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Performance Analysis of a Shell Tube Condenser for a Model Organic Rankine Cycle for Use in Geothermal Power Plant

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ABSTRACT
The global energy demand increases with the economic growth and population rise. Most electrical power is currently generated by conventional methods from fossil fuels. Despite the high energy demand, the conventional energy resources such as fossil fuels have been declining. In addition to this harmful combustion byproducts are resulting global warming. However, the increase of environmental concerns and energy crisis can be minimized by sustainable utilization of the low to medium temperature heat resources. The Organic Rankine Cycle power plant is a very effective option for utilization of low grade heat sources for power generation. Heat exchangers are the main components of the Organic Rankine Cycle power plant which receives heat energy from the heat source to evaporate and condense the low boiling temperature organic working fluid which in turn drives the turbine to generate power. This paper presents a simplified approach to the design, fabrication and performance assessment of a shell tube heat exchanger designed for condenser in a model Organic Rankine Cycle geothermal power plant. The design involved sizing of heat exchanger (condenser) using the LMTD method based on an expected heat transfer rate. The heat exchanger of the model power plant was tested in which hot water simulated geothermal brine. The results of the experiment indicated that the heat exchanger is thermally suitable for the condenser of the model power plant.

***Keywords* -** Shell tube heat Exchanger, Heat transfer Co-efficient, LMTD.

I. Introduction

Energy consumption increases with growth in economies and population. The increase in energy demand will occur mainly in developing countries, where economic growth rates are high and people are shifting from biomass energy sources like wood and agricultural waste to electricity. However, the increase of environmental concerns and energy crisis has resulted in need for a sustainable approach to the utilization of the earth's energy resources.

Worldwide, geothermal power plants have a capacity of about 12 GW power generations as of 2013 and in practice supply only about 0.3% of global power demand [1]. However, the conventional geothermal power plants utilize the high temperature and pressure geo-fluid and operate at low efficiency due to the heat loss in the exhausted steam and brine. The Organic Rankine Cycle power plant is a very attractive option for utilization of low-grade geothermal heat sources for power generation. Heat exchangers are the main components of the Organic Rankine Cycle geothermal power plant to receives heat energy from the hot water (brine) and vaporize and condense the low boiling temperature organic working fluid which drives the turbine to generate power.

This paper presents design, fabrication and performance assessment test of the shell and tube heat exchanger designed for the condenser of a model Organic Rankine Cycle power plant. Performance tests of the heat exchanger were conducted by relating the inlet and outlet temperatures and the overall heat transfer coefficient to the rate of heat transfer between the working fluid (n-pentane) and the cooling water.

Heat exchanger is an apparatus or equipment built for efficient heat transfer between two or more fluid streams at different temperatures. The shell tube heat exchanger is widely used in many binary cycle power plants. This widespread application can be justified by the availability of codes and standards for design and fabrication; it can be manufactured from a wide variety of materials and ability to be produced in the widest variety of sizes and styles. Moreover, It can be designed for a wide range of limits on the operating temperature and pressure [2, 3]. A typical counter flow shell and tube heat exchanger is shown in figure 1

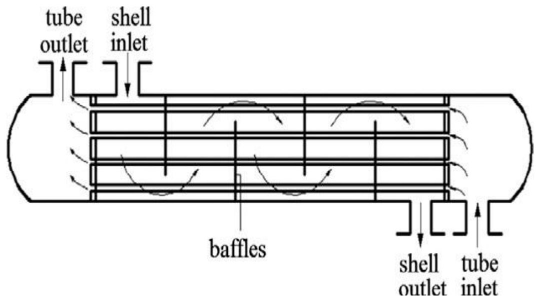

Figure 1: Counter flow shell and tube heat exchanger [1]

The three most commonly used types of shell and tube heat exchanger designs are fixed tube sheet, U-tube, and the floating head type. In all types, the front end head is stationary, while the rear-end head could be either stationary or floating depending upon the thermal stresses in the shell, tube, or tube sheet due to temperature differences as a result of heat transfer. The heat exchangers are built in accordance with three mechanical standards that specify design, fabrication, and materials of unfired shell and tube heat exchangers. The two most common heat exchanger design problems are those of sizing and rating of the heat exchanger [4].

Different types of heat exchangers have been developed to meet the widely varying application based on, operating principles, flow arrangement, compactness, number of fluids and constructional design features. Among the heat exchangers are plate, shell tube and shell coil heat exchangers. Shell and tube heat exchangers are basically non compact exchangers. Generally the Heat transfer surface area per unit volume is high ranging from about $50m^2/m^3$ to $100m^2/m^3$. Thus, they require a considerable amount of space, supporting structure, and higher capital and installation costs. The shell and tube heat exchangers are more effective for applications in which compactness is not a priority [4].

Madhawa [5] investigated the at plate and shell tube heat exchangers and found that at plate type heat exchangers are effective in the evaporator and condenser when considering low-temperature heat sources in which large heat exchanger area (per unit power output) is required. The plate type heat exchangers are preferred due to their compactness and high heat transfer co-efficient which result to less heat transfer area than would be needed using shell and tube heat exchanger. However, at plate exchanger involves high manufacturing cost and maintenance cost. Hence it affects the overall cost of the power plant.

Bambang [6] also compared the at plate and shell tube heat exchangers and found that the shell tube type is advantageous due to simplicity in geometry, well established design procedure, can be constructed from a wide range of materials, uses well-established fabrication techniques and is easily cleaned. Moreover the shell tube heat exchanger can be designed and manufactured locally from a wide range of materials. However the heat transfer optimization and minimizing the pressure drop within the shell and tube exchanger need to be addressed.

The design of heat exchangers involves establishing the right flow configuration of interacting fluids. Parallel and counter flows are the two common flow configuration of a shell tube heat exchanger. The counter flow configuration is the predominantly preferred flow direction in liquid to liquid heat exchangers since it results in a higher temperature difference driving the heat transfer within the heat exchanger, smaller heat transfer surface area required [7]. Moreover counter flow configuration is most effective design when there is a temperature cross. A temperature cross occurs when the desired outlet temperature of one fluid is between the inlet and outlet temperatures of the other fluid [8].

Furthermore, various studies have been carried out on optimization of the performance of shell and tube heat exchangers using the performance parameters approach. The heat transfer coefficient values are evaluated using the log mean temperature difference (LMTD) method from the temperature difference and the heat transfer area for known inlet and outlet temperature heat exchangers [9]. Thundil et al. [10] investigated the effect of inclination of baffles in the shell by simulating a model shell and tube heat exchanger. He compared the impact of baffle inclination on fluid flow, pressure drop and the heat transfer characteristics of a shell tube heat exchanger using three different inclination angles (0^o, 10^o and 20^o). They concluded that shell and tube heat exchanger with 20^o baffle inclination angle results in better performance compared to 10^o and 0^o inclination angles.

A. Singh et al. [11] also conducted an experimental analysis on the performance of a shell tube heat exchanger with segmented baffles at three different orientations (0^o, 30^o and 60^o). They analyzed the system for laminar flow with varying Reynold number and concluded that the heat transfer coefficient increases with increase in Reynold number in shell tube heat exchanger for both hot fluid and cold fluids. They observed that, with the introduction of the baffles, the heat transfer coefficient increases leading to more heat transfer rate due to introduction of swirl and more convective surface area.

In general, conventional shell tube heat exchangers result in high shell side pressure drop and formation of recirculation zones near the baffles. To overcome the challenge helical baffles, which gives better performance than single segmental baffles, can be used. But these baffles involve high manufacturing and maintenance cost. Hence the effectiveness and cost must be considered in the heat exchanger design [10].

II. Methodology

2.1 Heat exchanger selection

Heat is transferred from one fluid to other in the heat exchanger. The process was studied by taking into account the energy balance. Studies in the application of heat exchangers leads to the selection of three important attributes related to the performance of ORC. Which are heat transfer area, overall heat transfer coefficient and temperature difference [13].

Different type of heat exchanger (plates, tubular, shell and tube etc) can be used for the condenser of an Organic Rankine Cycle geothermal power plant. For indirect heat transfer between two fluids, the shell tube heat exchangers are more effective. In this study, due to the advantages of fairly simple geometry, well established design procedure, can be fabricated from a wide range of materials, uses well established fabrication techniques and ease of cleaning, a shell tube type heat exchanger was selected for the condenser of the model power plant.

The design of heat exchanger involves establishing the right flow configuration of interacting fluids. Parallel flow and counter flows are the two most commonly used flow configuration of shell tube heat exchanger. The counter-flow configuration is the predominantly preferred flow direction in liquid to liquid heat exchangers since it results in a higher temperature difference driving the heat transfer within the heat exchanger, smaller heat transfer surface area required. Moreover counter flow configuration is most effective design when there is a temperature cross. A temperature cross occurs when the desired outlet temperature of one fluid is between the inlet and outlet temperatures of the other fluid.

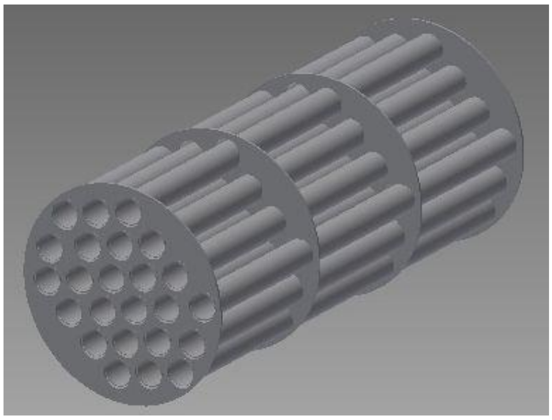

Figure 2: The design of copper tube configuration and baffle spacing

The design of the condenser is a fixed tube counter flow shell and tube heat exchanger as shown in figure 1. Moreover, the fluids flow pattern in the exchanger is in such way that the cooling water flows in the tube side and the secondary fluid (n-pentane) condenses in the shell side of the heat exchanger. This type of fluid pattern allows fouling fluid to flow through the tubes (easier to clean) and the organic working fluid (n-pentane) to flow through the shell side where a turbulent flow is obtained on the shell side due the baffles.

2.2 Design and construction of the heat exchanger

In design of short heat exchangers it is reasonable to assume a constant value of overall heat transfer coefficient (U). The log-mean temperature difference (LMTD) method is useful for the sizing and rating of heat exchanger of known mass flow rate and range of temperature difference between the inlet and outlet of fluid streams. The following equations have been used in relating the heat transfer surface area through which heat flow occurs under the driving force of temperature difference, amount of heat transferred and overall heat transfer coefficient.

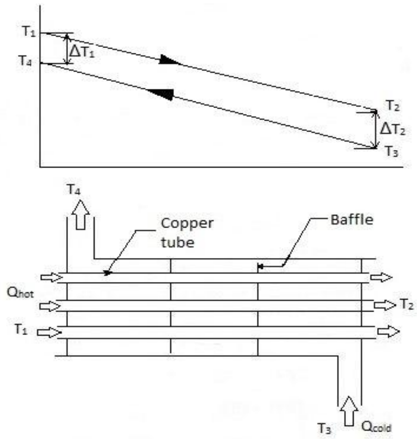

Figure 3: Schematic diagram of flow configuration of counter flow shell and tube heat exchanger

$$Q_{hot} = C_h(T_{hi} - T_{ho}) \quad (1)$$

Where heat capacity rate for hot or cold fluid, C = mC_p, and C_p is specific heat capacity at constant pressure.
During the heat transfer process energy is conserved therefore for the cold and hot fluid;

$$Q_{hot} = Q_{cold} = C_h(T_{hi} - T_{ho}) = C_c(T_{co} - T_{ci}) \quad (2)$$

Where heat capacity rate for hot or cold fluid C = mcp, subscripts h_i and h_o represent inlet and outlet of hot fluid respectively and C_i and C_o represent cold fluid inlet and outlet respectively.
Heat transferred in the process (Q) may be related to the overall heat transfer coefficient U and the mean temperature difference ΔT_{lmtd} by:

$$Q = AUT_{LMTD} \quad (3)$$

Where A is heat transfer surface area and ΔT_{lmtd} is the log mean temperature difference. The log mean temperature difference of the heat exchanger can be determined as follows:

$$T_{LMTD} = \frac{(\Delta T_1 - \Delta T_2)}{\ln[\frac{(\Delta T_1)}{(\Delta T_2)}]} = \frac{(T_1 - T_4) - (T_2 - T_3)}{\ln[\frac{(T_1 - T_4)}{(T_2 - T_3)}]} \quad (4)$$

Where, ΔT_1 and ΔT_2 represent the temperature differences at the inlet and exit of the heat exchanger respectively as shown in figure 3.

The heat exchanger was designed using the design equations 1-4. The details of the designed shell tube heat exchanger are shown in Table 1. Figure 2 shows the assembly of the copper tubes to the tube sheet and the baffles placed on the respective position.

Table 1: Design details of the condenser

Description	*Tube side*	*Shell side*
Fluid pattern	pentane	cooling water
Number of tubes	21	
Total tube length	8.4	
Tube diameter (DO/DI)(mm)	12.7/10.9	150
Heat transfer area (m^2)	0.2634	
Number of pass	1	
Tube configuration	30^0 triangular	
Tube pitch	16	
Material	Copper	Galvanized steel

The shell of the heat exchanger was constructed by rolling a plate of galvanized steel. The plate was cut to a rectangular shape of size 471mm by 750mm and four holes were drilled for the inlet and outlet of the two fluids before the plate was rolled. The plate was then rolled and brazed using gas welding and produced a cylinder of 150mm in diameter and height of 750mm. The tube sheets were made from a round at piece of galvanized steel with holes drilled for the tube ends in a precise location and pattern relative to one another.

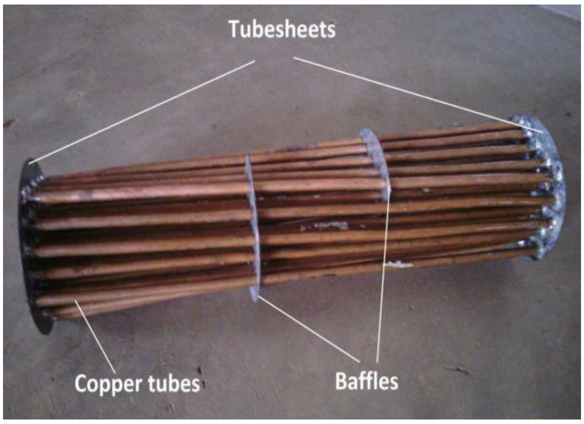

Figure 4**:** The shell and tube heat exchanger tube bundle

Two baffles were also provided as shown with a baffle pitch of 150mm. The optimum baffle pitch (spacing between segmental baffles) and the baffle cut were applied to determine the cross flow velocity and hence the rate of heat transfer and minimize pressure drop. A baffle spacing of 0.2 to 1 times the inside shell diameter is commonly used. A baffle cut of 20 to 25% provides a good heat-transfer with the reasonable pressure drop [14]. A 20% baffle cut was applied for this particular shell tube heat exchanger. The % cut for segmental baffle refers to the cut away height from its diameter.

Copper tubes were cut at a length of 400mm and each tube was brazed using a gas welding to the hole provided on the tube sheet to produce a tube bundle as shown in figure 4. The tube bundle was then inserted to the main shell and the two tube sheets were brazed to provide air tight joints.

2.3 Heat exchanger leakage test

It is important to test the leak tightness of the heat exchanger at the operating temperature and pressure before carrying out a performance test. Leak tightness test was carried out to ensure that the heat exchanger performs adequately at the design temperatures and pressures. The heat exchanger was tested for its leak tightness at room temperature and the design pressure of 2.5 bar using water circulating in the tube side. After inspecting for the leak, water was circulated on the shell side of the exchanger with the help of a water pump.

During the tests, leaks were mainly detected on the welded joints between the shell and the tube sheet, copper tubes and tube sheet joints and on the joints of the inlet and exit pipes of the heat exchanger. A sealing material called STAG was applied on the leaking edges and test was carried again until no further leakage was detected. STAG is easy break, smooth consistency and lead-free jointing compound. It is commonly used for sealing of high pressure and temperature pipes. It is a non-poisonous and resists to organic chemicals.

2.4 Performance test of the heat exchanger

The performance test unit consists of an overhead cooling water tank, the heat exchanger, secondary fluid tank and feed pump. A schematic sketch showing valves, pressure gauges, flow meters, and the location of temperature sensors is given in Figure 5. The cooling water flow rate was controlled by opening valve HV1. The feed pump supplies the organic working fluid (n-pentane) at constant flow rate of 0.12kg/s to the exchanger (evaporator) through the flow meter and condenses on the shell side in a counter-current to the cooling water flow. The condensed working fluid from the exchanger was then directed back to the secondary fluid tank.

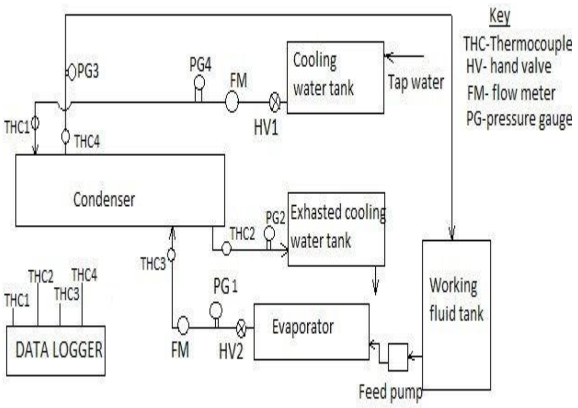

Figure 5: Schematic sketch of the heat exchanger performance test experiment.

The length of one test section was set to be two minutes. A set of four thermocouples were provided to record pertinent temperatures at the inlet and exit of each fluid as shown in the figure 5. The thermocouples were connected to a data logger of type TDS-530. The data logger was set to print each temperature in the test section of two minutes. A set of five measurements were conducted by varying the cooling water mass flow rate and maintaining the secondary fluid (n-pentane) flow rate constant. The temperature indicators on the data logger displayed the following temperatures.

- T_1 Cooling water inlet temperature to condenser
- T_2 Cooling water exit temperature from condenser
- T_3 Secondary fluid temperature inlet to condenser
- T_4 Secondary fluid temperature exit from condenser

2.5 Experimental procedure

The overhead cooling water tank was filled with water. The valve HV1 was opened and cooling water was allowed into the shell side of the heat exchanger (condenser). The feed pump was switched on and the secondary fluid (n-pentane) allowed to flow into the tube side of the heat exchanger. It was waited until the steady state has been reached. At steady state, all the four temperatures and flow rates of cooling water and secondary fluid were noted down. The flow rate of cooling water was changed and waited for new steady state to be reached.

2.6 Thermal parameters evaluation

The following relations were applied to evaluate the remaining performance parameters of the heat exchanger. The LMTD method was used since the inlet and outlet temperatures of both the cooling water and the secondary fluid are known.

$$Q_{hot} = C_h(T_{hi} - T_{ho}) = AUT_{LMTD} \quad (5)$$

$$Q_{cold} = C_c(T_{co} - T_{ci}) = mC_p\Delta T \quad (6)$$

Where subscripts c and h represent cold and hot fluid respectively, i and o represent for inlet and outlet respectively. Heat capacity rate for hot or cold fluid $C = mC_p$, A is the heat transfer area and U is overall heat transfer coefficient and T_{LMTD} is the log mean temperature difference.

1. Pressure drop of pentane, $P_p = P_i - P_o$
2. Cooling water temperature difference, $T_w = T_{wi} - T_{wo}$
3. Pentane Temperature difference, $T_p = T_{pi} - T_{po}$
4. Log mean temperature difference T_{LMTD} for counter flow heat exchanger:

$$T_{LMTD} = \frac{(T_{wi} - T_{po}) - (T_{wo} - T_{pi})}{\ln[\frac{(T_{wi} - T_{po})}{(T_{wo} - T_{pi})}]}$$

Where:

T_{wi} is cooling water inlet temperature; T_{wo} is Cooling water outlet temperature, T_{pi} is pentane inlet temperature, and T_{po} is pentane outlet temperature.

III. Results and discussions

Performance test of the heat exchanger (condenser) was carried out. Three parameters were studied to investigate the performance of the heat exchanger. The heat transferred, overall heat transfer coefficient and pressure drop within the shell and tube heat exchanger having secondary fluid (n-pentane) in tube side and cooling water in the shell side in counter flow configuration. Five experiments were conducted at different cooling water mass flow rates.

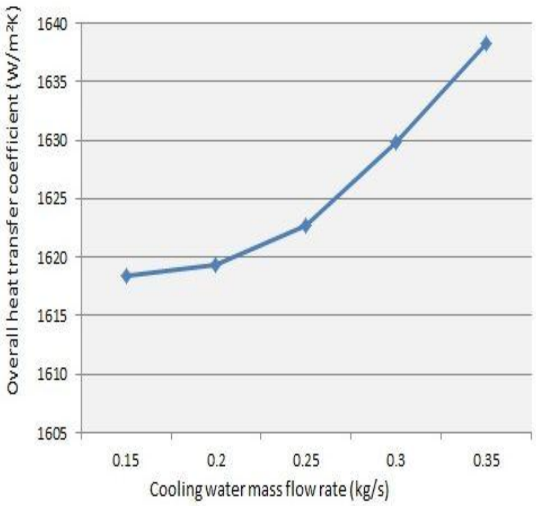

Figure 6: Variation of overall heat transfer coefficient with increase in cooling water mass flow rate

In the process, increase in the flow rate of cooling water resulted in increase in the overall heat transfer coefficient as can be seen from curve in figure 6. The curve shows that an increase of cooling water flow rate from 0.15kg/s to 0.35kg/s increased the overall heat transfer coefficient of the heat exchanger by 1.21%. This is because increase in the mass flow rate of cooing water increases the heat energy transferred. Since the specific heat remains almost constant, cooling water outlet temperature should increase to comply with law of conservation of energy and hence as the flow rate of the cooling water is increased, the tube side overall heat transfer coefficient also increases.

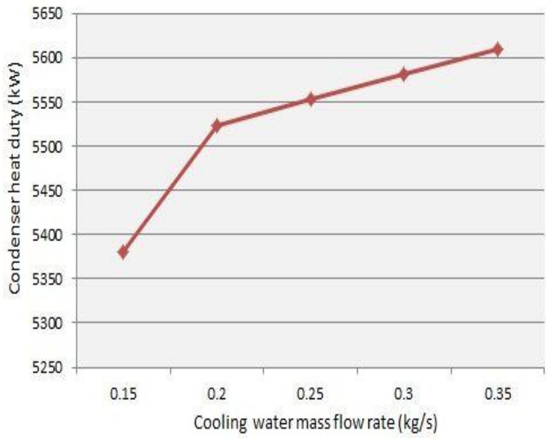

Figure 7: Variation of heat duty with increase in cooling water mass flow rate

Similarly, the variation of heat transferred (heat duty) with mass flow rate of the cooling water is shown in Figure 7. The curve shows that heat transferred increased by 4.26% with an increase of cooling water flow rate from 0.15kg/s to 0.35kg/s. This is because increase in the cooling water mass flow rate increases over all heat transfer coefficient in a lower rate than the heat energy transferred.

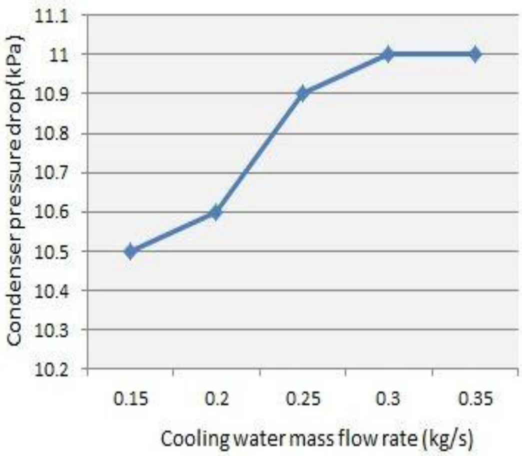

Figure 8: Variation of pressure drop with increase in cooling water mass flow rate

The curve in figure 8 shows the variation of pressure drop values with increase in cooling water mass flow rate. It was observed that the pressure drop increased from 1.5kPa to 3.5kPa in the tube side and in the shell side pressure drop increased from 1.3kPa to 2.4kPa with increase in cooling water flow rate in heat exchanger. Shell and tube heat exchangers generally experience pressure drop mainly due to friction, change in thermodynamic properties like viscosity and density through the heat exchanger as a result of heating or cooling, acceleration and deceleration of fluid with change is flow cross section etc. This pressure drop may increase pumping power and may affect the service time of structural components of the heat exchanger. However compared to rate of change of the heat transferred the rate of increase in pressure drop is reasonable.

IV. Conclusion

Shell and tube heat exchanger was designed; fabricated and experimental test was conducted to examine the performance of the heat exchanger (condenser). The performance parameters, the overall heat transfer coefficient, heat transferred and pressure drop at different cooling water mass flow rates were evaluated. The main points are summarized as follows.

- The pressure drop gradually increased with increase in cooling water flow rate in heat exchanger. This may increase pumping power which in turn decreases the efficiency of the model power plant. However, the rate of increase in pressure drop is very small compared to rate of increase in heat transfer. Hence the rate of increase in pump work is reasonably minimal.
- The results of the performance test indicated that the overall heat transfer coefficient is greater than the assumed overall heat transfer coefficient of heat exchanger in the design. This implies that the heat exchanger is thermally suitable for the evaporator of the model power plant.
- The results of the performance test revealed that the heat exchanger is working satisfactorily under standard conditions. The test results were compared with the design data and the performance parameters reasonably close to the design performance data. Thus the heat exchanger is reliable and can be applied for the evaporator of the model Organic Rankine Cycle power plant.

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