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Designing Optimized Organic Rankine Cycles Systems for Waste Heat-to-Power Conversion of Gas Turbine Flue Gases

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Abstract-
The focus of this work is to develop a flexible ORC design procedure that compares thermo-economic performances of simple and recuperative ORCs for both subcritical and supercritical cycles through a multi-objective optimization that relates the economic parameters to the network output for waste heat-to-power conversion of exhaust gases. Few researches have proposed rather simpler methods by modifications to the Jacob number (Ja) but these are insufficient to make technical and economic decisions on the subject matter as Ja is only appropriate for comparing performances of different working fluids at the same operating condition. Coupled with the barrier on the minimum temperature the exhaust gases from power plants may attain, the use of Ja as the only criteria is not sufficient for varying operating condition presented by Gas Turbine power plants. Hence, this review presents follow-through numerical methodology for designing adapted ORC for waste heat-to-power energy conversion.

Key words: Organic Rankine Cycle, Jacob Number, Supercritical Cycle, Low grade heat, Heat Exchanger

1 Introduction

Despite the strife towards the full exploitation of renewable energy (RE) sources to provide the bulk of the global energy demand, the world consumption of fossil fuels and its associated hazards have been steadily increasing over the years. Whereas, more than 60% of the energy content of the annual consumption of the fossil fuels is wasted away as low-grade heat. Hence the excessive depletion of the limited resources and the accompanying environmental hazards can thus be controlled to a sustainable level relative to how much of the waste low-grade heat that can be further utilized to meet part of the energy demand while on the other hand, further improving the efficiencies of thermal conversion systems.

Organic Rankine cycles (ORCs) is acknowledged to be a viable sustainable technology suitable for utilizing low-grade energies because of its high versatility. Though, its practical applications, cost effectiveness and thermal conversion efficiencies depend on a number of design criteria to various degrees, the optimum ORC design parameters must be selected and thoroughly investigated in order to adapt it to the thermal regime of the applicable operating condition. The essential considerations dependent on the appropriate performance indicators include the cycle architecture, best appropriate working fluid (pure or mixture), the temperatures of the vapour generator and condenser, the pinch point temperature differences (PPTD), the effectiveness of the heat exchangers, degree of superheating of the working fluid, isentropic

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efficiencies of the pump and turbine, the selection and sizing of the basic components of the ORC, and most importantly, cost and the environmental compatibility.

As with practical steam power plants, there are a number of possible modifications to the simple ORC by which the system can be adapted to the uniqueness of each waste heat stream [1]. The rational modifications to the practical cycle design are however limited. Reheat and turbine bleeding are essentially inappropriate, especially for ORC systems meant for waste heat recovery (WHR), but it is advisable to install a recuperator between the outlets of the feed pump outlet and the turbine to preheat the liquid as depicted in Fig.1 [2]. More often the ORC is operated at the subcritical level but as well operate at supercritical level with similar modifications.

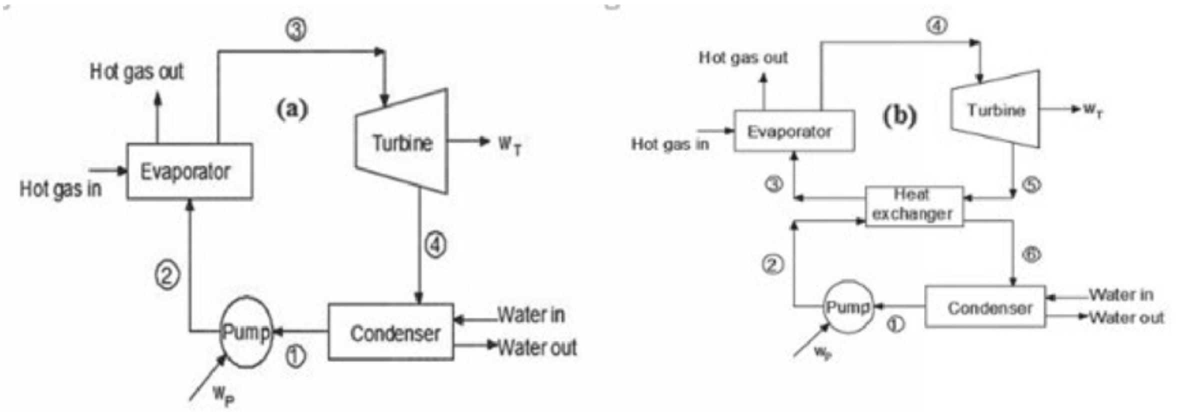


Figure 1: Basic ORC (a) and ORC with internal heat exchanger (b)

With the requisite selection of the best appropriate primary working fluid(s) from more about 50 commonly employed pure working fluids and their optimal composition of zeotropic mixtures [3] alongside the best cycle architecture adapted to the thermal regimes of the applicable operating conditions, ORC system design procedures are rather complex and hence the most researched sustainable technology partly because of its applications to renewable energy and most especially because of its capability to control the hazards of excessive resource depletion and environmental pollutions associated with the exploitation of fossil fuels to a sustainable level.

Much work has hitherto been done on the ORC technology for both subcritical and supercritical cycles but the core challenges in its design and development still remain the selection of the best thermos-economic cycle architecture employing the most suitable working fluid that can realize the optimum objective function. Whereas the choices depend on the selected performance indicators and as well, the optimum operation parameters are different for different indicators [4]. Several performance indices however exist in the study ORCs thermodynamics, economic and environmental performances. Bademlioglu et al [5] employed Taguchi and ANOVA approaches to determine the most effective parameters using thermal efficiency as the objective function.

They however discovered that the three parameters that significantly influence the performance of the ORC are the temperature of the vapour generator, the condenser temperature and turbine efficiency which jointly accounts for 70% and 82.5% of the total effect with the Taguchi and

ANOVA methods respectively while the least effective factors were observed to be the PPTD of the vapour generator, the PPTD of the condenser and feed pump efficiency with their combined influence below 1% in both investigation techniques. Essentially, the exclusive effect of the choice of working fluid on the system efficiency was discovered to be 10.8% and 5.8% by Taguchi and 5.8% ANOVA respectively.

Hence, the objective functions to be optimized vary subject to the target application. Several researches have hence recommended the net power output, W_{net}, is the most appropriate performance index in a multiparameter optimization for waste heat-to-power applications because it is crucial for maximizing heat retrieval from waste heat in practice. The net power output from ORCs is essentially influenced by the operating variable conditions, specifically the vapour generator pressure P_{evap}, the condensation pressure P_{cond}, the degree of superheat, DSH, and the amount of sub-cooling, SC [6]. The pressure variable parameters can thus be simultaneously optimized by merely optimizing the mass flow rate of the organic fluids to achieve the best net power output or heat recovery efficiency using pre-selected working fluids [7].

Wang et al. [8], Javanshir et al. [9], [10] etc. have presented a much simpler alternative to the exhaustive parametric multi-objective optimization through modifications to a thermodynamic index called the Jacob number (Ja). Though their works yielded close results with the conventional numerical analysis but none of the modifications so far compared the economic and cycle performances of supercritical and regenerative cycles. Additionally, Ja on its own is not a suitable selection criteria to select a working fluid for WHR from the flue gas of Gas power plants because of two reasons namely (i) varying power plant loading as a result of changing operation condition and (ii) the limit of 90ºC minimum temperature the exhaust gas can attain [11].

Since there is no working fluid that can be labelled as perfect or optimal, ORCs system design has to be carried out with a number of pre-selected potential working fluids based on selection criteria discussed in section 2.1. To carry out this review, the development, modeling of large capacity ORC systems and the essential performance indices are discussed in section 2 while the optimization routine is discussed in section 3. Section 4 concludes the review.

2 Methodology

Development of Waste Heat-to-Power Systems Using ORC

The procedure for designing waste heat-to-power using ORC essentially is made up of two vital steps iterated in numerical models. The first is to identify the operating condition for the ORC by determining the optimal temperature and heat rate from the waste heat. This is a key to selecting the potential working fluids, evaporative method and the cycle architectures for the operating condition. This process requires the development of accurate thermodynamic models and running numerical computations. Determining the optimal working fluid for the regime requires a concurrent two-side approach which relates the thermodynamic and heat transfer properties along with safety and environmental factors.

The second step involves specifying standard equipment sizes for the ORC through thermo-economic optimization of performance and objective functions by way of evaluation of thermodynamic and/or empirical models simultaneously with its economic and financial viability. The two basic steps are interrelated because the process is a highly iterative process and any change in one parameter requires reconsideration of the entire procedure. Invariably, the efficiency of the ORC for a particular operating state, the lifespan of the turboexpander, the size of the heat exchangers, and the feed pump work, and ultimately, the cost of the equipment all depends to a very large extent, on the chosen working fluid and the cycle architecture [12].

An optimization of the global thermo-economic model procedure is hence appropriate for deciding on the most suitable working fluid concurrently with the most appropriate ORC system architecture using the radial type of turbine. The preselection of potential working fluid can however be achieved by the operating map methodology presented by Quoilin et al. [13] as shown in Fig.2 below. This would provide a list of the few working fluids with acceptable efficiencies and component sizes within the operating condition.

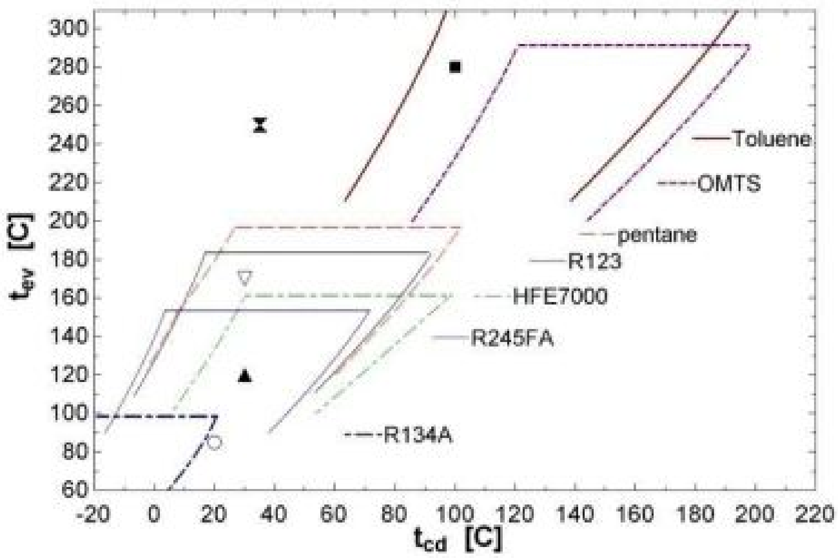


Figure 2: Radial inflow turbine operating map [13]

However, because of the overlapping as shown above in addition to the fact that the operating map approach does not take account of the cost implication of the turbine and other major components, the final choice of the most appropriate working fluid is arrived at by thermo-economic optimization which assigns a cost function to each component for a multi-parameter optimization through a rigorous computational procedure by means of algorithms.

Modeling High Capacity ORCs Systems for Waste Heat-to-Power Applications

Tabor et al. [13] opposed using simple equations of state (EoS) in computing the thermodynamic properties of organic fluids because it showed insufficient accuracy. They recommended the use of thermo-physical and transport properties databases linked to a numerical solver to implement the simulation of both the steady-state and dynamic ORC cycle architectures. According to them, steady-state models are primarily meant for design/component sizing and as well for part-load simulation while dynamic models additionally computes the energy and mass balance in each component, and are hence essentially suitable for designing the control systems for the transients periods or start-up/stop.

The most common steady-state tools are Engineering Equation solver (EES), MATLAB, Scilab, and CycleTempo, while Modelica, and Simulink are the most commonly applied dynamic modeling tools. Thermo-physical & transport properties of the working fluids can be furnished for the numerical solvers by CoolProp, RefProp, FluidProp, or AsenProp. Nonetheless, it is noteworthy that all the various tools available for ORC simulations have their different advantages and drawbacks. Several researches have established the key parameter to maximize for WHR applications is the net output power while minimizing the flow rate for the compactness of the unit. Kai et al. [14] investigated the effect of evaporation pressure, amount of superheat, and the minimum allowable temperature in the vapour generator of an ORC geothermal system. Results indicated that the effect of pinch temperature in the vapour generator on the power developed is much more pronounced than superheating.

Badescu et al. [15]on the other hand established that there is an optimum evaporation pressure that makes the net power attains the highest possible value, both of which are a function of degree of superheat when utilizing isentropic fluids but dry fluids require no superheating. The optimum degree of superheat was said to range between 30 and 40ºC for isentropic working fluids depending on the type of at both full and partial loads design conditions. It has likewise been established that the use of a recuperator or regeneration with turbine bleeding however reduces the amount of heat that can be recovered by the cycle from the heat source [16].

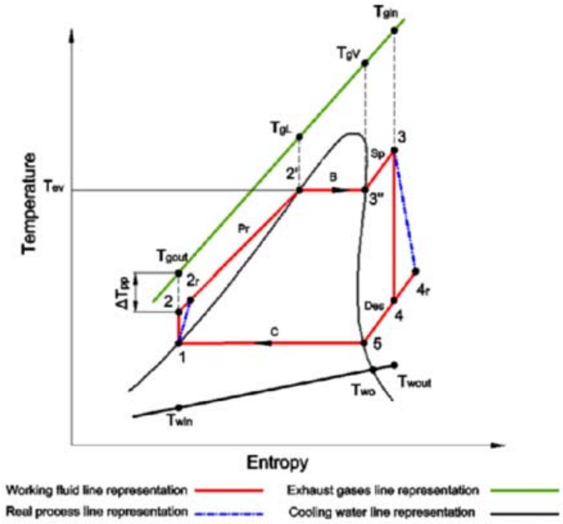
However, the recuperator contributes more than improving the cycle efficiency. It also retains the minimum fluid temperature required by the heat recovery vapor generator in order to prevent corrosion as a result of condensation of sulfur components in the exhaust gases [17], as well as significantly decreasing the condenser load [16]. Hence, it is of benefit to consider the optimal amount of superheat as well as recuperation for high capacity ORC systems design analysis when using isentropic fluids. The use of recuperator is however more essential when using dry fluids because dry fluids exiting from the turbine are still superheated and may hence be used to raise the temperature of the fluid flowing into the vapour generator, hence reducing the temperature to the condensing temperature corresponding to the two-phase state [15].

Peris et al. [19] worked on increasing the power output from a waste heat by simulating six ORC architectures using ten working fluids. The performances selected were cycle efficiency, safety, cost and environmental effects. Results indicated that little enhancement to the simple ORC delivered comparable improvements with the most intricate system configurations because of the small differential temperature between the exit of the turbine and the inlet of the condenser. Nonetheless, it is extremely important to factor in acceptable off-design limits from the design stage to deal with unavoidable transient conditions that may arise especially from sudden changes in energy demand or quantity of heat energy supply so as to maintain the operating condition of the system within reasonable range [20]. They in essence hence proposed developing appropriate models to simulate the steady state or the transient performance of the ORC consequently to examine the effect working constraints and as well evaluate the critical working state. Badescu et al. [15] hence proposed the optimization procedures for generator loads of100%, 75% and 50%. The following technique is hence projected for the design of ORC power units as discussed below:

Specification of the thermal source
The first step is to estimate the amount of power and temperature of the thermal source and the choice of a suitable cooling medium, water or air. Preferably, the condenser is air-cooled, having relatively high cycle efficiency and yet avoiding the requisite water treatment system for water cooling. Lion et al. [21] however employed 1:1 ratio mixture of water and ethylene-glycol as the coolant in order to avoid boiling in the condenser.
Calculation of thermodynamic properties and flow rates
It involves the development of models that capture the mass and energy balance for all the cycle modules and cycle configuration as a whole under the assumption of steady state setup. Beginning from the defined parameters of the low-grade heat input- power Q_{in} and inlet temperature T_{in} (step 1), and a preselected working fluid (step 2), a discretionary degree of superheating is defined and hence pinch point analysis of the vapour generator is employed to determine the inlet temperature of the turboexpander, TIT, using the efficiency of the vapour generator which in return indicates the evaporation pressure, P_{eva}.

Similarly, starting from the temperature of the cooling fluid, the condenser pressure and temperature are determined by defining reasonable condenser pinch point. Implementing steps 1 to 3 defines the thermodynamic states at exits of the condenser, vapour generator and the recuperator and hence the mass flow rate. The parameters at pump outlet can be obtained by assuming an isochoric process from P_{con} to P_{eva} with reasonable pump efficiency. The supposed efficiencies of the turbine and the recuperator factor into their preliminary exit parameters but their values can thereafter be corrected at the end of the 4th step. Based on a typical T-s diagram in Fig.3 below, the following constraints are established in literature for ORC systems employed for waste heat-to-power conversion from exhaust gases:

i. In order to account for errors associated with assumptions and also to keep the design within practically achievable limits, pressure boundary conditions for subcritical cycle is $0.01P_{crit} < P < 0.81P_{crit}$ [22], where P_{crit} is the critical pressure of the organic substance. However, for supercritcal cycles, the upper pressure of the cycle is fixed at $1.12P_c$ in order to maintain safe operation [23].
ii. In order to avoid subatmospheric operation of the system, for state 1, condensing pressure, $P_1 = P_2 \geq 1.2\ bar\ or\ 120kN/m^2$ and $T_1 \geq 50°C$.
iii. For safe operation of the subcritical cycle, the maximum allowable pressure is limited to 30 bar. To avoid the condensation of the exhaust gases, the exit temperature T_{gout} of the flue gases should not be less than 90°C to avoid condensation of toxic gases and low temperature corrosion. Hence, for state 2, $P_2 = P_{2r} = P_{2'} = P_{3''} = P_3 = 0.81 \times P_c \leq 30\ bar\ or\ 3000kN/m^2$ and $T_2 \geq 80°C$.
iv. Minimum pinch point temperature difference (ΔT_{pp}) between T_{gout} and T_2 = 10°C, hence, $T_2 \geq 80°C$ in order to maintain the exit temperature, $T_e \geq 90°C$.
v. Minimum PPTD (ΔT_{pp}) between T_{gin} and T_3 = 15°C
vi. For state 3, maximum inlet temperature of the expander, $T_3 \leq T_C - 10°C$
vii. The upper bound of superheating is 40ºC [15], [24].



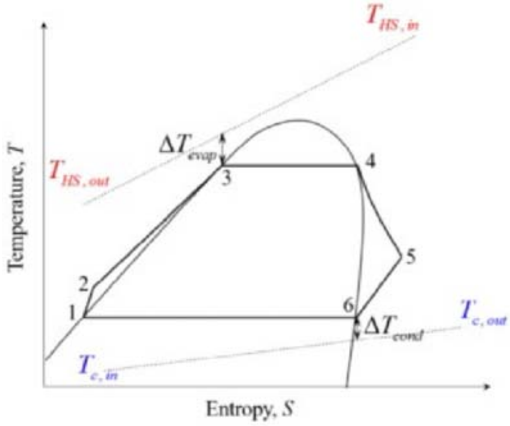


Figure 3: Typical Temperature Profile for (a) subcritical cycle and (b) Supercritical cycle

Formulation of Governing Equations

As depicted in figure 3, the ORC follows similar thermodynamic principles as the conventional steam Rankine cycle except that the ORC utilizes low-boiling point organic fluids (pure or azeotropic mixtures) to recuperate waste heat. The four basic components are the condenser, the feed pump, the heat recovery unit, and the expander but may as well employ a recuperator in the modified structure. The heat recovery unit conceptually consists of three sections namely Preheater or Economizer (section 1), Boiler or Evaporator (section 2), and the superheater (section 3) which are connected in series with a fixed total area but variable section areas for the three sections depending on the operating regime (Fig. 6). The condenser as well consists of two sections namely the desuperheater and the condensation sections. The heat recovery unit and the condenser are of the shell and tube form of heat exchanger sectioned into single- and double phase flow sections. The organic fluid flows inside the small-diameter tubes exchanging heat with the exhaust gas streaming on the outside in the heat recovery unit while the coolant flows in the outside tube of the condenser. A characteristic temperature profile for the subcritical and supercritical organic Rankine cycles is shown in Figure 4.

Heat Transfer Model

The vapour generator or heat recovery unit is abstractly sectioned into three, viz. preheater, boiler and superheater, respectively, serially connected (Fig. 4). Though the surface area of the vapour generator is fixed but that of each section can be varied based on the operating condition.

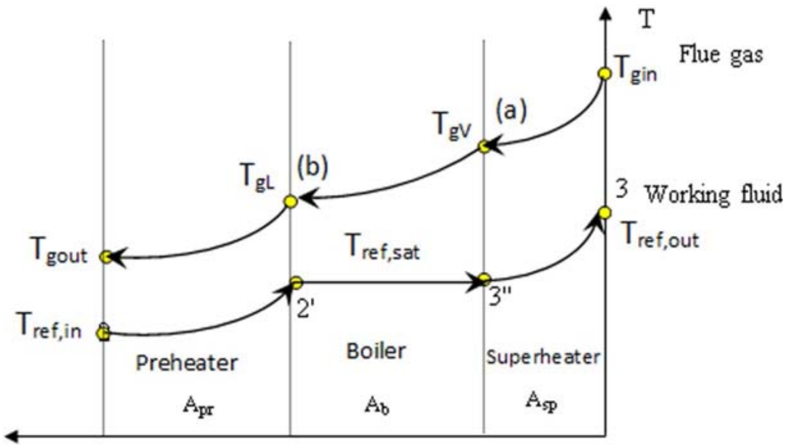


Figure 4: Three zones of the vapour generator (preheater, boiler and superheater)

Properties of the Flue Gas

The fuel determines the composition of the exhaust gases. The main constituents of the flue gas resulting from combustion of fossil fuels driving electric generators are CO_2, H_2O, N_2, and O_2. A typical gas composition generated from natural gas combustion given by Sung et al. [25] is 72.55% N, 12.34% O_2, 3.72% CO_2, 10.52% H_2O, and 0.87% Ar while that adopted by Badescu et al. [15] for combusting diesel fuel is g CO_2 = 9.1%, g H_2O = 7%4, g O_2 = 9.3%, g N_2 = 74.2%. However, the mass fractions of these constituents vary depending on operating regime subject to whether it bears full or partial load. The enthalpy of the mixture of the exhaust gases, assuming ideal gas behaviour, can hence be obtained as the weighted sum of each constituent of the mixture [26], expressed as:

$$h_{gmix} = \sum_{n=1}^{5} mf_n \, . \, h_n \quad (1)$$

where mf_n is the mass fraction in % of the each of the five constituents.

The temperature at each section of the vapour generator determines the properties and the Prandtl number at each section. Such essential properties include the conductivity, specific heat, and dynamic viscosity. As shown in Fig. 4, the temperature of gases in the vapour generator varieties from T_{gin} of the turbine exhaust gas at the inlet to T_{gout} at the outlet of the vapour generator. Hence, the effective temperature in each section of the vapour generator is the average in each section.

Properties of the Cooling Water

Water is the cooling medium usually adopted for high capacity ORC systems though may also be air-cooled. The condenser comprises of two sections- the desuperheater and the condensation sections and is water cooled. As applicable in the vapour generator, the temperature of the coolant determines the conductivity, heat capacity, dynamic viscosity, and Prandtl number at each section of the condenser. The temperature of the cooling fluid in the desuperheater section T_{des} varies from T_{wo} corresponding to the saturation temperature of the working fluid and T_{wout} of the water at the outlet. The effective temperature of cooling water T_{des} at the desuperheater section is likewise given by the average temperature in each section. In similar manner, effective temperature in the condensation zone is the average between T_{win} at the inlet andT_{wo}.

Thermodynamic Models

The thermodynamic models of the pump, vapour generator, turboexpander, condenser and the recuperator developed according to the cycle structure. Since the turboexpander is the most vital unit which has most significant influence on the performance and efficiency of the ORC system, a detailed expander model is essential.

Heat Exchanger Analysis

Counter current heat exchangers (HE) are modeled either by the Effectiveness-NTU or the log mean temperature difference (LMTD) method but the LMTD method is more appropriate and often employed for sectioned areas of heat exchangers [27]. For the thermal capacities at each section of the HE to be suitable for the thermal capacities required by the process conditions, defined by the temperature and flow rate at each zone, the corresponding heat transfer areas must be sufficient.

Heat Transfer Analysis for each Section

The governing equations for the heat recovery unit and condenser can be analyzed the same way except that the phenomenon of heat transfers in boiling and condensation are not similar and hence, require different and appropriate correlations [28]. The vapour generator comprises of a countercurrent heat exchanger conceptually divided into three sections. Evaluation of the heat transfer coefficient is hence divided into three representing the three processes- superheating, boiling and preheating. The organic fluid flows inside the tube while the flue gas flows outside in all thorough the sections as represented in Fig. 5.

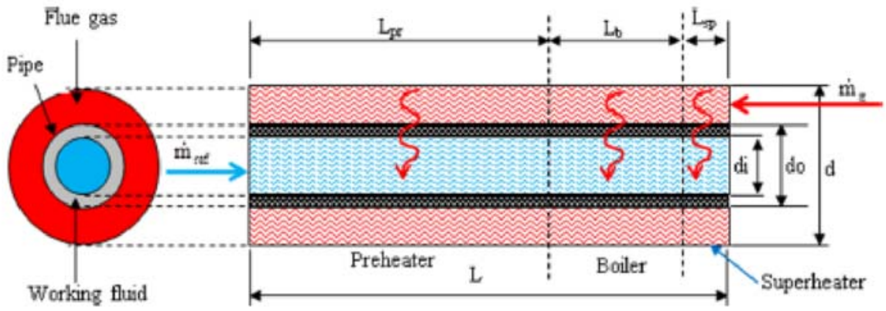


Figure 5: Geometry of the Vapour Generator

As with the vapour generator, the condenser too is sectioned but may compose of only two sections- the desuperheater and the condensation sections [29]. Likewise, the evaluation of the heat transfer coefficient has two parts comprising of the desuperheating and condensation processes. Fig. 6 shows the geometry and flow pattern in the condenser.

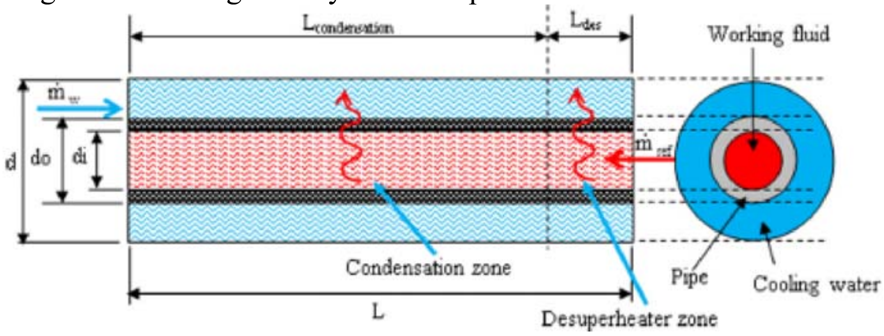


Figure 6: Geometry of the Condenser

Appropriate correlations to be employed in the processes taking place in sequence at each section of these heat exchange devises are provided in references [15], [30].

Performance Indices of the ORC Unit

Since the all heat in the exhaust gas cannot be fully regained via the ORC technology, several indicators have been employed to assess the various performances of the ORC. These essential ones include:

Heat Recovery Efficiency

An efficiency index for the heat recuperation by the HRU as cited to be proposed by Naijun et al. [31] ., (2013) and employed by [15] is the heat recovery efficiency, expressed as:

$$\eta_r = \frac{Q_{gases}}{Q_{max}} \quad (2)$$

where Q_{max} is given by :

$$Q_{max} = \dot{m}_g C_{P,g}\left[T_{gin} - (T_2 + \Delta T_{amb})\right] \quad (3)$$

$\dot{m}_g$ and $C_{P,g}$ are respectively mass flow rate and specific heat capacity at constant pressure of the exhaust gas. $(T_2 + \Delta T_{amb})$ represents the minimum possible temperature the exhaust gas temperature at the outlet of the HRU can attain which is 90ºC.

Net Power Output per Heat Transfer Area, ε

This index as proposed by Li et al. [10] is an indicator of the cost-effective performance of the ORC system. It is expressed as:

$$\varepsilon = {W_{net}}/{A_T} \quad (4)$$

Back Work Ratio (BWR)

This is an index that presents the proportion of the power delivered by the turboexpander that is utilized by the feed pump. The BWR is a good indicator of the ORC sensitivity to the pump efficiency and it is highly dependent on the critical temperature of the organic fluid favoured by being a little lower than the critical temperature [26]. The BWR has been found to increase with temperature and greater for fluids with lower critical temperatures [32]. The BWR is expressed as:

$$BWR = \frac{W_P}{W_T} \quad (5)$$

Thermal Efficiency

This is the fraction of the net power output of the ORC to the available heat flux in the exhaust gas, is given as:

$$\eta_{th} = \frac{W_n}{Q_{source}} = \frac{W_{exp,actual} - W_{p,actual}}{Q_{source}} \quad (6)$$

It is however essential to apply cost correlations such as given by Lecompte et al. [33] using recent data from a statistical review of producers of equipment to perform cost estimates with accuracies established to be in the range of +40% to -25%. This will allow analysis of economic parameters such as specific investment cost, payback period, levelized cost of electricity, etc.

3 Discussions

Optimization with Generic Algorithms

Along with the constraints on established in section 3.5, the temperature of the coolant at the condenser outlet should be below the boiling point of the coolant and the vapour quality exiting from the expander should be higher than 90% in order to avoid forming of liquid droplets which is damaging to turboexpanders. Relevant variable parameters for the simple and modified ORCs can be performed with multi objective algorithm when the thermodynamic and heat transfer equations have been appropriately coded in a numerical solver such as EES, MATLAB, Scilab, etc. An optimal temperature exists for the vapour generator operating a subcritical cycle when:

(i) the temperature of the flues gas is less than the critical temperature of the pre-selected organic fluid and,
(ii) (ii) the critical temperature of the potential organic fluid is less than the temperature of the flue gas by 13 to 23ºC when the PPTD of the vapour generator is fixed at 5ºC [34].

The net power output of the ORC architecture and the overall expander efficiency are maximized while the fluid mass flow rate is minimized to in order to reduce the size of the equipment.

4 Conclusion

In this paper, a review of studies on designing ORC systems waste heat-to-power conversion of the exhaust gases from turbine power plants carried out. Maximizing the net-power output for highest obtainable power from the working fluids and minimizing the fluid mass flow rate so as to minimize the size and reduce cost of equipment are the primary objective of the design. Design constraints and considerations have been considered and presented from various authors. Hence, a follow-through knowledge and procedure for feasibility of designing ORC system suitable for the operating regime of a waste heat source has been established in this study.

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