APPROXIMATE ANALYSIS OF THE ECONOMIC ADVANTAGE OF A DUAL SOURCE ORC SYSTEM OVER TWO SINGLE ORC SYTEMS IN THE CONVERSION OF DUAL LOW AND MID GRADE HEAT ENERGY TO ELECTRICITY

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Abstract

An approximate economic analysis reported in this paper shows that in a novel dual source ORC system where a low-grade heat source is used to preheat and vapourise the working fluid and mid-grade heat source is used to provide the superheating, the payback period is very much smaller than when the two heat sources are used separately to drive a single ORC system.

The simulation results shows that on the same waste heat basis, the size of the heat exchangers and the quantity of working fluid used for the dual source ORC is smaller than the cumulative heat exchangers size and refrigerant quantity used for the two single ORC systems.

Keywords: Organic Rankine Cycle (ORC), Dual Heat Source, Mid-grade, Low-grade, Economic

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1 Introduction

Energy efficiency through waste heat recovery has been identified as one of the major ways of reducing CO_2 emission and its associated environmental impact. Waste heat recovery processes involve the conversion of waste low grade heat sources into useful energy. There are different approaches used to recover waste heats. Any approach adopted for any given process depends on the use of the recovered waste heat which includes electricity generation, space heating and cooling, heat integration and so on. Based on the uses listed above, the different waste heat recovery approaches likely to be adopted include Binary Power Cycles (Organic Rankine Cycle (ORC) or Kalina Cycle), Absorption Chillers Cycles and Heat Exchanger Network Systems. This paper concentrates on the use of ORC for electricity generation from dual source low and mid grade waste heat.

ORC technology is not a new technology as it evolved from Rankine Cycle (RC) technology. The only difference between the two cycles is the nature of the working fluid used. The former makes use of organic working fluids while the later uses water/steam as the working fluid. The idea of utilising organic working fluid was borne as far back as 1823 [1]. Although the cost of conventional Steam RC seems to be lower than that of the ORC [2], the ability of the later to utilise low temperature waste heat sources makes it a better alternative.

A lot of simulation based reports had been presented in the literature on the ORC systems using different heat sources. Hettiarachchi et al [2] presented their work on the optimum design criteria for an ORC cycle and concluded that the working fluid has a significant impact on the cost of the ORC plant. Kosmadakis et al [3] designed a two stage ORC system for reverse osmosis desalination and concluded that a two stage ORC system can efficiently be used to recuperate heat and produce fresh water. Doty and Shevgoor [4] showed how to improve the efficiency of an ORC system using a dual heat source system. Saleh et al [5] conducted a work on the working fluids for low-temperature ORC systems and found that isentropic organic fluids show little increase in thermal efficiency when superheated while for dry organic fluids, superheating tends to lower the thermal efficiency. They also concluded that the thermal efficiency of an ORC system using an isentropic organic fluid can be significantly improved by combining superheating with a regenerator.

Many low grade ORC systems presented in the literature operate as a single ORC power plant while the mid-grade heat sources are usually utilised for cascade ORC power plants in which the condenser of the top cycle also serves as the evaporator of the bottom cycle. However, not much work had been done on using a dual low- and mid-grade heat sources to power a single ORC system and this is the issue which this report tries to investigate.

In this report, the design and economic advantage of using a single ORC system with dual heat sources at different temperatures as against two single ORC systems each operating from its own heat source is investigated.

The motivation to carry out this investigation occurs as a result of the fact that in a process plant sited very close to a low temperature geothermal ORC power plant, there may be mid-temperature waste heat sources from the plant which should be integrated into the geothermal ORC power plant instead of developing another single ORC power plant to capture the waste heat from the process plant or in a location where there are many sources of waste heats at different temperatures, it may be more economical to design a multiple heat source ORC system order than to design a single ORC system for each waste heat source.

The models used in this investigation are designed and optimised using IPSEpro process simulation software version 4.0 [6]. The properties of the working fluid (see Table I) are calculated using NIST Standard Reference Database 23, REFPROP version 7.0 [7].

Table I: Thermodynamic Properties of R134a (CF_3CH_2F – 1,1,1,2 – tetraflouroethane (CAS# 811-97-2))

Molar Mass (kg/mol)	Triple Pt. Temp (°C)	Normal B.P. (°C)	Critical Points			Range of Applicability			
			Temp (°C)	Pressure (MPa)	Density (kg/m^3)	Min Temp (°C)	Max Temp (°C)	Max Pressure (MPa)	Max Density (kg/m^3)
102.03	-103.30	-26.074	101.06	4.0593	511.90	-103.30	181.85	70	1591.7

2 ORC Cycles Description

2.1 Single Heat Source ORC System

Figure 1 shows the conventional ORC flow diagram. The organic working fluid contained in the receiver tank is pumped to an evaporator where it is vapourised using the available heat source (either from geothermal, furnaces, industrial waste heat, and so on). The vapourised working fluid is passed to the expander/turbine where it is expanded to provide mechanical energy which turns the shaft connected to the generator to produce electricity.

The low pressure vapour at the exit of the expander is passed to the condenser where it is condensed back to liquid phase and sent back to the receiver tank, then to the pump and the process continues again.

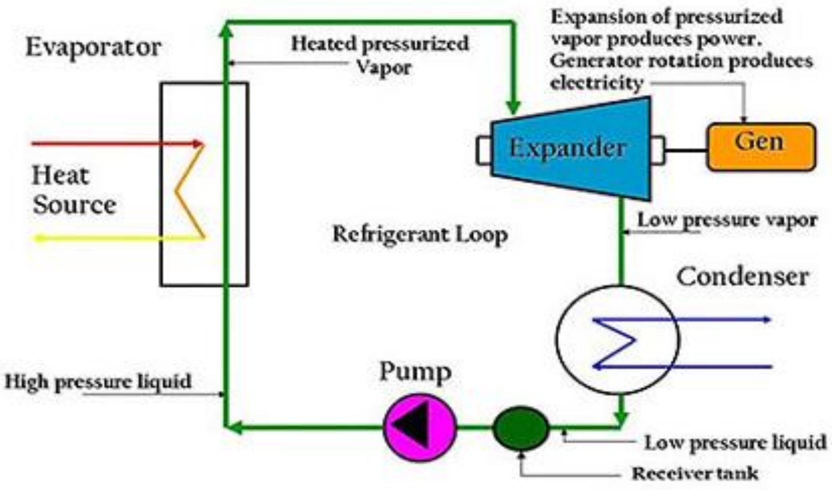

Figure 1: Single Heat Source ORC System

2.2 Dual Heat Source ORC System

The cycle is similar to the single heat source ORC system except that two heat sources (low-and – mid grade) are used to drive the cycle as shown in Fig. 2. The low temperature heat source serves as a primary cycle while the mid-temperature heat source comes on whenever there is a mid-grade heat supply available to the system. In the absence of a mid-grade heat supply, the working fluid is automatically passed through the bypass loop to complete the cycle. The low temperature heat source is used to pre-heat/vapourise the working fluid while the mid temperature heat source is used to superheat the working fluid before it is sent to the expander. The cycle continues as explained in section 2.1 for the single heat source ORC system.

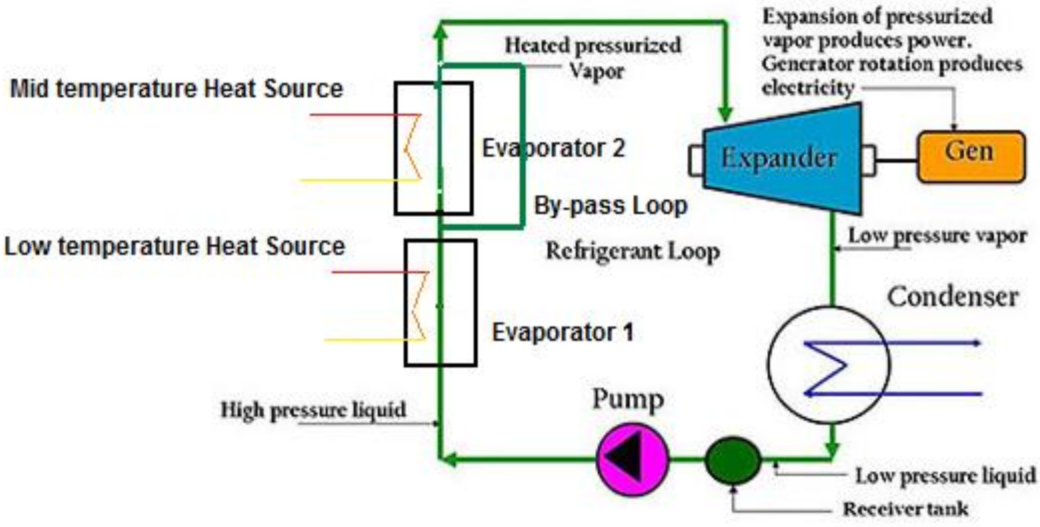

Figure 2: Dual Heat Source ORC System

3.0 Modelling & Simulation of the Single and Dual Heat Source ORC Systems

Since the objective of any power plant project is to generate as much power as possible from any given resources, all the ORC system presented in this paper are modelled in IPSEpro for maximum power generation.

3.1 Single ORC Using Low-Grade Heat Source

The heat source is modelled as hot water from a geothermal source at a temperature of 73.33 ^{o}C. Assuming the heat source is to be used downstream for another operation, it means that the exit temperature of the heat source from the evaporator must be fixed. Furthermore, the exit cooling water temperature is normally regulated because it is usually discharged to the environment. Table II shows the operating conditions used for modelling the single ORC system powered by low-grade heat source.

Figure 3 shows the simulated optimised IPSEpro model of the single ORC System powered by the low-grade heat source. The thermodynamic cycle undergone by the R134a working fluid during the ORC process is presented in Figure 4 and explained in Table III. The simulation result for the single ORC system using low grade heat source is shown in Table IV.

Table II: Operating Conditions for the Single ORC System using Low-Grade Heat Source

Parameter	Nominal Value
Low-Grade Heat Source Temperature (°C)	73.33
Low-Grade Heat Source Exit Temperature (°C)	50.00
Low-Grade Heat Source Mass Flowrate (kg/s)	33.39
Cooling Water Source Temperature (°C)	4.44
Cooling Water Source Exit Temperature (°C)	10.00

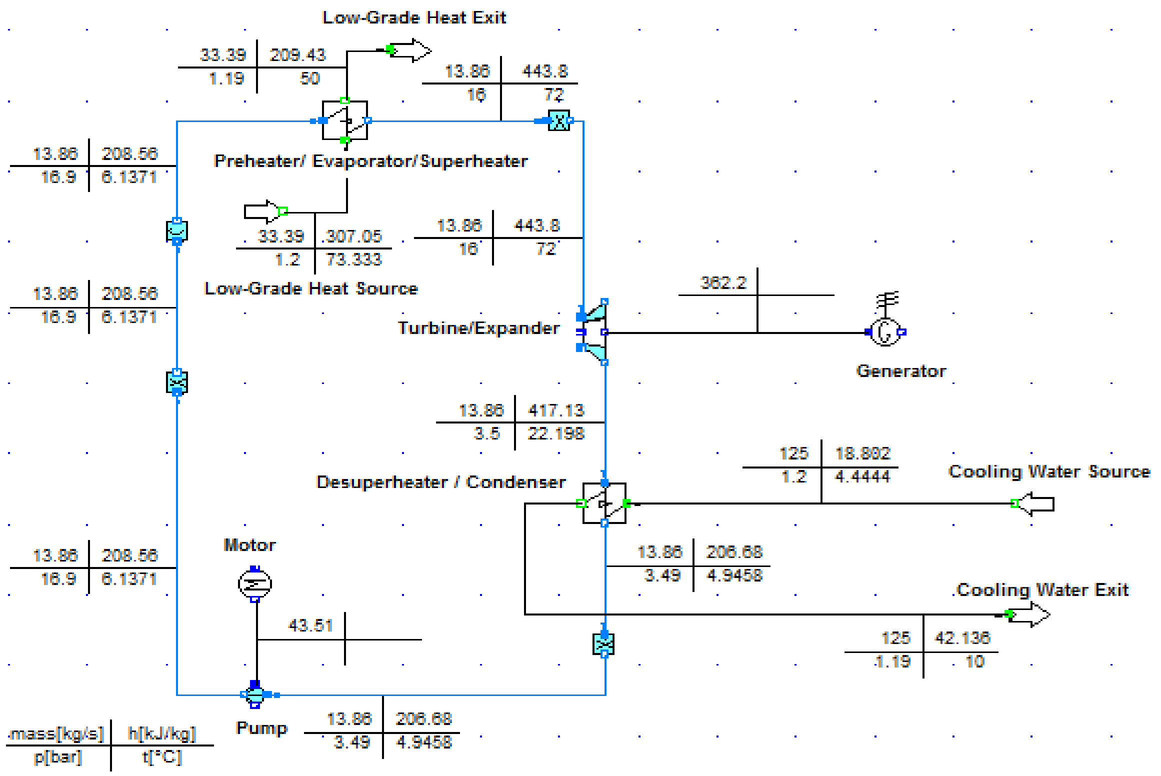

Figure 3: Simulated Optimised IPSEpro Model of the Single ORC System powered by Low-Grade Heat Source

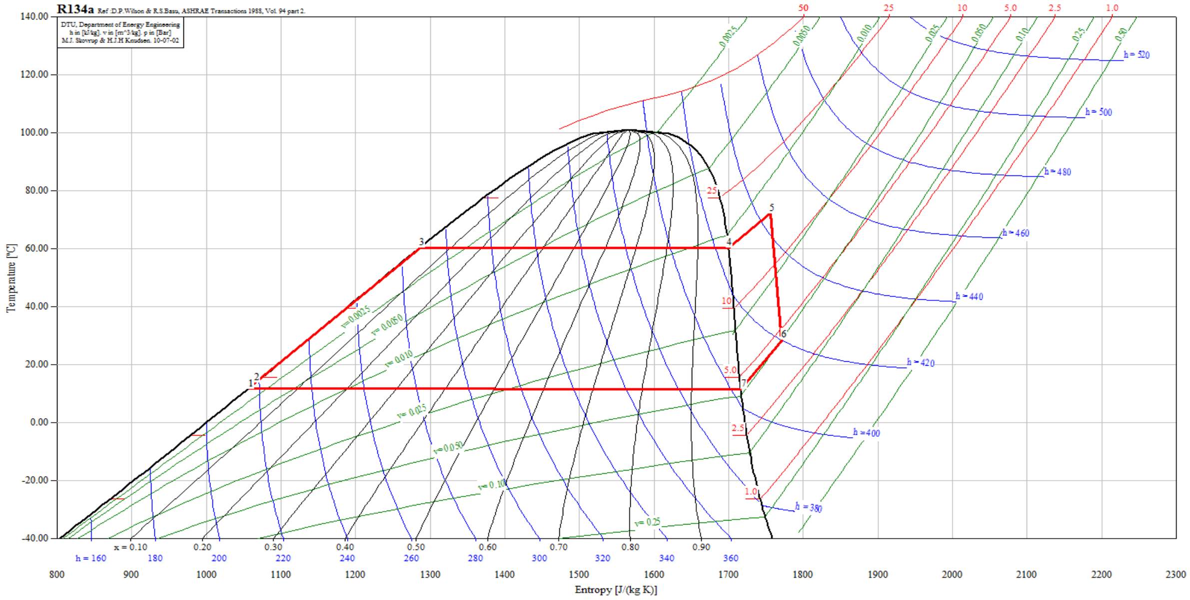

Figure 4: T-s diagram of R134a showing the Thermodynamic Cycle for a Low-Grade Heat Source

Table III: Thermodynamic Processes Exhibited by the R143a Working Fluid during the ORC Process

States	Process
1 – 2	The liquid working fluid is pumped to the pre-heater/evaporator/super-heater unit.
2 – 3	The working fluid is pre-heated in the pre-heater section at constant pressure using the heat from the low grade heat source.
3 – 4	The working fluid is evaporated at constant temperature and pressure using the heat from the low grade heat source.
4 – 5	The working fluid is superheated using the low grade heat source.
5 – 6	The super-heated vapour is expanded in the turbine to generate work.
6 – 7	The exit vapour from the turbine is de-superheated in the condenser.
7 – 1	The de-superheated working fluid is condensed completely to liquid to complete the cycle.

Table IV: Simulation Result for Single ORC using Low-Grade Heat Source

Parameter	IPSEpro Simulation Result
Low-Grade Heat Source Temperature (^{o}C)	73.33** (Nominal)
Low-Grade Heat Source Exit Temperature (^{o}C)	50.00** (Nominal)
Cooling Water Source Temperature (^{o}C)	4.44** (Nominal)
Cooling Water Exit Temperature (^{o}C)	10.00** (Nominal)
Low-Grade Heat Source Water Mass Flowrate (kg/s)	33.39** (Nominal)
Cooling Water Mass Flowrate (kg/s)	124.99*
R134a Working Fluid Mass Flowrate (kg/s)	13.86*
Turbine Efficiency (%)	80.00**
Turbine Inlet Pressure (bar)	16.00**
Turbine Outlet Pressure (bar)	3.50**
Pump Power (kW)	46.74*
Gross Generator Power (kW)	347.88*
Net Plant Power (kW)	301.11*
Thermal Efficiency (%)	9.20*
Evaporator UA value (kW/K)	267.79*
Condenser UA Value (kW/K)	795.90*
Evaporator Heat Transfer (kW_{th})	3260.16*
Condenser Heat Transfer (kW_{th})	2916.65*

* **calculated Variables** ** **Set Variables**

3.2 Single ORC Using Mid-Grade Heat Source

The single ORC system using mid-grade heat source is modelled similar to the low-grade heat source except that the heat source is modelled as a steam at a temperature of 125 ^{o}C and pressure of 1.2 bar flowing at 0.39 kg/s. The operating conditions used in the model are as shown in Table V.

Table V: Operating Conditions for the Single ORC System using Mid-Grade Heat Source

Parameter	Nominal Value
Mid-Grade Heat Source Temperature (°C)	125.00
Mid-Grade Heat Exit Temperature (°C)	50.00
Mid-Grade Heat Source Mass Flowrate (kg/s)	0.39
Cooling Water Source Temperature (°C)	4.44
Cooling Water Exit Temperature (°C)	10.00

Figure 5 shows the simulated optimised IPSEpro model of the single ORC system using mid-grade heat source. The thermodynamic cycle undergone by the R134a working fluid during the ORC process is presented in Figure 4. The explanation of the cycle is the same as that shown in Table III. The simulation result is shown in Table VI. The combined performance of the two single ORC power plants is shown in Table VII.

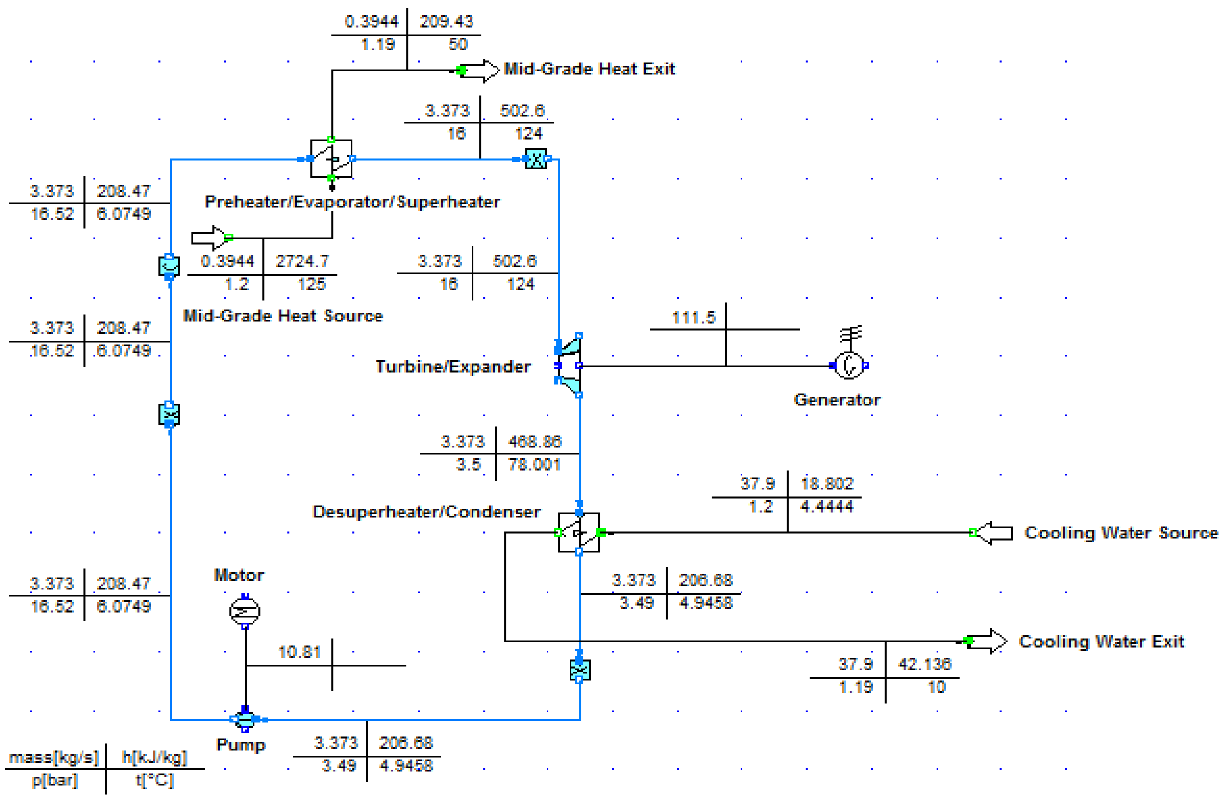

Figure 5: Simulated Optimised Model of the Single ORC System Using Mid-Grade Heat Source

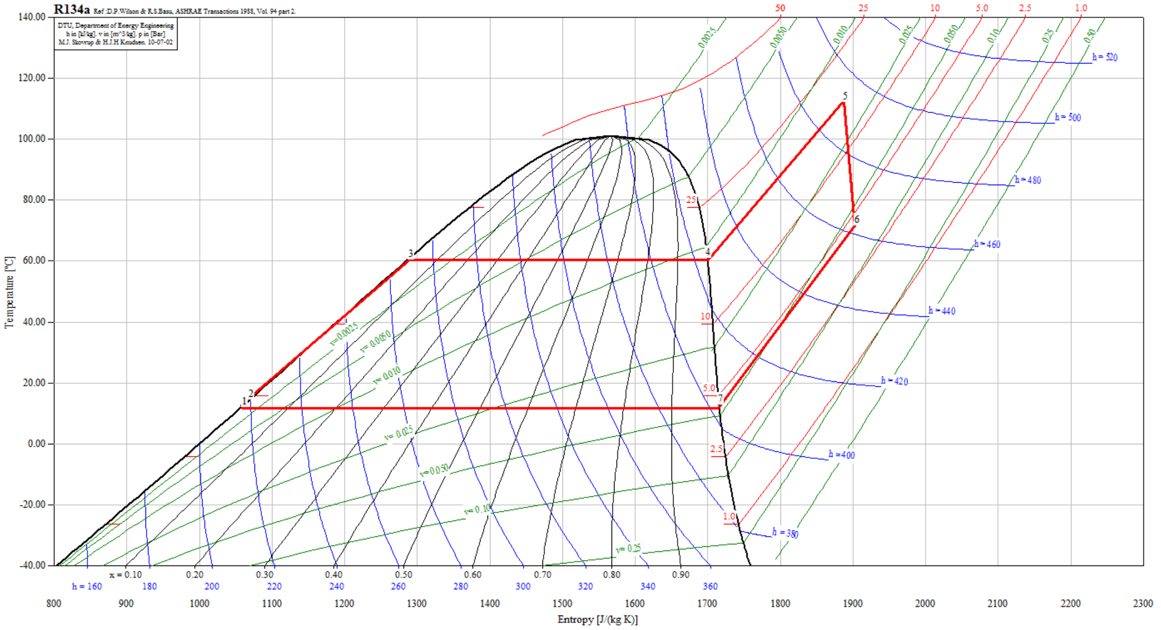

Figure 6: T-s diagram of R134a showing the Thermodynamic Cycle for a Mid-Grade Heat Source

Table VI: Simulation Result for Single ORC using Mid-Grade Heat Source

Parameter	**IPSEpro Simulation Result**
Mid-Grade Heat Source Temperature (^{o}C)	125.00 (Nominal)
Mid-Grade Heat Exit Temperature (^{o}C)	50.00 (Nominal)
Cooling Water Source Temperature (^{o}C)	4.44 (Nominal)
Mid-Grade Heat Source Mass Flowrate (kg/s)	0.39 (Nominal)
Cooling Water Mass Flowrate (kg/s)	37.90*
Cooling Water Exit Temperature (^{o}C)	10.00 (Nominal)
Refrigerant Mass Flowrate (kg/s)	3.37*
Turbine Efficiency (%)	80.00**
Turbine Inlet Pressure (bar)	16.00**
Turbine Outlet Pressure (bar)	3.50**
Pump Power (kW)	11.61*
Gross Generator Power (kW)	107.12*
Net Plant Power (kW)	95.51*

Thermal Efficiency (%)	9.60*
Evaporator UA value (kW/K)	87.42*
Condenser UA Value (kW/K)	64.32*
Evaporator Heat Transfer (kW_{th})	992.03*
Condenser Heat Transfer (kW_{th})	884.26*

* **calculated Variables** ** **Set Variables**

Table VII: Combined Performance of the Two Single ORC Power Plant each using a Single Heat Source

Parameter	**Calculated Result**
Total Waste Heat Transfer (kW_{th})	4252.19
Total Refrigerant Mass Flowrate (kg/s)	17.23
Total Cooling Water Mass Flowrate (kg/s)	162.89
Total Pump Power (kW)	58.35
Total Gross Generator Power (kW)	455.00
Total Net Plant Power (kW)	396.65
Total Evaporator UA value (kW/K)	355.21
Total Condenser UA Value (kW/K)	860.22

3.3 Single ORC System Using Dual Low- and Mid-Grade Heat Source

The model makes use of dual heat sources as explained in section 2.2. The low-grade heat source is used to preheat and vapourise the R134a working fluid while the mid-grade heat source is used in the superheating. The parameters used to model the system are shown in Table VIII. The simulated optimised model is shown in Fig. 7 while the T-s diagram showing the thermodynamic cycle is shown in Figure 8. The explanation of the cycle processes is similar as that shown in Table III. The simulation result for the single ORC system using dual low-and mid-grade heat sources is shown in Table IX.

Table X shows the comparison between the combined single heat source ORC system and the single dual heat source system.

Table VIII: Operating Conditions for the Single ORC System using Dual Low-and Mid-Grade Heat Source

Parameter	Nominal Value
Low-Grade Heat Source Temperature (°C)	73.33
Low-Grade Heat Source Mass Flowrate (kg/s)	33.39
Mid-Grade Heat Source Temperature (°C)	125.00
Mid-Grade Heat Source Mass Flowrate (kg/s)	0.39
Cooling Water Source Temperature (°C)	4.44
Cooling Water Exit Temperature (°C)	10.00

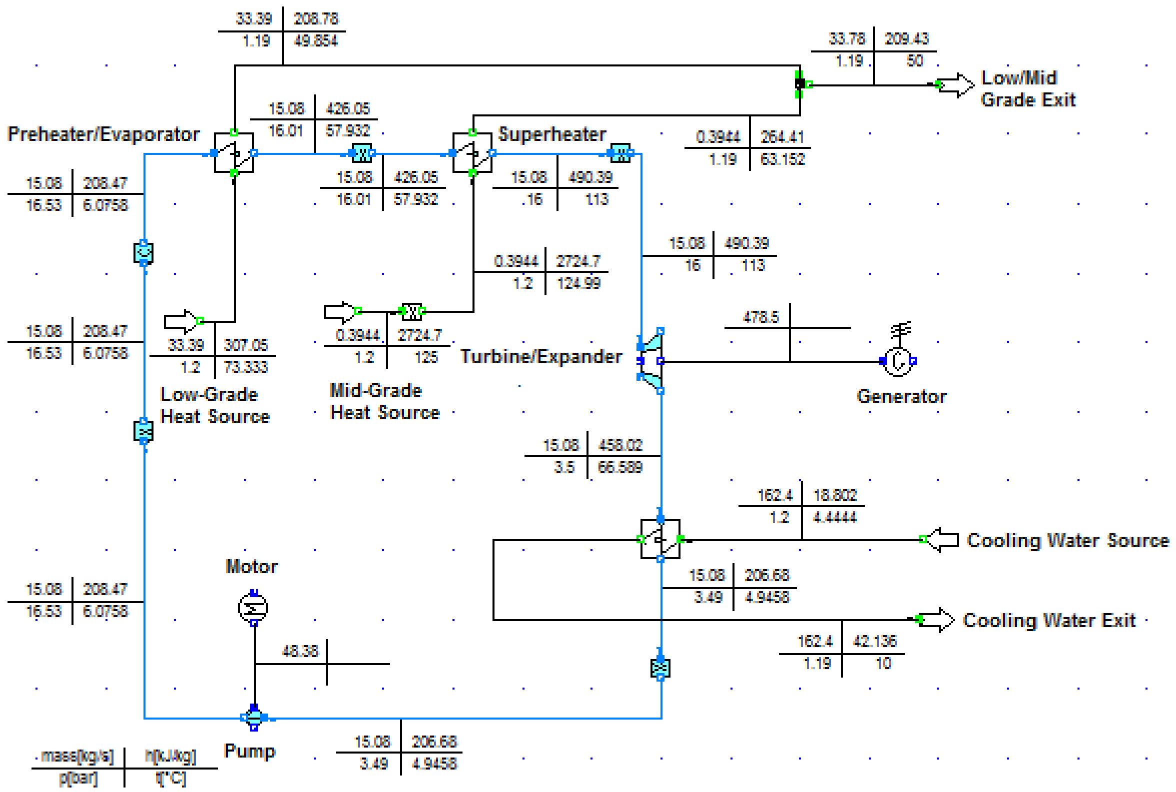

Figure 7: Simulated Optimised IPSEpro Model of the Single ORC System Using Dual Low-and Mid-Grade Heat Source

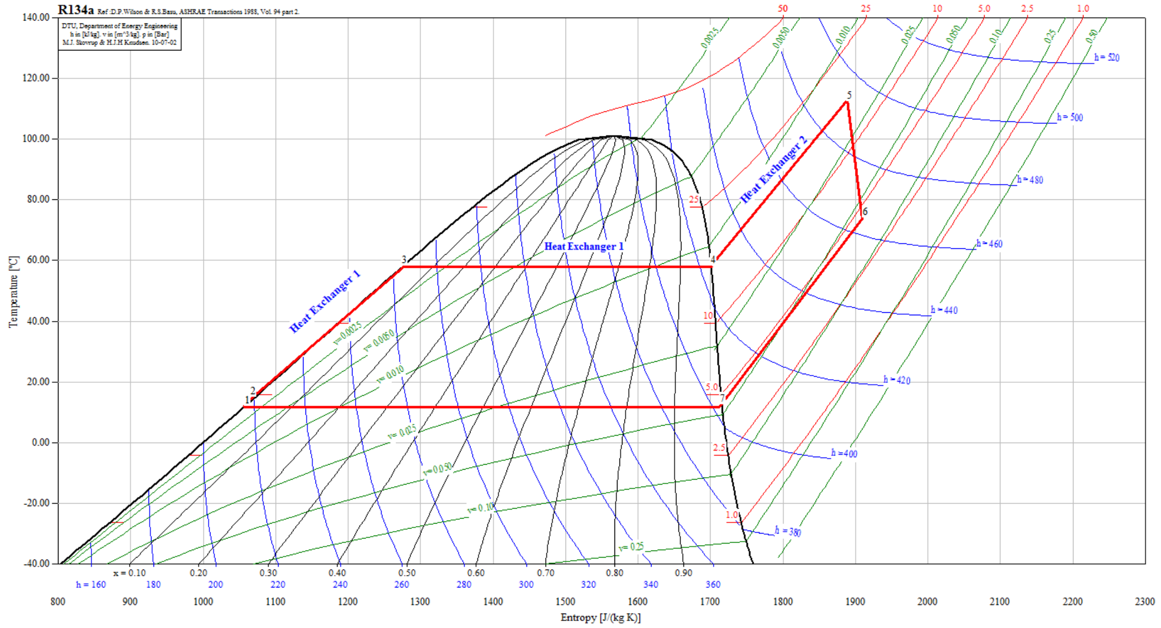

Figure 8: T-s diagram of R134a showing the Thermodynamic Cycle for the Dual Low-and Mid-Grade Heat Source

Table IX: Simulation Result for Single ORC using both Low- and Mid-Grade Heat Source

Parameter	**IPSEpro Simulation Result**
Low-Grade Heat Source Temperature (^{o}C)	73.33 (Nominal)
Low-Grade Heat Source Mass Flowrate (kg/s)	33.39 (Nominal)
Mid-Grade Heat Source Temperature (^{o}C)	125.00 (Nominal)
Mid-Grade Heat Source Mass Flowrate (kg/s)	0.39 (Nominal)
Cooling Water Source Temperature (^{o}C)	4.44 (Nominal)
Cooling Water Exit Temperature	10.00 (Nominal)
Low/Mid-Grade Heat Exit Temperature (^{o}C)	50.00**
Refrigerant Mass Flowrate (kg/s)	15.08*
Cooling Water Mass Flowrate (kg/s)	162.44*
Turbine Efficiency (%)	80.00**
Turbine Inlet Pressure (bar)	16.00**
Turbine Outlet Pressure (bar)	3.50**
Pump Power (kW)	51.97*

Gross Generator Power (kW)	459.55*
Net Plant Power (kW)	407.58*
Thermal Efficiency (%)	9.60*
Evaporator 1 UA value (kW/K)	120.80*
Evaporator 2 UA value (kW/K)	119.17*
Condenser UA Value (kW/K)	319.41*
Evaporator 1 Heat Transfer (kW_{th})	3281.85*
Evaporator 2 Heat Transfer (kW_{th})	970.34*
Condenser Heat Transfer (kW_{th})	3790.55*

* **calculated Variables** ** **Set Variables**

Table X: Comparison between two Single heat source ORC System and Single dual heat source ORC System

Parameter	**Combined Single ORC System**	**Single Dual Heat Source ORC System**
Total Waste Heat Transfer (kW_{th})	4252.19*	4251.19*
Total Refrigerant Mass Flowrate (kg/s)	17.23	15.08
Total Cooling Water Mass Flowrate (kg/s)	162.89	162.44
Total Pump Power (kW)	58.35	51.97
Total Gross Generator Power (kW)	455.00	459.55
Total Net Plant Power (kW)	396.65	407.58
Total Evaporator UA value (kW/K)	355.21	239.97
Total Condenser UA Value (kW/K)	860.22	319.41

* **Basis of comparison**

4.0 Economic Evaluation of the Processes

In this section, an approximate economic evaluation of the two approaches presented in this paper (electricity generation from low and mid-grade heat sources using two separate single ORC systems each operating with single heat sources and a single ORC system operating as a dual low- and mid-grade heat source) is carried out. The study is done using the concept of payback period. In order to carry out this analysis, the following assumptions were made:

- The cost of a heat exchanger is based on the UA value which reflects the heat exchanger area. The higher the UA value, the higher the heat exchanger cost.
- The pump cost is based on the pump's power rating. The higher the pump power rating, the higher the pump cost.
- Since the specific volume of the refrigerants at the inlet and exit of the turbine are similar in all the simulations, the turbine size is assumed to be similar.
- The piping cost is assumed to be the same for any single standalone ORC plant.
- The installation cost is assumed to be the same for any single standalone ORC plant.
- The maintenance cost is assumed to be the same for any single standalone ORC plant
- The cost of land acquired for the installation of the ORC plant is the same for any standalone ORC power plant.

Based on the above assumptions, the following variables are defined:

α = cost of an evaporator per unit UA value

ξ = cost of cooling water per unit mass (in kg)

β = cost of a condenser per unit UA value

λ = cost of a pump per unit kW

ω = cost of maintenance of a single ORC plant

φ = cost of a turbine for a single ORC plant

σ = cost of piping for a single standalone ORC plant

τ = cost of installation of a single standalone ORC plant

ρ = cost of refrigerant per unit mass (in kg)

μ = cost of land for installation of a single ORC plant

ℓ = cost of electricity per unit kWh

Table XI shows the cost and revenue (assuming an operation of 330 days in a year) associated with two single ORC systems using low and mid- grade heat source respectively and a single ORC using dual low- and mid-grade heat source

Table XI: Cost and Revenue from the two Approaches

Parameter	Two Single ORC using Low and	Single ORC using Dual Low-and

	Mid-Grade Heat Source Respectively	Mid-Grade Heat Source
Total Refrigerant Cost	17.23ρ	15.08ρ
Total Cooling Water Cost	162.89ξ	162.44ξ
Total Pump Cost	58.35λ	51.97λ
Total Evaporator Cost	355.21α	239.97α
Condenser UA Cost	860.22β	319.41β
Turbine Cost	2φ	φ
Piping Cost	2σ	σ
Installation cost	2τ	τ
Maintenance Cost	2ω	ω
Cost of Land	2μ	μ
Yearly Revenue from Electricity Sale	3141468.00ℓ	3228033.60ℓ

Using the concept of payback period (PBP), the payback period for the two approaches can be calculated as,

$$PBP = \frac{Capital\ Cost\ of\ the\ Plant}{Yearly\ Revenue} \tag{1}$$

Applying equation (1) to the two different systems, the payback period is calculated as

$$PBP_{two\ Single\ ORC} = \frac{17.23\rho + 162.89\xi + 58.35\lambda + 355.21\alpha + 860.22\beta + 2\Phi + 2\sigma + 2\tau + 2\omega + 2\mu}{3141468.00\ell/year}$$

$$PBP_{Dual\ Source\ ORC} = \frac{15.08\rho + 162.44\xi + 51.97\lambda + 239.97\alpha + 319.41\beta + \Phi + \sigma + \tau + \omega + \mu}{3228033.60\ell/year}$$

$$PBP_{two\ Single\ ORC} - PBP_{Dual\ Source\ ORC} =$$

$$\frac{8.25\rho + 15.51\xi + 25.09\lambda + 392.77\alpha + 1773.40\beta + 3.31(\Phi + \sigma + \tau + \omega + \mu)}{1.01 * 10^7 \ell/year}$$

As ρ, λ, α, β, Φ, σ, τ, ω, μ and ℓ are positive numbers,

$$PBP_{two\ Single\ ORC} - PBP_{Dual\ Source\ ORC} > 0$$

It means that,

$$PBP_{two\ Single\ ORC} > PBP_{Dual\ Source\ ORC}$$

From the preliminary analysis carried out in this work it can be confirmed that the payback period of a single dual source ORC system using low-and mid-grade heat source is smaller than the payback period of two single ORC system using low and mid-grade heat sources respectively.

5.0 Conclusion

From the modelling and approximate economic analysis carried out in this work, the following conclusions can be made:

1. It should be possible to design a single cycle that takes a much better advantage of the availability of two or more separate heat sources at different temperatures.
2. A single cycle using two separate low- and mid- grade heat sources gives a better payback period than two single cycles each operating with a separate heat source.
3. The use of a dual source ORC system reduces the size of heat exchangers (evaporators and condensers) when compared with the single ORC system.
4. The refrigerant consumption is also reduced and this helps in the reduction of cost.
5. For a given quality of low and mid-grade heat source, the dual source ORC cycle gives a better power output than two single source ORC system combined together.

6.0 Recommendation for Future Work

The use of a dual source ORC system instead of two single ORC systems when faced with a low and mid-grade heat sources is a promising venture as has been established in this paper. As explained in the introduction, since the working fluid is still highly superheated at the exit of the turbine, the efficiency of the system can still be improved by the use of a regenerator. One major shortcoming likely to be encountered in this approach is the decomposition or thermal cracking of the organic working fluid at very high temperature. Hence, the author highly recommends that more detailed investigation into the potentials of this innovative approach for waste heat recovery should be encouraged.

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