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Thermo-economic environmental optimization of Organic Rankine Cycle for diesel waste heat recovery

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ABSTRACT

An Organic Rankine Cycle for diesel engine waste heat recovery is modeled and optimized. The design parameters are nominal capacity of diesel engine, diesel operating partial load, evaporator pressure, condenser pressure and refrigerant mass flow rate. In addition four refrigerants including R123, R134a, R245fa and R22 are selected and studied as working fluids. Then, the fast and elitist NSGA-II (Non-dominated Sorting Genetic Algorithm) is applied to maximize the thermal efficiency and minimize the total annual cost (sum of investment cost, fuel cost and environmental cost) simultaneously. The results of the optimal design are a set of multiple optimum solutions, called Pareto optimal solutions. The optimization results show that the best working fluid is R123 in both of economical and thermo dynamical view point for a specified value of output power. R245fa, R134a and R22 are placed in the next ranking, respectively. The optimum result of R123 shows the 0.01%, 4.39%, and 4.49% improvement for the total annual cost in comparison with R245fa, R22, and R134a, respectively. The above values for efficiency are obtained 1.01%, 12.79% and 10.57%, respectively. Furthermore R123 needs the highest investment cost while the environmental and fuel costs are the lowest.

1. Introduction

ORC (Organic Rankine Cycle) enable efficient power generation unit from low-grade heat sources by replacing water with organic working fluids such as refrigerants or hydrocarbons. Najjar and Radehwan recovered waste heat by combining a heat-exchanger gas turbine cycle with closed Organic Rankine Cycle [1]. Some authors investigate the effect of working fluids on Organic Rankine Cycle for waste heat recovery [2–7]. Mago et al. presented an analysis of regenerative Organic Rankine Cycles using dry organic fluids to convert waste energy to power from low-grade heat sources [8]. Dai et al. described the Rankine cycles for low grade waste heat recovery with different working fluids [9]. Papadopoulos et al. presented the first approach to the systematic design and selection of optimal working fluids for ORCs (Organic Rankine Cycles) based on CAMD (computer aided molecular design) and process optimization techniques [10]. The results were compared in the regions when net power outputs were fixed at 10 kW. The outcomes indicated that R11, R141b, R113 and R123 manifested slightly higher thermodynamic performances than the others. Some authors investigated the performance of a low-temperature solar Rankine cycle system using various working fluids [11–19]. Shengjun et al. presented an investigation on the parameter optimization and performance comparison of the fluids in subcritical ORC and transcritical power cycle in low-temperature binary geothermal power system [20]. A supercritical Rankine cycle using zeotropic mixture working fluids for the conversion of low-grade heat into power was proposed and analyzed by Chen et al. [21]. Unlike a conventional Organic Rankine Cycle, a supercritical Rankine cycle does not go through the two-phase region during the heating process. By adopting zeotropic mixtures as the working fluids, the condensation process also happens non-isothermally. Both of these features create a potential for reducing the irreversibilities and improving the system efficiency. Alessandro Franco analyzed and discussed the exploitation of low temperature, water-dominated geothermal fields with a specific attention to regenerative Organic Rankine Cycles [22]. Yamada et al. proposed a new pump less Rankine-type cycle for power generation from low-temperature heat sources [23]. The new cycle mainly consists of an expander, two heat exchangers, and switching valves for the expander and heat

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Nomenclature

a	annual cost coefficient (−)
A	condenser heat transfer surface area (m^2)
C	investment cost ($)
h	enthalpy (kJ/kg K)
i	interest rate (−)
LHV	fuel lower heating value (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
p	pressure (kPa)
$\dot{Q}$	rate of heat transfer (kW)
U	overall heat transfer coefficient (W/m^2 K)
$\dot{W}$	power (kW)
y	depreciation time (year)

Greek abbreviation

ψ_{em}	pollutant emission cost ($/kg)
ψ_f	fuel cost ($/kg)
τ	hours of operation per year (h)
ν	specific volume (m^3/kg)
ε	total cycle thermal efficiency (−)
η	efficiency (−)

Subscripts

a	actual
D	diesel
i	inlet
o	outlet
evap	evaporator
T	turbine
s	isentropic
cond	condenser
LMTD	logarithmic mean temperature difference
CW	cooling water
p	pump
env	environment
inv	investment
f	fuel
nom	nominal
PL	partial load (%)
total	total
wj	water jacket

exchangers. Chen et al. studied transcritical Rankine cycles using refrigerant R32 (CH_2F_2) and carbon dioxide (CO_2) as the working fluids for the conversion of low-grade heat into mechanical power [24]. Wang et al. used waste heat from stationary and mobile engine cycles to generate cooling for structures and vehicles [25]. It combined an ORC (Organic Rankine Cycle) with a conventional vapor compression cycle. In order to maintain high system performance while reducing size and weight for portable applications, micro channel based heat transfer components and scroll based expansion and compression were used. Sun and Li presented a detailed analysis of an Organic Rankine Cycle heat recovery power plant using R134a as working fluid. Mathematical models for the expander, evaporator, air cooled condenser and pump were developed to evaluate and optimize the plant performance [26]. Wagar et al. developed a model of an ammonia–water Rankine heat engine and examined with the inclusion of a two-phase expansion process. A general model for the optimal cycle was developed based upon the maximum operating temperature and the operating concentration [27]. Jing Li et al. presented a quantitative study on the convection, radiation, and conduction heat transfer from a kW-scale expander. A mathematical model was built and validated [28]. Xu and He proposed a regenerative Organic Rankine Cycle that used a vapor injector as the regenerator [29]. The thermal performance of both the novel cycle and the basic ORC was calculated and compared by using R123 as the working fluid. Invernizzi et al. investigated the possibility of enhancing the performances of micro-gas turbines through the addition of a bottoming Organic Rankine Cycle [30]. They showed ORC cycles were particularly suitable for the recovery of heat from sources at variable temperatures. Quoilin and et al. developed a thermodynamic model of a waste heat recovery ORC in order to compare both the thermodynamic and the thermoeconomic performance of several typical working fluids for low to medium temperature-range ORCs [31].

In this paper after thermo-economic modeling of ORCD (Organic Rankine Cycle for Diesel) waste heat recovery, this equipment is optimized by maximizing the thermal efficiency as well as minimizing the total annual cost, simultaneously. nominal capacity of diesel engine, diesel operating partial load, evaporator pressure, condenser pressure and refrigerant mass flow rate are taken as five design parameters and fast and elitist NSGA-II (Non-dominated Sorting Genetic Algorithm) is applied to provide a set of Pareto multiple optimum solutions.

As a summary, the followings are the contribution of this paper into the subject:

- Applying four simultaneous system analysis including energy, efficiency, economic and environment (4E analysis) for equipment selection.
- Selecting the nominal capacity of diesel engine, diesel operating partial load, evaporator pressure, condenser pressure as well as refrigerant mass flow rate as design parameters (not selected as a group of variables in other available literature).
- Performing the multi objective optimization of ORCD with efficiency and the total annual cost as two objectives (not selected in other available literature).
- Applying the optimization for four working fluids including R123, R134a, R245fa and R22.
- Sensitivity analysis of change in total annual cost when the price of diesel fuel varies.

2. Thermal modeling

Schematic diagram of an ORCD (Organic Rankine Cycle for Diesel) waste heat recovery is shown in Fig. 1. It mainly consists of diesel engine and Rankine cycle including turbine, condenser, pump and evaporator (heat exchanger). Refrigerant enters the evaporator at a given pressure and temperature (state 4), where it is vaporized by the absorbed heat energy from waste heat recovery in diesel engine. The refrigerant exits the evaporator as superheated vapor (state 1), and then passes through the expander (turbine). The high quality refrigerant (state 2) enters the condenser and transfers heat to the cooling tower. The condensed liquid refrigerant (state 3) is next pumped to the evaporating pressure and enters directly to the evaporator (state 4). There are two sources of power generation here, including the net power from diesel engine and Rankine cycle.

In order to do the thermal modeling, mass and energy balances on the system are required to determine the flow rates and energy transfer rates at the control surface. Appling the first law of thermodynamic in the steady state, one can find the formula for mass and energy balance as follow [32]: