DESIGNING ORGANIC RANKINE CYCLE PLANTS BASED ON A DESIGN TO RESOURCE METHOD

Denny Budisulistyo and Susan Krumdieck[1]
[1]University of Canterbury, Private Bag 4800, Christchurch, 8140 New Zealand
denny.budisulistyo@pg.canterbury.ac.nz
susan.krumdieck@canterbury.ac.nz

Keywords: *Organic rankine cycle, waste heat recovery, design to resource, optimum design*

ABSTRACT

This paper discusses a methodology for analysing and optimization of organic Rankine cycle (ORC) designs using a new thermo-economic design to resource (DTR) method. The objective of the DTR method is to obtain the best designs, which are the closest match to the resource and the most cost-effective. The design analysis is constrained by the available main components and heat resource. The ratio of net power output to the total heat exchanger area is used as the objective function. The new design methodology was implemented on an existing lab-scale ORC as a case study. Experiments were conducted to obtain the data to identify the heat transfer coefficients of the real processes and validate the simulation model results. Design evaluations were carried out on the ORC plant by using three Capstone gas turbine load conditions and four design alternatives. The results indicate that design 1 has the highest objective function of all the design alternatives. It is able to increase the objective function from 100% to 391% of the base case depending on the Capstone gas turbine load conditions. The design to resource analysis reveals that the ORC plant is more suitable to Capstone load at condition 1 with the highest waste heat utilization rate (UR) of 76.9%.

1. INTRODUCTION

The manufacturing process industries release a huge amount of energy to the environment in the form of waste heat. Due to a significant increase of fuel prices and environment issues, an important number of new solutions have been proposed to increase conversion efficiencies to optimally exploit the potential energy resources.

The thermo-economic optimization of WHR ORCs is also an area of the research with a large number of papers. Li and Dai et al. [1] investigated the effect of internal heat exchanger (IHE) and superheat degree on the thermo-economic performance of ORC using zeotropic mixtures. The results indicate that the IHE has a higher impact on thermal and exergy efficiencies of the ORC with zeotropic mixtures than that of the ORC with pure fluid. The rising superheat degree impacts to the decline of the net power output but increase the thermal and exergy efficiencies at the constant turbine inlet pressure. Imran and Park et al. [2] analysed the thermo-economic optimization of basic and regenerative ORC under constant heat source condition. The optimization results show that R245fa is best working under considered conditions and basic ORC has low specific investment cost and thermal efficiency compared to regenerative ORC. Hajabdollahi et al. [3] modelled and optimized a WHR ORC for diesel engine. 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, but it needs the highest investment cost which the environmental and fuel costs are the lowest. Quoilin, Declaye [4] proposed a fluid selection based on thermo-economic considerations. The results stated that the thermo-economic optimization leads to the selection of a higher evaporating temperature, because it increases high-pressure vapour density and decreases the cost of the expander and of the evaporator. In addition, if the thermodynamic optimization can give a good ideas of the best fluids, it will not be necessarily to select the optimal working fluid in the terms of economical aspect.

Most of the foregoing research works tend to focus on some aspects of the system design such as fluid selection, cycle optimization and thermo-economic optimization. They do not consider selection and design as two terms that are interchanged during the ORC design process. According to Jaluria [5], selection and design are frequently employed together in the development of a system. Design involves starting with a basic concept, modelling and evaluating different designs and obtaining a final design that fulfils the given requirements and constraints. Based on design results, the requirements and specifications of the desired component or equipment are matched with whatever is available in the markets. If an item possessing the desired characteristics is not available, design is needed to obtain one that is acceptable for the specific purpose.

The main objective of the study is to propose a comprehensive design methodology for a new ORC plant and an existing ORC plant to obtain a cost-effective optimum design. The methodology considers the selection of components required for a system as a step in design process of an ORC system and the approach of design to resource. The objective of the DTR method is to obtain the best design, which is the closest match to the resource and the most cost-effective design. The methodology is applied to a small-scale ORC system for WHR of the Capstone gas turbine. Main components are modelled in detail according to real products. The models are validated by experimental data to have more reliable prediction of the system performance.

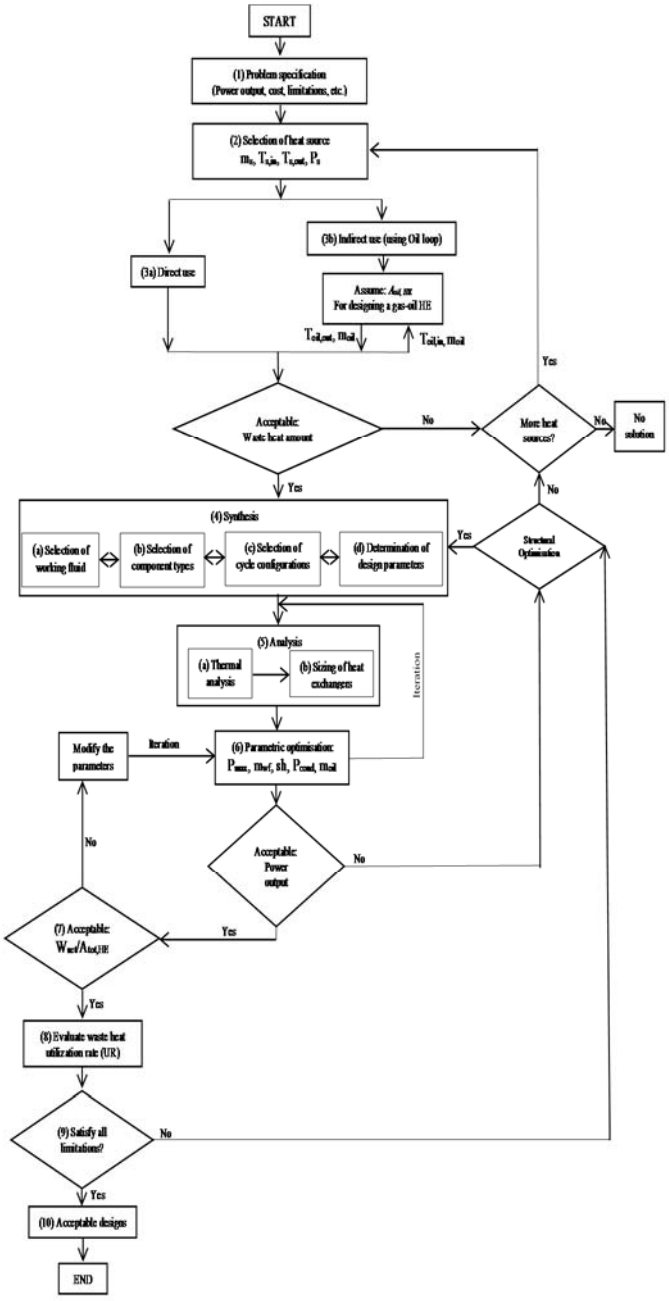


Figure 1: Flow chart for developing the ORC designs based on DTR method

2. DTR METHODOLOGY

Figure 1 shows a design methodology for designing and optimizing the ORC plant based on the DTR method for WHR applications. The main parameters for optimization are $W_{net}/A_{tot,HE}$.

1. Problem specification:
   The goals and some limitations of the cycle and each component are fixed. Two important goals are power output and optimal cost. These goals have to be satisfied by all the steps in the methodology. The limitations must be fulfilled in the cycle to achieve the goals.
2. Selection of heat source:
   The possible heat sources need to be evaluated to identify sources with higher heat power level using energetic and/or exergetic studies. The maximum heat available in the exhaust gas is that heat rejected under the hypothesis that the exhaust gas is cooled to the ambient temperature at 25^0C [6]. However, in some cases the exhaust gas temperature ($T_{s,out}$) of waste heat resources requires to be above the dew temperature level to prevent corrosive effects [7]. The selected heat sources must have the available heat power higher than power output objective. If the amount of available waste heat is less than the requirement, the design problem has no solution.
3. Selection of heat recovery setup:
   Two different setups can be used in waste heat recover (WHR) system [4]:
   (3a) Direct use: direct heat exchange between the waste heat source and the working fluid
   (3b) Indirect use: a heat transfer fluid loop is integrated to transfer the heat from the waste heat site to the evaporator.
   The heat transfer area of a gas-oil heat exchanger (A_{HE}) is assumed based on the objective of power output and an oil pump capacity used in the oil loop. Therefore, the oil-loop can deliver the heat power according to the ORC system requirement.
4. Synthesis:
   Synthesis is concerned with combining separated cycle elements into a thermodynamic cycle. The step consists of four-cycle elements that should be conducted simultaneously.
   a. Selection of working fluid: The selection of the most appropriate working fluid is very important step in designing ORC systems because an used working fluid type influences a produced power output, sizes of the components, system stability, cost, safety and environmental issues.
   b. Selection of component types: the type of four basic main components of the plant (turbine, evaporator, condenser, and pump) is selected. The selection depends on operating conditions and the size of the plant. The turbine can be categorised into two main types: turbomachines (the axial turbine and the radial inflow turbine) and positive displacement types (piston, scroll, screw and vane expanders). The turbomachines are not suitable for very small-scale units because their rotating speed increases significantly with decreasing turbine output power [8]. The positive displacement types are good for small scale ORC units, while technically mature turbomachines are available on the market for large ORC units.
   c. Selection of cycle design: Four types of cycle layouts are available for ORC cycle: 1) subcritical cycle without a recuperator, 2) subcritical cycle with a recuperator, 3) supercritical cycle without a recuperator, and 4) supercritical cycle with a recuperator.
   d. Determination of design parameters: the initial assumption values for creating a thermodynamic cycle. The parameters are superheat, subcooling, pinch point, pump and turbine efficiencies.
5. Analysis
   This step involves thermal analysis and sizing of the heat exchangers.
   a. Thermal analysis generally entails solving mass and energy balances in overall thermodynamic cycle and in each component of the cycle. The thermal analysis here is implemented based on the strategy proposed by Franco and Villani [9] et al. The strategies divide the ORC system into

three subsystems (thermodynamics cycle, evaporator and condenser) and two hierarchical levels which sequentially define system level (thermodynamic cycle) and component level (evaporator and condenser). Figure 2 shows hierarchical organization proposed by Franco et al. At the system level, the thermal problems (mass and energy balances) are solved by thermodynamic variables matching between binary cycle and geothermal resource. At the component level, the convergent results from the system optimization level produce the input data for the detail design of component level (evaporator and condenser). The results of the optimum component design (pressure losses (Δp), pumping power (W_T) and fan power (W_{fans}) are iterated in the system level. Thus, the results of the component level optimization can affect the results of the first level optimization particularly in the design of the dry cooling system.

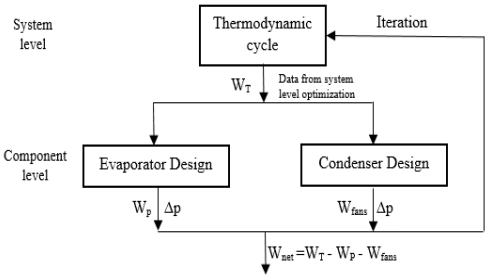


Figure 2: Hierarchical organization for the thermal analysis in the design of binary plants

b. *Sizing of heat exchangers*. The dimensions of the various sections of the heat exchangers (pre-heater, evaporator, superheater and condenser) are calculated by considering the required heat transfer, the allowed pressure drop and the minimum allowed temperature difference.

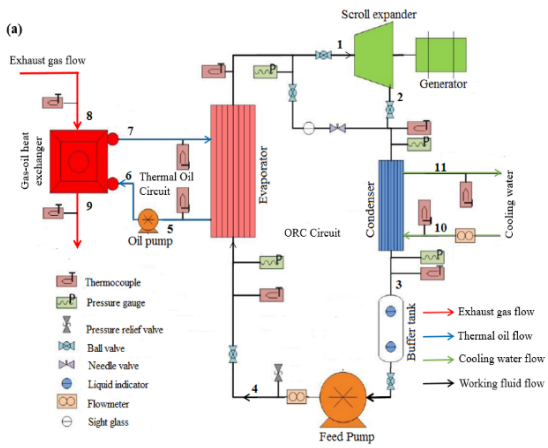
6. Parametric optimization
   The parametric optimization involves five decision parameters: (1) cycle maximum pressure (P_{max}); (2) mass flow of the working fluid (m_{WF}); (3) degree of superheating (sh), measured from the specific entropy of the point on saturated vapour curve for subcritical cycles; (4) condensation pressure (P_{cond}) [10] and an additional parameter: (5) mass flow of oil loop (m_{oil}). The iteration is generally necessary to obtain an acceptable power output. If the power output obtained from parametric optimisation does not satisfy the target of power output, structural optimisation and/or heat sources selected will have to be considered. If not, the design problem has no solution.
7. Acceptable: $W_{net}/A_{tot,HE}$
   The ratio of total net power output (W_{net}) to total heat transfer area ($A_{tot,HE}$) is suggested as an objective function to obtain the best cost-effective design [11]. This is based on the assumption that the total cost of the heat exchanger area dominates largely to the total cost of ORC especially for the system utilizing a low temperature of waste heat.
8. Evaluate waste heat utilization rate
   The concept of waste heat utilization rate (UR) is applied to further analysis of heat recovery capability of each ORC design. This concept is able to indicate how match between the design and the heat resource. The waste heat UR is the ratio of heat absorbed by the ORC system to maximum available heat power in a heat source [6].
9. Any limitations have been fulfilled by the designs:
   Other pre-imposed limits and targets must be evaluated before determining the best final design. In this last step, other feasibility criteria such as limitations of component operating conditions and/or maximum installation cost could be evaluated and the best final design must fulfil all the limits that have been fixed in the first step.
10. Acceptable designs
    The last step is to conclude the acceptable designs among several design alternatives.

The methodology is implemented in the small-scale ORC plant in our laboratory to illustrate the methodology implementation in redesigning the heat exchangers in the system.

3. THE ORC PLANT

Figure 3 shows the principle schematic diagram of the bottoming ORC for WHR of the Capstone gas turbine. The ORC system consists of four separate fluid circuits: exhaust gas flow (in red line), thermal oil circuit (in blue line), ORC circuit (in black line) and cooling water (in green line). All circuits are connected through heat exchangers. The whole system operates as follows: the exhaust gas from the Capstone gas turbine rejects heat to thermal-oil circuit through a gas-oil HE and then is discharged to atmosphere; working fluid in vapour state (point 1) flows into the scroll expanders, and its enthalpy is converted into expansion power; low pressure vapour (point 2) exits from scroll expander and flows into condenser where it uses cooling water to condense working fluid into saturated liquid (point 3), the buffer tank after the condenser is used to maintain sufficient liquid therefore pump does not run dry; working fluid is pumped into high pressure state (point 4), and then is boiled through evaporator and leaves as a superheated vapour (point 1). Thus a whole cycle completes. The cycle is repeated in a closed loop to generate continuative power.

The current ORC system consists of a scroll expander, a feed pump, an oil pump, oil-working fluid heat exchanger (evaporator), a water-working fluid heat exchanger (condenser) and a gas-oil heat exchanger.



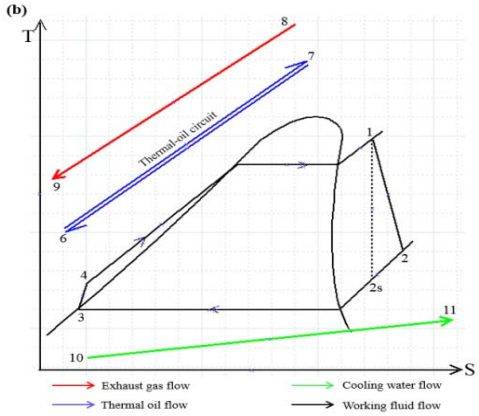


Figure 3: Schematic diagram (a) and T-S diagram of the ORC system for WHR (b)

3.1. Modelling the ORC system

The modelling approach consists of developing a semi-empirical model for a scroll expander and LMTD method for heat exchangers. The model uses R245fa as a working fluid to represent a zeotropic mixture (M1) used in the ORC system. The M1 consists of R245fa and R365mfc with a mole fraction of 50% and 50%, respectively. Both fluids have almost similar properties.

3.1.1. The scroll expander model

The semi-empirical model of a scroll expander used here-under is adopted from one proposed and validated by the authors [12]. The expander mechanical power ($\dot{W}_T$) can be divided into the internal expansion power and the mechanical losses ($\dot{W}_{loss}$). These losses are lumped into one unique mechanical loss torque T_{loss}, that is a parameter to identify. The expander mechanical power is expressed by

$$\dot{W}_T = \dot{W}_{in} - \dot{W}_{loss} = \dot{W}_{in} - 2.\pi.N.T_{loss} \quad (1)$$

where N is rotating speed of the expander shaft.

The ambient losses are calculated by introducing a global heat transfer coefficient AU_{amb} between the envelope and the ambient:

$$\dot{Q}_{amb} = AU_{amb}(T_w - T_{amb}) \quad (2)$$

The uniform temperature of a fictitious envelope (T_w) is computed by establishing a steady-state heat balance on this envelope, as proposed by Winandy, Saavedra [13]:

$$\dot{W}_T - \dot{Q}_{ex} + \dot{Q}_{su} - \dot{Q}_{amb} = 0 \quad (3)$$

where $\dot{Q}_{su}$ and $\dot{Q}_{ex}$ are supply and exhaust heat transfers that are calculated by introducing a fictitious metal envelope of the uniform temperature (T_w). The

3.1.2. Evaporator and condenser models

The evaporator and condenser use the plate heat exchangers and they are modelled by means of the LMTD method for counter-flow heat exchangers. The heat exchanger is subdivided into three zones: liquid zone (l), two-phase zone (tp) and vapor zone (v). Every zone is characterized by a heat transfer area (A) and a heat transfer coefficient (U). The heat transfer coefficient U in each zone is calculated by considering two convective heat transfer resistances in series (refrigerant and secondary fluid sides).

$$\frac{1}{U} = \frac{1}{h_r} + \frac{1}{h_{sf}} \quad (4)$$

The total heat transfer area of the heat exchanger is summed the respective heat transfer area of each zone:

$$A_{tot} = A_l + A_{tp} + A_v \quad (5)$$

3.1.3. The gas-oil heat exchanger model

The gas-oil heat exchanger is modelled by means of the LMTD method for a cross-flow heat exchanger. The heat transfer process is in single phase between heat transfer oil and the exhaust gas of waste heat.

3.1.4. Pump model

A non-isentropic compression process models the pump.

3.2. Comparison between experimental and model results

To ensure that the experimental data are collected in steady state conditions, a steady state standard proposed by Woodland, Braun [14] was used. The change of pressure and rotating equipment speed is less than 2% and the change of temperature is less 0.5 K

3.2.1. Expander model validation

The input variables of the expander model are the supply pressure, the supply temperature, the exhaust pressure, the ambient temperature and the rotational speed of the expander. The parameters of the expander model are tuned to best fit the three model outputs (the mass flow rate displaced by the expander, the delivered mechanical power and the exhaust temperature) to experimental data.

The parameters of the model are identified by minimizing an error-objective function accounting for the errors on the prediction of the mass flow rate, shaft power, and exhaust temperature (using a direct algorithm available in the EES software):

$$error = \frac{1}{3}\left(\sqrt{\sum_1^{N_{tests}}\left(\frac{\dot{M}_{calc}-\dot{M}_{meas}}{\dot{M}_{calc}}\right)^2}\right) + \frac{1}{3}\left(\sqrt{\sum_1^{N_{tests}}\left(\frac{\dot{W}_{T,calc}-\dot{W}_{T,meas}}{\dot{W}_{T,calc}}\right)^2}\right) + \frac{1}{3}\left(\sqrt{\sum_1^{N_{tests}}\left(\frac{\dot{T}_{ex,calc}-\dot{T}_{ex,meas}}{\dot{T}_{ex,calc}}\right)^2}\right) \quad (6)$$

The model requires nine parameters that is identified to best match the values of the outputs to the experimental results. They are listed in Table 1:

Table 1: Parameters of semi-empirical model.

Swept volume	$V_{s,T}$	12 cm^3
Built-in volume ratio	$r_{v,in}$	3.5
Leakage area	A_{leak}	11.53 mm^2
Supply heat transfer coefficient	$AU_{su,n}$	26.02 W/K
Exhaust heat transfer coefficient	$AU_{ex,n}$	144.7 W/K
Heat transfer coefficient with the ambient	AU_{amb}	144.5 W/K
Supply port cross-section area	A_{su}	44.17 mm^2
Nominal mass flow rate	$\dot{M}_n$	1.207 kg/s
Mechanical loss torque	T_{loss}	0.6024 Nm

A relative error between the predictions by the model and the measurements is about 9.6% for the exhaust temperature, 8.3% for mass flow rate and 8.5% for electrical power output.

3.2.2. Heat exchanger model validation

The experimental data of the ORC-B plant in our laboratory is used to validate the heat exchanger models described above. The coefficient values for two-phase of the evaporator and condenser are identified by imposing some measurements as input variables and by

minimizing the deviation between the measured and model output variables. The coefficient values of evaporator and condenser are 19.18 and 4.253, respectively and they are listed in Table 2.

Table 2: Heat exchanger model parameters.

Evaporator	Condenser
$h_{tp} = 19.18 h_l Bo^{0.5}$	$h_{tp} = 4.253\, h_l \left(0.25\, Co^{-0.45}\, Fr_l^{0.25} + 75\, Bo^{0.75}\right)$

For given inlet temperatures of hot and cold fluids and saturation pressure, the evaporator and condenser models calculate the heat flow rate and the exhaust temperature. The exhaust temperature of evaporator and condenser models is predicted with a relative error of about 4.3% and 1%, respectively. The gas-oil heat exchanger in the oil loop calculates the outlet temperatures and heat flow rate for given inlet exhaust gas and oil temperature and the pressure level of both fluids. The comparison between measured and predicted exhaust oil temperatures of the gas-oil HE with a relative error of about 2.25%.

3.3. Base case design performance

The possible maximum performance of the ORC plant using the selected main components described above are calculated as a base case. The base case is calculated to be used as a reference of comparison to new alternatives produced by applying the proposed methodology. Table 3 shows the existing ORC plant performance with three Capstone gas turbine load conditions. The main parameters of Capstone gas turbine in each condition are shown in Table 5. The optimization of the plant performance is constrained by the maximum outlet temperature of the evaporator in oil side at 100^{0}C (point 5 in Figure 1) to avoid the damage of the seals in the oil pump [15] and maximum characteristic parameters of the main components such as pinch point and a revolution speed (RPM) of the expander.

4. APPLICATION OF THE METHODOLOGY

The proposed methodology is implemented to a small scale of ORC plant described above for modifying the size of the heat exchangers in the system to obtain the more economical designs. Three heat exchangers in the system are a gas-oil HE, evaporator and condenser. The four design alternatives that are investigated in the application of the methodology are shown in Table 4. Note that "v" means that the heat exchanger is redesigned and "-" means that the heat exchanger is not redesigned.

Table 3: The possible maximum performance of the current ORC plant.

Parameters	Condition 1	Condition 2	Condition 3
N (RPM)	1950	3134	3189
$\dot{M}$ (kg/s)	0.0312	0.0498	0.0525
P_{max} (kPa)	499	675	717
$T_{T,in}$ (^{0}C)	126.7	121.5	129.2
P_{cond} (kPa)	107	124	124
m_{oil} (kg/s)	0.12	0.31	0.24
$T_{out,oil}$ (kg/s)	89.33	100	100
$PP_{gas\text{-}oil\ HE}$ (^{0}C)	70.2	65.3	88
PP_{ev} (^{0}C)	1.24	1.75	3.14
PP_{cond} (^{0}C)	0.24	0.38	0.47
W_T (W)	154	449	515
W_P (W)	11.1	25	28
W_{net} (W)	143.4	424	487

Table 4: Design alternatives investigated by the methodology application.

Heat exchanger	Design 1	Design 2	Design 3	Design 4
Gas-oil HE	v	v	-	-
Evaporator	v	v	v	v
Condenser	v	-	v	-

4.1. Problem specification

It has been shown in previous works of the WHR system that the power output should be maximized instead of the cycle efficiency [16, 17]. However, the most economical design is considered as the most important goal of the design. Considering these aspects into account, the methodology is applied by two objectives:

- Maximum ratio of W_{net}/A_{HE}
- The power output must be higher than a base case with the same Capstone gas turbine load.

The limitations of the problems are fixed as follows:

- The ORC system is designed with the same expander and working fluid for all design alternatives.
- The maximum outlet oil temperature of the evaporator is 100^{0}C to avoid melted seals in the oil pump.

4.2. Selection of heat source

The selection process depends on the objectives and limitations established in the specification problem (first step). The heat source candidates must have an available power higher than 1 kW since the system uses an expander with a capacity of 1 kW. The heat source in this study used a waste heat of the Capstone gas turbine. Three important conditions of the engine operation are shown in Table 5.

The available power between the inlet and outlet condition (8 and 9) of each heat source is calculated considering the gases as ideal and perfect gases using an energetic analysis:

$$Q = \dot{M}.(H_8 - H_9) = \dot{M}.c_p(T_8 - T_9) \quad (7)$$

where C_p is 1 kJ/kgK and 1.15 kJ/kgK for fresh air and combustion gases, respectively. The ambient conditions are considered as the reference state. Assuming that the exhaust gas is cooled into 100^{0}C, then the available power of the waste heat in three different Capstone gas turbine load conditions is shown in the Table 5.

Table 5: Three typical conditions of the Capstone gas turbine.

Parameters	Condition 1	Condition 2	Condition 3
Engine Power output [kW]	5	10	15
Temperature of exhaust gas [C]	230	251	268
Mass flow of exhaust gas [kg/s]	0.14	0.18	0.21
The available power [kW]	20.9	31.3	40.6

4.3. Selection of heat recovery setup

The design of WHR in this chapter selects an indirect use setup to recover the heat source due to more stable and controllable systems than choosing a direct use setup. The existing heat transfer area of a gas-oil heat exchanger (A_{HE}) at 1.025 m^2 (oil-side area) is used as an initial assumption. The mass flow rate of oil ranging from 0.028 kg/s to 0.354 kg/s is used in the optimization analysis. In the case of design 1 and design 2, both parameters $A_{oil,HE}$ and m_{oil} are used in the optimisation because these conditions redesign the gas-oil heat exchanger design.

4.4. Synthesis

Synthesis is concerned with combining cycle elements into an ORC cycle.

4.4.1. Selection of working fluid

For this study, a zeotropic mixture (M1) has been considered as a working fluid. The fluid is selected because the fluid is available in New Zealand with a small quantity at approximately 15 kg [15]. Moreover, the zeotropic mixture is expected to perform better than pure fluids in an ORC system.

4.4.2. Selection of main component types

The scroll expander and positive displacement pump are selected, since the design is for a small-scale ORC plant. The plate heat exchanger is selected as a heat exchanger construction type for evaporator and condenser.

4.4.3. Selection of cycle configurations

In WHR applications, the output power should be maximized instead of the cycle efficiency [17]. The subcritical cycle without a recuperator is therefore selected in the present work. The basic configuration integrates four main components: an evaporator, a turbine, a condenser and a working fluid pump.

4.4.4. Determination of cycle parameters

The assumptions of superheat, sub-cooling and pinch point are required by the design alternatives that need to redesign the heat exchangers, especially design 1 that needs to redesign all heat exchangers in the system. The isentropic efficiency of pump is set at a constant value of 80% and the turbine efficiency is calculated by a semi-empirical model. The semi-empirical model represents more precisely the real turbine performance.

4.5. Analysis

Analysis and optimisation (step 5 and step 6) are two consecutive steps that are connected each other. The main objective function of optimization is to maximize the ratio of W_{net}/A_{HE} considering a higher power output than the base case. The optimization is constrained by using the same expander and working fluid for all design alternatives and the maximum outlet oil temperature of evaporator at 100^0C. The optimization of models is carried out by means of a direct algorithm available in the EES software [18]. An iterative process is conducted between step 5 and step 7 in Figure 1.

Two heat exchanger models (one modelling the evaporator and one for the condenser) are used to calculate the heat transfer areas required by every heat exchanger in the ORC system. The inputs of the component models are the optimal results of the system level (described in Figure 2). In this calculation, the pressure drops are neglected. The results of these sizing problems are shown in Table 6. The sizes of the existing evaporator and condenser (the base case) are significantly larger than the required heat transfer areas especially under the low Capstone gas turbine load condition (condition 1). In comparison to design 3, which has the same size of the gas-oil HE as the existing ORC plant, the oversize of the existing evaporator and condenser under condition 1 is 153% and 137%, respectively. These oversized figures decrease by increasing Capstone gas turbine load (from condition 1 to condition 3). The oversize of the evaporator and condenser under condition 2 are 67% and 88%, respectively, while condition 3 reduces the oversize of evaporator and condenser at 42% and 58%, respectively. Moreover, the size of the existing gas-oil HE is significantly small for condition 1 and 2. The existing size is more suitable for condition 3, in which the different size of base case from design 1 and design 2 under condition 3 is only 5% and 13.9%, respectively.

Table 6: Heat exchanger sizes

	HE areas	Base case	Design 1	Design 2	Design 3	Design 4
Condition 1	$A_{gas\text{-}oil\ HE}$ (m^2)	1.025	1.877	2.314	1.025	1.025
	A_{ev} (m^2)	6.400	3.662	3.457	2.531	2.52
	A_{cond} (m^2)	1.805	0.944	1.805	0.759	1.805
	A_{tot} (m^2)	9.23	6.483	7.576	5.814	5.350
Condition 2	$A_{gas\text{-}oil\ HE}$ (m^2)	1.025	1.236	1.215	1.025	1.025
	A_{ev} (m^2)	6.400	4.209	4.021	3.828	3.507
	A_{cond} (m^2)	1.805	1.049	1.805	0.961	1.805
	A_{tot} (m^2)	9.23	6.494	7.041	5.814	6.337
Condition 3	$A_{gas\text{-}oil\ HE}$ (m^2)	1.025	0.973	0.883	1.025	1.025
	A_{ev} (m^2)	6.400	4.257	3.880	4.517	3.981
	A_{cond} (m^2)	1.805	1.104	1.805	1.141	1.805
	A_{tot} (m^2)	9.230	6.334	6.568	6.683	6.811

4.6. Acceptable power output

Figure 4 shows the results of optimum power outputs produced by four design alternatives with three Capstone gas turbine conditions. They are compared to the power outputs produced by the base case. The optimal design parameters obtained at different designs are summarized in Table 6. The new four designs produce higher power outputs than the base case under different Capstone gas turbine load conditions. The power outputs produced by four design alternatives under condition 1 increase significantly in comparison to the base case under the same condition 1, because the power output of base case is very low. This occurs because the existing gas-oil HE size is smaller and the existing evaporator is significantly larger than the requirement size of the heat exchangers for the load condition (condition 1). As a result, they cause a crossover of both temperature profiles with only a low mass flow rate of oil loop in the base case results. Thus, re-sizing of the gas-oil HE and evaporator sizes under condition 1 and 2 influences on significantly higher increment of power output than the base case. The power outputs increase with an increasing of the Capstone gas turbine load conditions (from condition 1 to condition 3), because higher grade of the exhaust gas is easier to be recovered by the ORC system.

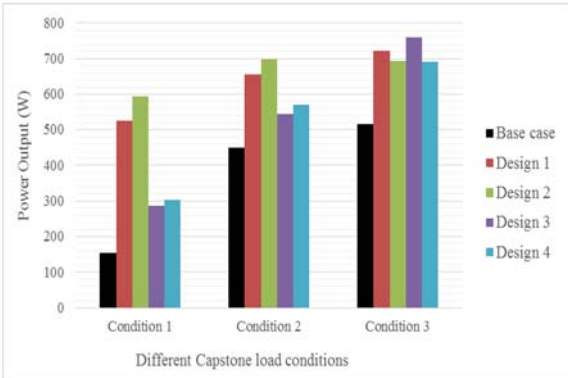


Figure 4: The optimum power output of four designs compared to a base case.

Table 6: Optimum design parameters of new four designs.

Parameters		Condition 1	Condition 2	Condition 3
Design 1	N (RPM)	3590	3590	3570
	$\dot{M}$ (kg/s)	0.0597	0.0667	0.0704
	P_{max} (kPa)	685.2	766.7	810
	$T_{T,in}$ (^{0}C)	79.41	83.77	85.96
	P_{cond} (kPa)	128	128	128
	m_{oil} (kg/s)	0.25	0.29	0.35
Design 2	N (RPM)	3541	3491	3591
	$\dot{M}$ (kg/s)	0.0609	0.0666	0.0660
	P_{max} (kPa)	703	773.3	758.4
	$T_{T,in}$ (^{0}C)	80.4	84.11	83.35
	P_{cond} (kPa)	108.8	107.8	108
	m_{oil} (kg/s)	0.26	0.31	0.34
Design 3	N (RPM)	3096	3564	3478
	$\dot{M}$ (kg/s)	0.0465	0.0608	0.0726
	P_{max} (kPa)	562.8	700	844.4
	$T_{T,in}$ (^{0}C)	72.04	80.23	87.63
	P_{cond} (kPa)	128	128	128
	m_{oil} (kg/s)	0.26	0.28	0.35
Design 4	N (RPM)	2651	3298	3397
	$\dot{M}$ (kg/s)	0.0449	0.0597	0.0664
	P_{max} (kPa)	575	708.7	779.4
	$T_{T,in}$ (^{0}C)	73	80.7	84.4
	P_{cond} (kPa)	106	106	108
	m_{oil} (kg/s)	0.26	0.29	0.25

4.7. Acceptable W_{net}/A_{HE}

Figure 5 shows the results of objective function with three different conditions of Capstone gas turbine load for base case and four new design alternatives. The figures are generally increased by increasing Capstone gas turbine load (from condition 1 to condition 3). This pattern is the same trend as the power output because the figure is influenced by the power output level. The objective function achieves the highest level for the design 1 and the lowest level for the design 4 in three Capstone gas turbine load conditions. Moreover, the objective function of design 3 increases significantly and reaches almost the same level as design 1 under condition 3, but the figure of design 3 is lower than the design 1 and 2 under condition 1 and 2. This occurs because the heat transfer area of the current gas-oil HE is more suitable for condition 3, but it needs to be larger for load condition 1 and 2.

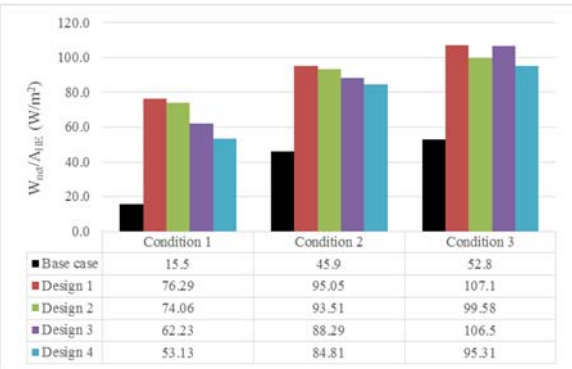


Figure 5: The ratio of Wnet/A_{HE} of four designs in comparison to the base case.

4.8. Waste heat utilization rate (UR)

The UR analysis is investigated to measure the capability of ORC design to recovery the waste heat. In other words, this figure measures how match the design to heat resource. The higher the UR level which is achieved, the better the match is between the design and the heat resource. As shown in Figure 6 that the ORC designs are more suitable for low Capstone gas turbine load conditions such as condition 1 and condition 2, because all designs under condition 3 have the lowest UR level (less than 50%) among other load conditions. The highest UR level is achieved by design 2 and design 1 under condition 1 at 76.90% and 73.83%, respectively.

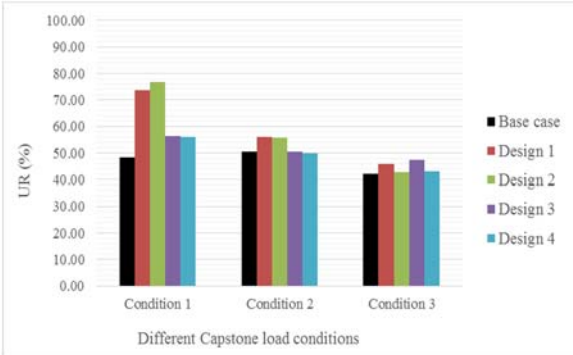


Figure 6: The UR of four designs in comparison to the base case.

4.9. Acceptable designs

The best modification of the existing ORC plant depends on the Capstone gas turbine load condition (the heat resource) and the number of modified exchangers in the system.

- Design 1 has the highest objective function in all design alternatives. This increases the objective function from 100% to 391% of the base case depending on the Capstone gas turbine load conditions. This design has a new design in all heat exchangers in the system.
- Design 2 is the best choice when the number of modified heat exchanger is limited to two units, the gas-oil HE and the evaporator need to be modified under condition 1 and 2.
- Design 3 is the best choice when the Capstone gas turbine runs in condition 3 because the design is able to produce the highest power output with comparable objective function to design 1 (Note that design 1 has the highest level of objective function in all design alternatives).
- Design 4 is the best choice when the Capstone gas turbine runs in condition 1 because the design is able to increase the power output and objective function from the base case at 96% and 242%, respectively. This design modifies one unit of the heat exchanger, which is the evaporator.

5. CONCLUSIONS

This chapter proposes a comprehensive methodology to design and optimize an ORC system based on DTR method for WHR applications. The design based on DTR method aims to develop a cost-effective design that is the best match to a heat resource. The methodology has been tested in a lab-scale ORC system. The design methodology is also valid for a larger-scale ORC system and other applications because all ORC system has the same principle.

NOMENCLATURE

A	Heat transfer area (m^2)
AU	Heat transfer conductance (W/K)
Bo	Boiling number (-)
C_p	Specific heat (J/kgK)
Co	Convection number (-)
Fr	Froude number (-)
H	Specific enthalpy (J/kg)
h	Heat transfer coefficient (W/m^2K)
HE	Heat exchanger
$\dot{M}$	Mass flow rate (kg/s)
$\dot{m}_{oil}$	Mass flow rate of oil (kg/s)
N	Rotating speed (RPM)
Out	Output
P	Pressure (kPa)
PP	Pinch Point (^{0}C)
Q	Total energy transfer by heat (J)
$r_{,v,in}$	Build-in volume ratio (-)
T	Temperature (^{0}C)
T_{loss}	Torque (Nm)
U	Overall heat transfer coefficient (W/m^2K)
v	Specific volume (m^3/kg)
$\dot{V}$	Volume flow rate (m^3/s)
W_{net}	Net electrical power output (W)
W_p	Power of pump (W)
W_T	Power of turbine (W)

Subcripts:

1,2,3,..	State point in the system
amb	Ambient condition
calc	Calculated data
cond	Condenser
ev	Evaporator
ex	Exhaust
In	Inlet
l	Liquid
max	Maximum
meas	Measured data
n	Number of main components
P	Pump
T	Turbine/ Expander
tp	Two-phase
r	Refrigerant
s	Isentropic, swept
sf	Secondary fluid
su	Supply
sh	Superheating (^{0}C)
tot	Total
v	Vapour
w	Wall

Greek Symbols:

Δ	Delta
Δp	Pressure drop (kPa)

Acronyms:

LMTD	Log-Mean Temperature Difference Method
ORC	Organic Rankine Cycle
WHR	Waste heat recovery

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