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The Performance of waste Heat Recovery Systems using Steam Rankine Cycle and Organic Rankine Cycle For Power Generation

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Abstract: ***This paper presents extensive modelling of an Organic Rankine Cycle (ORC) system for a combined cycle power plant and to compare and evaluate the performance of ORC and Steam Rankine Cycle (SRC). In addition, ORC as a second stage waste heat recovery system after SRC too was modelled. Conceptual design of an ORC was made to replace the SRC system used in the power plant and its performance was compared with that of the SRC above. Upon replacing the steam cycle with ORC, the system efficiency is 7.63 %. The total energy destruction is 5140.41 kW. The result shows that ORC delivers very low system efficiency. The steam cycle produces 202.5MW whereas the presented ORC produces just 1.016MW of power. On the other hand, if ORC is implemented on the chimney the system will produce 0.2% of extra power on top the current power production of 675MW. The efficiency of this system is 7.81%. It is recommended to add the ORC at the chimney to tap more useful energy from the otherwise waste energy rejected into the environment.***

Keywords: ***Comparative Study, Waste Heat Recovery System, Organic Rankine Cycle (ORC), Power Generation***

I.INTRODUCTION

In a traditional combined cycle power plant, burning of gas takes place in compressor, the produced thrust will rotate the gas turbine which will produce electricity when the turbine is coupled with generator. The hot gas will enter waste heat recovery boiler leaving from gas turbine. At the boiler steam will be produced upon exchange of temperature to water from hot gas. Heat Recovery Steam Generator (HRSG) is the term used to refer such boiler. This will increase the overall efficiency of a power plant. The steam produced from the heating of HRSG, will be used to run the steam turbine. It will contribute to the extra production of electricity, on top of the electricity produced by the gas turbine. Steam Rankine Cycle is widely used for power generation in combined cycle power plants, however Organic Rankine Cycle (ORC) too can be a great alternative to Steam Rankine Cycle (SRC) in many aspects. In SRC, water would be the working fluid, the evaporated gas would be steam, and the engine would be a steam turbine. ORC uses organic compounds as working fluid. SRC would be highly efficient when the temperature of heat source is more than 200ºC for steam and 400ºC for gas.

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When heat source is of the 'low grade', SRC would no longer be favorable. This is because the cycle efficiency will be very low if water to be used as working fluid. The organic compounds used in ORC, boil at relatively lower temperatures than water. This contributes to the possibilities of efficient and economic power generation using 'low grade' heat sources. A study to be carried out to compare the efficiencies and net power output of waste heat recovery systems using SRC and ORC. The comparisons were carried out to identify which cycle is more favorable when with varying thermodynamic parameters. The comparisons were carried out to prove the hypothesis that ORC can achieve superior performance compared to traditional steam Rankine cycle using 'low grade' heat sources. The objectives of this paper are to design and model an ORC system for a gas power plant, to analyze and compare the data of a real waste heat recovery system using SRC in a power plant with theoretical data of ORC and to compare and evaluate the performance of ORC and SRC based waste heat recovery systems.

II.LITERATURE REVIEW

Waste Heat Recovery

The need for immediate remedy or solution to reduce the greenhouse gases that our world is being captured by, is reflected on waste heat recovery applications [1]. Adding to that, it is a more economical to reuse the waste heat and thus increasing the profit of a power production company. The dependency on fossil-based fuels and impacts on environment as a product of power generation can be minimized through alternative source of energy such as renewable energy or diminishing the energy consumption which is highly not possible in the exponentially growing globalization. Renewable energy is where the world is headed to but its viabilities and profitability are still in discussions amongst many. Renewable energy may require to invest huge amount of model and may not be suitable for all geographical locations [2].

In addition, waste heat utilization has an upper hand in comparison with the renewable energy. Renewable energy such as wind and hydro based energies require mass development of land and space to initiate the extraction of resources process. Waste heat does not depend on such requirement. Moreover, waste heat energy harvesting has a relatively higher potential utilization factor than some renewable energies like solar or wind energy [3-4].

Reducing energy demand can be seen from three perspectives, which are a reduction in total activity of energy usage or power generation, better energy management, recovery and utilization of waste energy [2]. Although the performance of a company in terms of profit may not be affected by reducing the total activity, in this case power generation, a change in the company's business model is definitely required. However, this approach will not work for all types of companies. The discovery of energy recovery techniques are on the basis of energy is converted from one form to another and therefore there are useful energy available to be used as energy supply. Usually waste heat can be recovered in closed loop which will be recycled in its same process. Apart from that, waste heat could also be recovered in extended loop. These two approaches act as solution for additional energy demand for a system or a thermodynamic facility. Therefore, Heat Recovery systems are indeed vital not only for the environment but also in the business revenue gain.

Organic Rankine Cycle and Steam Rankine Cycle

The necessity and the role of heat recovery systems in capturing waste heat to put into reuse were discussed above. Currently we have various thermodynamic cycles that utilizes waste heat dissipated from power generation. Those cycles are SRC, Striling cycle, Brayton cycle, Kalina Cycle, and ORC [5]. When ORC is compared with the other waste heat recovering cycles, the latter exhibit certain degree of shortcomings. ORC has advantages such as low maintenance requirement, high safety and good thermal performance in recovering waste heat [6]. ORC is a similar thermodynamic cycle as SRC in terms of its operation, but they exhibit strong differences in terms of operating temperature range and the very working fluid of the cycles.

On the other hand, dry fluids are used in ORC. In other words, ORC uses volatile organic liquids or refrigerants as working fluid, rather than water as per in the SRC [7]. The advantages of ORC over SRC are the inlet temperature of turbine is relatively lower compared to that of the SRC. Low evaporating pressure, high condensing pressure, and last but not least low temperature heat recovery are the conditions that ORC operates on [8]. Organic fluids, unlike water, has the ability to recover heat energy from low grade heat waste. This is only possible because organic fluids have lower boiling points than that of water [9].

Comparison on Working Fluids of Both Cycles

Fluidprop and Cycle Tempo, computer simulations showed that effect of operating conditions, like the turbine and condenser inlet temperatures on the ORC [10]. The heat source's temperature range and condenser conditions determine the thermal efficiency. The characteristics of several organic fluids were evaluated by using simulation software packages such as Fluidprop and Cycle Tempo which were developed at Technical University of Delft. Unlike in the classic steam cycle, the critical pressure, turbine's inlet operating pressure in an ORC are much lower [10].

Based on these results, it was found that ORC display a superior performance compared to the simplified steam cycle, assuming equal temperatures at the turbine inlet, of the working fluids considered, the best ORC performance is achieved by toluene; and finally the application area of ORC based on conventional working fluids, without superheating, is confined to 300°C or lower temperatures [10]. A study on working fluid groups such as hydrocarbons, fluorocarbons, and siloxanes were done [11] which deduced that if the evaporation pressure is kept under working fluid's pressure, higher critical temperature of working fluid will contribute to higher thermodynamic cycle efficiency. The results of the study show that hydrocarbons of low critical temperature and fluorocarbons are the most apt working fluids for low grade thermal applications [11].

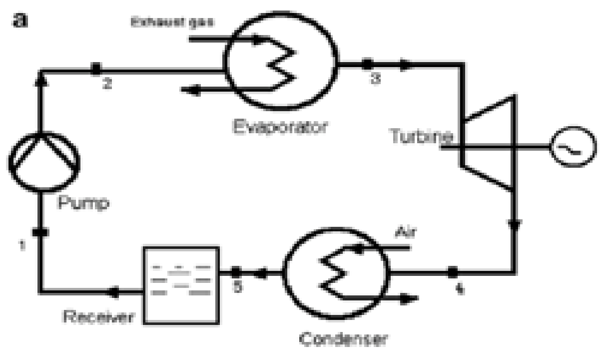
III.METHODOLOGY

The aim of this project was achieved by utilizing quantitative approach. This approach was mainly used in the project to develop and employ mathematical models for thermodynamic analysis of ORC system. Power plant that uses SRC as a waste heat recovery system has been identified. It is Kuala Langat Power Plant (KLPP), a 675MW combined cycle gas turbine power plant, owned by Edra Power Holdings Sdn. Bhd. [12]. Refrigerant R245fa is selected as a working fluid. The choice of working fluid is made due to the fact that the fluid can deliver suitable pressure for evaporation, helps in creating overpressure in condenser.

The R245fa fluid also promises an optimum level of process efficiency. The selected working fluid is also not subjected to greenhouse gas emission as it does not contribute to the depletion of ozone layer. The working fluid also can be handled safely as it is not flammable, has very low toxicity and thermally stable [13]. S. Masheiti*et al* studied on various organic working fluids for utilization of low temperature geothermal resource a sustainable energy supply for a city. From the research it was concluded by the researcher that R245fa had better performance [14] and it possesses high environmental safety levels.

Conceptual Design of Simple Organic Rankine Cycle

An ORC system consists of several vital thermodynamic processes as shown in Fig. 1, which are [15] heat addition process (point 2-3), expansion process (point 3-4a), heat rejection process (point 4a-5) and isentropic compression (point 1(5)-2).



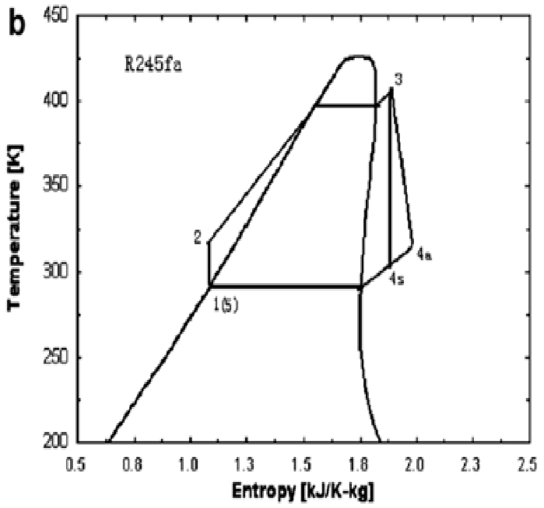


Fig. 1 (a)Schematic diagram of Simple ORC system [15]. (b)T-s diagram of Simple ORC [15]

In isentropic compression stage, exergy destruction rate, İ, in the pump is assumed to be relatively small, therefore this process can be assumed to be isentropic : $s_1 = s_2$ and İ = 0 kJ/s. The work done by the pump is [15]:

$$W_P = \dot{m}\ (h_2 - h_1)/\eta_p \quad \textbf{(1)}$$

Where W_P is work done by the pump in kW, $\dot{m}$ is mass flow rate of working fluid in kg/s, h is specific enthalpy at respective stage in kJ/kg and η_p is pump efficiency.

In heating addition process to the working fluid in evaporator, the heat transferred from the exhaust gas from the gas turbine to the working fluid of ORC [15] is:

$$q_{23} = \dot{m}(h_3 - h_2) \textbf{(2)}$$

Whereq_{23} is rate of heat transferred to working fluid, $\dot{m}$ is mass flow rate of working fluid and h is specific enthalpy at respective stage. Arithmetic mean temperature, T_H, of the inlet and outlet temperatures of exhaust gas is taken into consideration because, during the process, the temperature is not constant. T_H will be assumed to be temperature of the heat source [15]:

$$T_H = \text{Tin} + \text{Tout}/_2 \textbf{(3)}$$

Where T_H is Arithmetic mean temperature, T_{in} is temperature of exhaust gas at the inlet of evaporator and T_{out} is temperature of exhaust gas at the outlet of evaporator. Therefore, the exergy destruction rate in the evaporator is [15]:

$$\dot{I_{23}} = \dot{m}\, T_o [(s_3 - s_2\ ) - \ (h_3 - h_2)/T_H\ ] \quad \textbf{(4)}$$

where $\dot{I_{23}}$ is exergy destruction rate in the evaporator, $\dot{m}$ is mass flow rate of working fluid, T_o is temperature of ambient air and s is specific entropy at respective stage, h is specific enthalpy at respective stage and T_H is arithmetic mean temperature.

In expansion process, the turbine converts heat energy into mechanical energy finally to electrical energy here. Ideally, this is an isentropic process 3–4s but hundred percent efficiency of the energy transformation in the turbine can never be reached. Point 4_a is the turbine exit, the state of the working fluid in real case scenario. Turbine's isentropic efficiency is [15]:

$$\eta s = (h_{4a} - h_3)/(h_{4s} - h_3) \textbf{(5)}$$

Whereη_s is isentropic efficiency of turbine and $h$is specific enthalpy at respective stage.Power generated by turbine is [15]:

$$W_t\ = \dot{m}(h_3 - h_{4s})\eta_s\eta_m\ = \dot{m}(h_3 - h_{4a})\eta_m \quad \textbf{(6)}$$

WhereW_t ispower generated by turbine, and η_m is mechanical efficiency of turbine. Therefore, the exergy destruction rate in the turbine is [15]:

$$\dot{I_{34a}} = \dot{m}\, T_o [(s_{4a} - s_3)] \quad \textbf{(7)}$$

Where$\dot{I_{34a}}$is exergy destruction rate in the evaporator. There is heat rejection process from working fluid to the cooling water happens in the condenser. The rate of heat rejection is [15]:

$$q_{4a5}\ = \dot{m}\ (h_5 - h_{4a}) \quad \textbf{(8)}$$

Whereq_{4a5} is rate of heat transferred to working fluid. Since the temperature of cooling water changes during the process, the arithmetic mean temperature of cooling water is taken [15]:

$$T_{cooling} = (\text{Tin, c} + \text{Tout, c})/2 \quad \textbf{(9)}$$

Where $T_{cooling}$ is arithmetic mean temperature of cooling water, $T_{in,c}$ is temperature of cooling water at the inlet of condenser and $T_{out,c}$ is temperature of cooling water at the outlet of condenser. Therefore, the exergy destruction rate in the condenser is [15]:

$$\dot{I_{4a5}} = \dot{m}\, T_o [(s_5 - s_{4a}\ ) - \ (h_5 - h_{4a})/T_{cooling}\ ] \quad \textbf{(10)}$$

where $\dot{I_{4a5}}$ is exergy destruction rate in the condenser, $\dot{m}$ is mass flow rate of working fluid, $T_{cooling}$ is arithmetic mean temperature of cooling water and s is specific entropy at respective stage. Hence, net power generated by ORC system, W_{sys} is [15]:

$$w_{sys} = W_t - W_p \quad \textbf{(11)}$$

Where $W_{sys}$is net power generated by ORC system,W_t is power generated by turbine, W_P is work done by the pump.Therefore, net efficiency of system, η_{sys} [15]:

$$\eta_{sys}\ = w_{sys}/(\ m\ (h_3 - h_2) \quad \textbf{(12)}$$

Whereη_{sys} is net efficiency of ORC system, $W_{sys}$is net power generated by ORC system, $\dot{m}$ is mass flow rate of working fluid, and h is specific enthalpy at respective stage. Total exergy destruction rate of cycle, $\dot{I_{sys}}$ [15]:

$$\dot{I_{sys}} = \dot{I_{23}} + \dot{I_{34a}} + \dot{I_{4a5}} \quad \textbf{(13)}$$

IV.RESULTS AND DISCUSSIONS

Technical data from Kuala Langat Power Plant (KLPP) running Steam Cycle in a Combined Cycle Gas Turbine (CCGT) were obtained. These data comprised of operating conditions and parameters set by the power station for efficient running of the plant as shown in Table 1.

The obtained data were comprised of operating parameters of gas turbine, High Pressure (HP) and Low Pressure (LP) drums of Heat Recovery Steam Generator (HRSG), parameters of steam turbine, condenser, evaporator and that of cooling water and flue gas leaving chimney. Taking these data input into consideration, a conceptual design of simplified ORC is modelled mathematically using thermodynamic equations.

Before performing the mathematical analysis, certain general assumptions needed to be made to avoid complexity of calculations and analysis. The assumptions are steady state and steady flow condition, mass and energy are conserved in each stage of process, no pressure loss or energy loss; External heat loss are negligible , pressure and temperature of condenser and evaporator should not exit the critical limit of the working fluid, R-245fa which are 3640 kPa and 427.2 K.

Organic Rankine Cycle (ORC) Replacing Steam Cycle at KLPP

Table 1 shows the operating conditions of KLPP at which the plant is designed to run. These conditions were decided to be maintained in further calculations to ensure the design of systems to be as practical as possible.

Table. 1 Operating condition of plant

No	Parameters of systems at KLPP	Values
1	Mass flow rate of refrigerant, m(ref)	60 kg/s
2	Pump efficiency, n_p	78%
3	Isentropic efficiency of turbine, n_s	90 %
4	Mechanical efficiency of turbine, n_m	70 %

The result as shown in Table 2, shows very low system efficiency of Organic Rankine Cycle (ORC). Replacing the in-use steam cycle will severely reduce the total cycle efficiency of the power plant. The current steam cycle produces 30% of the total power generation of 675MW. In other words, the steam cycle produces 202.5MW whereas the presented ORC produces just 1.016MW of power.

Table. 2 Result of mathematical analysis on net power generation and plant efficiency

No.	Net power generated by ORC	Values
1	Power produced by turbine, W_t	1170.2124 kW
2	Required work energy, W_p	153.846 kW
3	Net power generated, W_{sys}	1016.366 kW
4	Mass flow rate of refrigerant, m(ref)	60 kg/s
5	Specific enthalpy at state 3, h_3	475.95 kJ/kg
6	Specific enthalpy at state 2, h_2	254 kJ/kg
7	System efficiency, N_{sys}	7.6318%

Organic Rankine Cycle (ORC) at Chimney of KLPP

Table 3 shows the result of mathematical analysis conducted at HRSG, where its inlet and outlet temperature, mass flow rate were the operating parameter at KLPP and the available heat energy in exhaust gas is 107.844 MW.

Table. 3 Result of mathematical analysis at HRSG

No	HRSG	Values
1	Mass flow rate of steam, m(steam)	60 kg/s
2	Specific heat capacity of ste4am, c_p	4.185kJ/kg/K
3	Inlet temperature of water at HRSG	65^{o}C
4	Exit temperature of steam at HRSG	495^{o}C
5	Heat into steam from exhaust gas, q	107.844 MW

Table 4 shows the result of mathematical analysis conducted at chimney, where its inlet and outlet temperature limits are conditional parameters and the available heat energy in flue gas is 117.906 MW.

Table. 4 Result of mathematical analysis at chimney

No	Chimney	Values
1	Mass flow rate of exhaust gas, m(ex)	500 kg/s
2	Specific heat capacity of exhaust gas, c_p	1.075kJ/kg/K
3	Inlet temperature at HRSG	540^{o}C
4	Heat into steam from exhaust gas, q	107.844 MW
5	Temperature before chimney's exit	339.36^{o}C
6	Exit temperature at chimney	120^{o}C
7	Available heat in flue gas, q_{ORC}	117.906 MW

Table 5 shows the result of mathematical analysis conducted at evaporator, where its upper limit is set at 1.35 MPa considering the critical pressure of the working fluid, the specific enthalpies and entropies and thus the exergy destruction in evaporator is 2177.251 kW.

Table. 5 Result of mathematical analysis at evaporator

No.	Evaporator	Values
1	Mass flow rate of refrigerant, m(ref)	60 kg/s
2	Evaporator pressure (upper limit)	1.35 MPa
3	Specific enthalpy at state 3, h_3	475.95 kJ/kg
4	Specific enthalpy at state 2, h_2	225.6 kJ/kg
5	Heat absorbed from exhaust gas, q_{in}	15021.48 kW
5	Arithmetic mean temperature, T_h	503.005 K
6	Specific entropy at state 3, s_3	1.7926 kJ/kg/K
7	Specific entropy at state 2, s_2	1.179 kJ/kg/K
8	Exergy destruction in evaporator, i_{23}	2177.251 kW

Table 6 shows the result of mathematical analysis conducted at compressor, where the resulted work energy that is required is 468.46 kW when maintaining the pump efficiency of 78% at KLPP.

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Table. 6 Result of mathematical analysis at compressor

No	Isentropic compressor/pump	Values
1	Mass flow rate of refrigerant, m(ref)	60 kg/s
2	Evaporator pressure (upper limit)	1.35 MPa
3	Specific enthalpy at state 2, h_2	225.6 kJ/kg
4	Specific enthalpy at state 1, h_1	219.51 kJ/kg
5	Pump efficiency, n_p	78%
6	Required work energy, W_p	468.46 kW

Power produced by turbine and exergy destruction are shown in Table 7. The result of power produced by turbine and exergy destructed in evaporator are 1642.7124 kW and 1078.523 kW respectively.

Table. 7 Result of mathematical analysis at turbine

No.	Turbine of ORC	Values
1	Isentropic efficiency of turbine, n_s	90 %
2	Specific enthalpy at state 3, h_3	475.95 kJ/kg
3	Specific enthalpy at state 4, h_4	432.5 kJ/kg
4	Actual specific enthalpy at state 4, h_{4a}	436.84 kJ/kg
5	Mechanical efficiency of turbine, n_m	70 %
6	Power produced by turbine, W_t	1642.7124 kW
7	Actual specific entropy at state 4, s_{4a}	1.85 kJ/kg/K
8	Specific entropy at state 3, s_3	1.7926 kJ/kg/K
9	Exergy destruction in turbine, i_{34a}	1078.523 kW

Table 8 shows the result of mathematical analysis conducted at condenser, where the lower limit is set in regards of the critical pressure of working fluid, the specific enthalpies, and specific entropies are also shown in the table below. The destructed exergy in condenser is 749.933 kW.

Table. 8 Result of mathematical analysis at condenser

No	Condenser	Values
1	Mass flow rate of refrigerant, m(ref)	60 kg/s
2	Condenser pressure (lower limit)	0.10133MPa
3	Specific enthalpy at state 4, h_{4a}	436.8458 kJ/kg
4	Specific enthalpy at state 5, h_5	219.51 kJ/kg
5	Heat rejected from refrigerant, q_{out}	13040.148 kW
6	Mass flow rate of cooling water, m(cool)	9500 kg/s
7	Inlet temperature of condenser	305.55 K
8	Specific heat capacity of cooling water, c_p	4.185 kJ/kg/K
9	Outlet temperature of condenser	305.877 K
10	Specific entropy at state 5, s_5	1.179 kJ/kg/K
11	Actual specific entropy at state 4, s_{4a}	1.85 kJ/kg/K
12	Exergy destruction in turbine, i_{4a5}	749.933 kW

Table 9 shows the result of mathematical analysis on net power generated by the system and plant efficiency. The net power generated is 1174.25 kW and the system efficiency is 7.8171 %.

Table. 9 Result of mathematical analysis on net power generation and plant efficiency

No	Net power generated by ORC	Values
1	Power produced by turbine, W_t	1642.7124 kW
2	Required work energy, W_p	468.46 kW
3	Net power generated, W_{sys}	1174.250 kW
4	Mass flow rate of refrigerant, m(ref)	60 kg/s
5	Specific enthalpy at state 3, h_3	475.958 kJ/kg
5	Specific enthalpy at state 2, h_2	225.6 kJ/kg
6	System efficiency, N_{sys}	7.8171%

The operating conditions were maintained as the first approach. However, in a different perspective, the ORC can be installed at the chimney to tap more useful heat from the flue gas temperature. This is an alternative or additional method to increase the power production in KLPP. Power rejected from exhaust gas to steam is 107844 kW. The temperature exit from the exhaust gas is 540ºC as per data obtained from KLPP. Therefore, the temperature of flue gas reaching the chimney leaving the HRSG is 339.36ºC. Flue gas leaving chimney must be minimum of 100ºC according to Original Equipment Manufacturer (OEM) of KLPP. The actual temperature of flue gas leaving the chimney in KLPP is 120ºC. Taking this value to be exit temperature of chimney, the flue gas has 117906 kW of available energy in it.

The mathematical procedures in the first approach, that is replacing the steam cycle with ORC, is repeated in this alternate way of tapping more energy from chimney. The upper pressure limit is 1.35MPa, as the enthalpy at stage 3 is 475.958 kJ/kg and entropy 1.7926 kJ/kg/K. Eventually, the net power generated from tapping the hot flue gas from chimney is 1174.250 kW. On the other hand, the exergy destructed is 4005.707 kW. Additional 1.17 MW of extra power will be produced if ORC is implemented on the chimney. In other words, the ORC system will produce 0.2% of extra power on top the current power production at KLPP which is 675MW. Although extra power is generated, the efficiency of the system is just 7.81%. The exergy destruction in the second implementation of ORC is lower than that of in the first implementation of ORC, replacing the steam cycle. This shows that, useful energy destructed in the first implementation is higher than the second implementation.

The comparisons made above, apart from achieving the objectives, it also serves as a structured analysis that contributes to the feasibility assessment of ORC system to be implemented as heat recovery system in a gas power plant. The methodology and thermodynamic analysis presented have aided in the comparison and evaluation of performance of ORC and SRC and their strategic implementation in gas power plant. The strategy of implementing the ORC at chimney will further absorb more useful heat from the otherwise waste heat which will be released to the environment through chimney as proven in the calculation above.

V.CONCLUSION

The core idea of the objectives is to propose a thermodynamic and mathematical analysis that proves the feasibility of implementing ORC at gas power plant to recover waste heat from exhaust gas. Implementation of ORC at the chimney have been deduced to be the best strategy to recover waste heat at KLPP power plant. 1.174 MW of power can be produced, adding 0.2% extra power in the power production at KLPP. Hence, instead of replacing the readily functioning SRC as a heat recovery system to the Gas Turbine of KLPP, it is optimum to add the ORC at the chimney to tap more useful energy from the otherwise waste energy rejected into the environment.

The study compares ORC and SRC systems as waste heat recovery systems and also aim to find out whether or not implementing ORC in gas power plant will be a feasible concept. Cost analysis that will show the economic viability of this proposed idea will require change in initial thermodynamic assumptions which would further complicate the general proving of ORC implementation at gas power plant to recover waste heat. Therefore, cost analysis have not been included in the focus of this preliminary study.

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