)
U	overall heat transfer coefficient	(kW/m^2K)
ε	effectiveness of heat transfer performance	(-)
ΔT_m	logarithmic mean temperature	(-)
ϕ	irreversibility coefficient	(-)
ϕ^*	normalized irreversibility coefficient by Eq.(17)	(-)
η	efficiency	(-)

Subscript

C	condenser
E	evaporator
ec	energy conversion
H	high temperature heat source
HS	heat source
I	inlet
L	low temperature heat source

m	maximum
O	outlet
T	turbine
th	thermal
WF	working fluid

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