

# Performance Assessment of a Shell Tube Evaporator for a Model Organic Rankine Cycle for Use in Geothermal Power Plant

Haile Araya Nigusse, Hiram M. Ndiritu, Robert Kiplimo

Department of Mechanical Engineering, Jomo Kenyatta University of Agriculture and Technology, Nairobi, Kenya

Email: [hayhrty@gmail.com](mailto:hayhrty@gmail.com)

Received 1 September 2014; revised 30 September 2014; accepted 7 October 2014

Copyright © 2014 by authors and Scientific Research Publishing Inc.

This work is licensed under the Creative Commons Attribution International License (CC BY).

<http://creativecommons.org/licenses/by/4.0/>



Open Access

---

## Abstract

The global energy demand increases with development 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 and harmful combustion byproducts are causing global warming. The Organic Rankine Cycle power plant is a very effective option for utilization of low grade heat sources for power generation. In the Organic Rankine Cycle heat exchangers such as evaporators and condensers are key components that determine its performance. Researches indicated that shell tube heat exchangers are effectively utilized in this cycle. The design of the heat exchanger involves establishing the right flow pattern of the interacting fluids. The performance of these exchangers can be optimized by inserting baffles in the shell to direct the flow of fluid across the tubes on shell side. In this work heat exchangers have been developed to improve heat recovery from geothermal brine for additional power generation. The design involved sizing of heat exchanger (evaporator) 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 brine. The results indicated that the heat exchanger is thermally suitable for the evaporator of the model power plant.

## Keywords

Shell Tube Evaporator, Heat Transfer Co-Efficient, LMTD, ORC

---

## 1. Introduction

Energy consumption increases with growth in population. The increase in energy demand and high cost of fossil

**How to cite this paper:** Nigusse, H.A., Ndiritu, H.M. and Kiplimo, R. (2014) Performance Assessment of a Shell Tube Evaporator for a Model Organic Rankine Cycle for Use in Geothermal Power Plant. *Journal of Power and Energy Engineering*, 2, 9-18. <http://dx.doi.org/10.4236/jpee.2014.210002>

fuels is the main challenge for the developing countries. Furthermore, 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. Geothermal energy is the energy derived from the natural heat of the earth. The heat from the earth's own molten core is conducted to the adjacent rocks and transferred to underground water reservoirs by convection. The steam (hot water) heated by the geothermal heat can be tapped using different technologies. Geothermal energy is among the most reliable forms of renewable energy. Most of the other clean energy sources are weather dependent [1].

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 [2]. Moreover, 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.

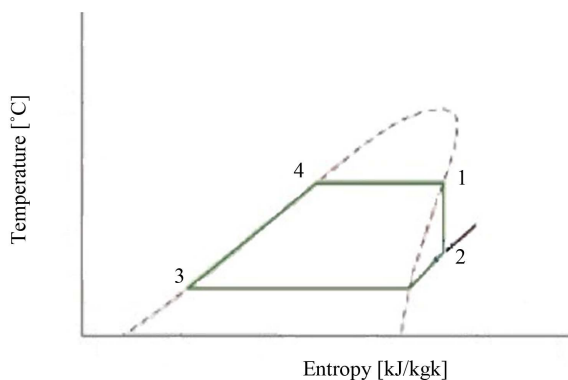
Despite the simplicity and least cost, flash steam geothermal power plants operate at lower efficiency mainly due to the untapped available energy in the brine. Discharged geothermal brine generally has a temperature higher than 100°C and mass flow rate of hundreds of tons/hour [3]. This rejected high temperature brine wastes significant amount of available energy and causes thermal pollution to the environment. Organic Rankine Cycle power plant is an advantageous technology that consents the power generation from low temperature water dominant geothermal resource and energy recovery from the geothermal brine.

The lab scale model Organic Rankine Cycle (ORC) power plant was designed to operate at a heat source temperature of 50°C to 60°C and pressure between 1 and 2.5 bars to simulate the energy recovery from the low temperature geothermal brine discharged at the reservoir of flash steam geothermal power plant. The components of a model Organic Rankine Cycle power plant for recovery of waste heat from the geothermal brine to increase power generation were fabricated and performance assessment of each component was conducted. The model power plant utilizes hot water simulating the geothermal brine and comprises of the evaporator, turbine, condenser and feed pump. Heat exchangers are the important component linking the geo-fluid and ORC.

In the cycle shown in **Figure 1**, the feed pump supplies pentane from the working fluid tank to the evaporator, where the working fluid is vaporized by heat transferred from the hot water. The high pressure vapor from the evaporator flows into the turbine and expands producing mechanical power. The low pressure vapor exhausted from the turbine is condensed by cooling water as it flows through the condenser. The condensed working fluid is directed back into the reservoir and pumped again into the evaporator and a new cycle began. **Figure 1** is the T-S diagram of the cycle and shows that the working fluid pentane is a dry expansion. The saturation curve is skewed to the right and shows expansion ends on the dry region on which no condensation happens in the turbine.

The most commonly used heat exchangers in ORC are the shell tube evaporator and condenser. Their widespread application can be attributed to ease of manufacture from a variety of materials. Moreover, there is no limit on the operating temperature and pressure [4] [5]. 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 [6].

Shell and tube heat exchangers are non-compact exchangers. Generally the heat transfer surface area per unit volume is high ranging from about 50 to 100 m<sup>2</sup>/m<sup>3</sup>. Thus, they require a considerable amount of space, support-



**Figure 1.** The T-S diagram of the cycle.

ing 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 [6].

Madhawa [7] investigated the flat plate and shell tube heat exchangers and found that flat plate type heat exchangers were effective in the evaporator and condenser when considering low-temperature heat sources in which large heat exchanger area (per unit power output) was 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, flat plate exchanger involves high manufacturing cost and maintenance cost which would affect the overall cost of the power plant.

Bambang [8] also compared the flat plate and shell tube heat exchangers and found that the shell tube type was advantageous due to simplicity in geometry, well established design procedure, could be constructed from a wide range of materials, used well-established fabrication techniques and was easily cleaned. However, the heat transfer optimization is a key challenge mainly due to large pressure drops within the shell tube heat exchangers.

The design of heat exchangers involves establishing the right flow pattern of the interacting fluids. Parallel and counter flows are the two common flow patterns in shell tube heat exchanger. The counter flow is the predominantly preferred flow direction in liquid to liquid heat exchangers since it results in a higher temperature difference [9].

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 [10]. Thundil *et al.* [11] investigated the effect of inclination of baffles in the shell by simulating a model shell and tube heat exchanger. This involved comparing 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 ( $0^\circ$ ,  $10^\circ$  and  $20^\circ$ ). They concluded that shell and tube heat exchanger with  $20^\circ$  baffle inclination angle resulted in better performance compared to  $10^\circ$  and  $0^\circ$  inclination angles.

A. Singh *et al.* [12] analyzed on the performance of a shell tube heat exchanger with segmented baffles at three different orientations ( $0^\circ$ ,  $30^\circ$  and  $60^\circ$ ). They analyzed the system for laminar flow with varying Reynolds number and concluded that the heat transfer coefficient increased with increase in Reynolds 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 increased leading to more heat transfer rate due to introduction of swirl and more convective surface area. This means that baffles are necessary components that improve performance of heat exchangers.

In general, conventional shell tube heat exchangers result in high shell-side pressure drop and formation of re-circulation zones near the baffles. To overcome the challenge, helical baffles, which give better performance than single segmental baffles, can be used. But these baffles involve high manufacturing cost, installation cost and maintenance cost. Hence the effectiveness and cost must be considered in the heat exchanger design [11].

This paper presents design, fabricate and performance assessment of the shell and tube heat exchanger designed for the evaporator of a model Organic Rankine Cycle power plant. Performance tests were conducted by relating the inlet and outlet temperatures and the overall heat transfer coefficient to the rate of heat transfer between the two fluids.

## 2. Methodology

### 2.1. Heat Exchanger Selection

Heat is transferred from one fluid to other in the heat exchanger. Heat transfer area, overall heat transfer coefficient and temperature difference are important factors to be considered [13]. For indirect heat transfer between two fluids, the shell and tube heat exchangers are more effective. In this study, a shell and tube type heat exchanger is selected for the evaporator of the model power plant due to the advantages of fairly simple geometry, can be fabricated from a wide range of materials and ease of cleaning.

The design of heat exchangers involves establishing the right flow configuration of heat exchangers. Parallel flow and counter flow are the main flow configurations in 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 is required [9]. Moreover counter flow configuration is most effective design when the desired outlet temperature of secondary fluid is between the inlet and outlet temperatures of the hot water [14].

The design of the evaporator is a fixed tube counter flow shell tube heat exchanger as shown in **Figure 2**. Moreover, the fluid flow in the exchanger is in such way that the hot water flows in the tube side and the secondary fluid (*n*-pentane) vaporizes in the shell side of the evaporator. This type of fluid flow configuration allow fouling fluid to flow through the tubes (easier to clean) and the organic working fluid (*n*-pentane) flows through the shell side where a turbulent flow is obtained due the baffles. Turbulence eddies are induced due to recirculation near the baffles and which would result in more pressure drop with 0° baffle orientation [11].

Pentane has been selected for the working fluid of the model power plant due to its convenience for thermodynamic cycle working between normal atmospheric condition and the boiling condition of water. Moreover pentane is environmentally friendly (non-ozone depleting) and lower global warming potential organic fluid. Pentane is liquid state at atmospheric conditions (minimizes handling cost) and has good performance for ORC. The thermodynamic properties of pentane are acquired from a database program called NIST WebBook (NIST standard Reference Database Number 69) [15].

## 2.2. Design and Construction of the Evaporator

In design of 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 shown in **Table 1**. The following equations have been used to relate 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.

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

where heat capacity rate for hot or cold fluid,  $C = mC_p$ . Where  $C_p$  is specific heat capacity at constant pressure and  $m$  is the mass flow rate of working fluid.

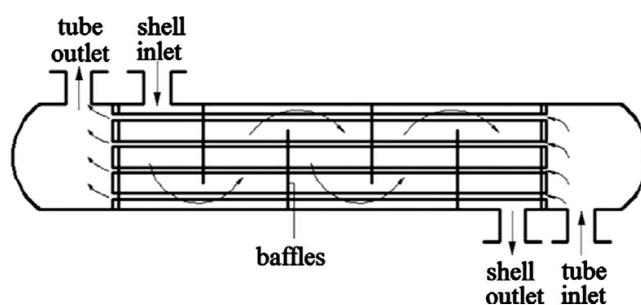
During the heat transfer process energy is conserved therefore for the cold and hot fluid;

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

where heat capacity rate for hot or cold fluid  $C = mc_p$ , 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  $\Delta T_{\text{LMTD}}$  by:

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



**Figure 2.** Counter flow shell and tube heat exchanger [4].

**Table 1.** Flow conditions of the fluid streams in the heat exchanger.

Sr. No	Description	Tube side	Shell side
1	Mass flow rate (kg/s)	0.1	0.12
2	Inlet temperature (°C)	54	25
3	Outlet temperature (°C)	36	51

where  $A$  is heat transfer surface area and  $\Delta T_{LMTD}$  is the log mean temperature difference. The log mean temperature difference of the heat exchanger can be determined as follows:

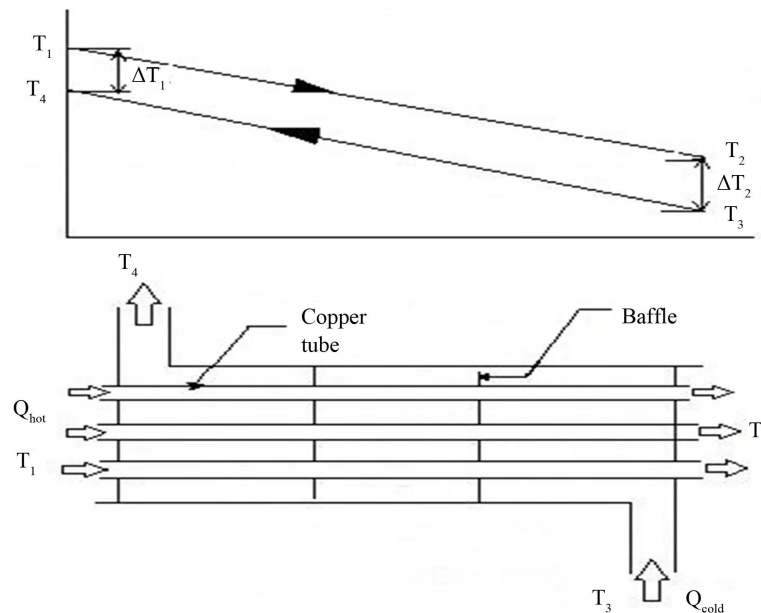
$$T_{LMTD} = \frac{(\Delta T_1 - \Delta T_2)}{\ln \left[ \frac{(\Delta T_1)}{(\Delta T_2)} \right]} = \frac{(T_1 - T_4) - (T_2 - T_3)}{\ln \left[ \frac{(T_1 - T_4)}{(T_2 - T_3)} \right]} \quad (4)$$

where,  $\Delta T_1$  and  $\Delta T_2$  represent the temperature differences at the inlet and exit of the heat exchanger respectively as indicated in **Figure 3**.

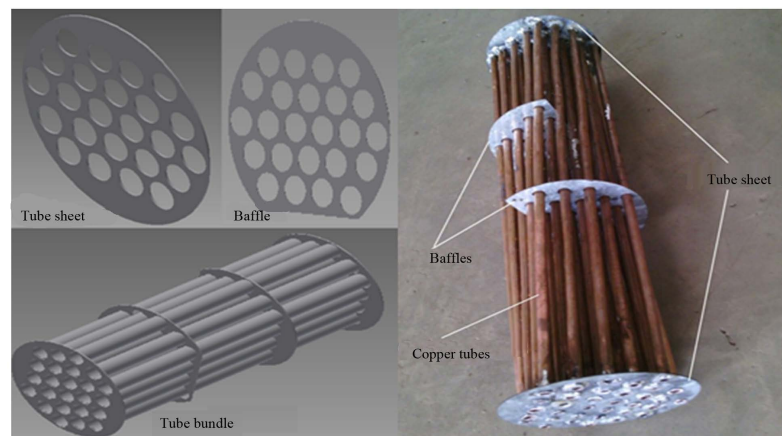
The heat exchanger was designed using the design Equations (1)-(4). The details of the designed shell and tube heat exchanger are shown in **Table 2**. **Figure 4** shows the assembly of the copper tubes to the tube sheet and the baffles placed on the respective position.

Evaluation of the thermal parameters:

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 hot water and the secondary fluid are known.



**Figure 3.** Flow configuration of the evaporator.



**Figure 4.** The shell and tube heat exchanger.

**Table 2.** Dimensions and parameters of the heat exchanger.

Sr. No	Description	Tube side	Shell side
1	Interacting fluids	Hot water	Pentane
2	Number of tubes	33	
3	Total tube length (m)	13.1	
4	Tube diameter (outer/inner) (mm)	12.7/10.9	150
5	Heat transfer area (m <sup>2</sup> )	0.41	
6	Number of pass	1	
7	Tube configuration	30° triangular	
8	Tube pitch (mm)	16	
9	Material	Copper	Galvanized steel

1) Heat duty ( $Q$ )

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

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

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  $T_{\text{LMTD}}$  is the log mean temperature difference.

2) Cooling water pressure drop,  $\Delta P_w = P_i - P_o$

3) Pressure drop of pentane,  $\Delta P_p = P_i - P_o$

4) Cooling water temperature difference,  $\Delta T_w = T_{wi} - T_{wo}$

5) Pentane temperature difference,  $\Delta T_p = T_{pi} - T_{po}$

6) Log mean temperature difference  $T_{\text{LMTD}}$  for counter flow heat exchanger:

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

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.

### 2.3. Fabrication of the Heat Exchanger

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 471 mm by 750 mm and four holes were drilled for the inlet and outlet of the two fluids before rolling the plate. After rolling a shell of 150 mm diameter by 750 mm length was obtained. The copper tubes were held at the ends by means of galvanized iron sheets as shown in **Figure 4**.

Two baffles were also provided as shown with a baffle pitch of 150 mm. The optimum baffle pitch (spacing between segmental baffles) and the baffle cut were used 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 of the inside shell diameter is commonly used [16].

Copper tubes were cut at a length of 400 mm 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 then inserted to the main shell and the two tube sheets were brazed using gas welding to provide air tight joints.

## 2.4. Heat Exchanger Leakage Test

It is important to ensure there is no leakage of fluids of the heat exchanger at the operating temperature and pressure before carrying out a performance test. The heat exchanger was tested for leakages using water both for the shell side and tube side. Water pumped at a pressure of 2.5 bars was used for this test. Any leaking sections (particularly at the welded joints) were sealed using a sealing material called STAG. STAG is easy break, smooth consistency and lead-free joining compound. It was applied to the evaporator since it is non-poisonous and does not react with pentane and water.

## 2.5. Performance Test of the Heat Exchanger

The performance test unit consists of an overhead hot 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 hot water, produced by an electric heater in the hot water tank, flows by gravity through the heat exchanger in the tube side. Hot water flow rate is controlled by opening a flow valve HV1.

The feed pump supplies the secondary fluid (*n*-pentane) to the exchanger through the flow meter (FM1) and evaporates on the shell side in a counter-current to the hot water flow. The vapor which evaporated in the heat exchanger is then directed back to the secondary fluid tank through the condenser.

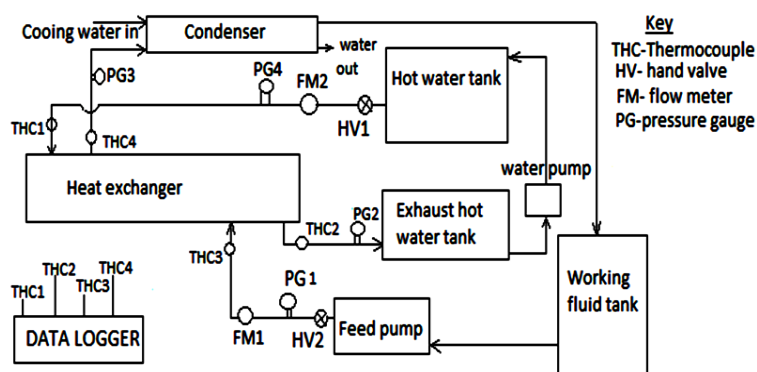
The length of one test section is set to be two minutes. A set of four thermocouples are provided to record pertinent temperatures at the inlet and exit of each fluid as shown in the **Figure 5**. The thermocouples are connected to a data logger called TDS-530. The data logger was set to print each temperature in a test section of two minutes. A set of five measurements was taken by varying the hot water mass flow rate and maintaining the secondary fluid (*n*-pentane) flow rate constant. The temperature indicators on the data logger displayed the temperatures of hot water at the inlet ( $T_1$ ), hot water at the exit ( $T_2$ ), secondary fluid at the inlet ( $T_3$ ) and secondary fluid at the exit ( $T_4$ ) of the heat exchanger.

## 2.6. Experimental Procedures

The overhead hot water tank was filled with water. The three kilowatt water heater built in the hot water tank was switched on and temperature was set to 54°C. The inlet valve HV1 was opened and hot water was allowed into the tube side of the heat exchanger. The feed pump was switched on and the secondary fluid (*n*-pentane) allowed to flow into the shell side of the heat exchanger. A steady state of the measurements was attained in 36 minutes and all the four temperatures and flow rates of hot water and secondary fluid was recorded. The flow rate of hot water was changed and waited for new steady state to be reached.

## 3. Results and Discussion

Performance test of the heat exchanger was carried out. Three parameters were studied to understand the performance of the heat exchanger. The heat transferred, overall heat transfer coefficient, tube side and shell side pressure drop within the exchanger. The shell tube heat exchanger was having hot water in tube side and



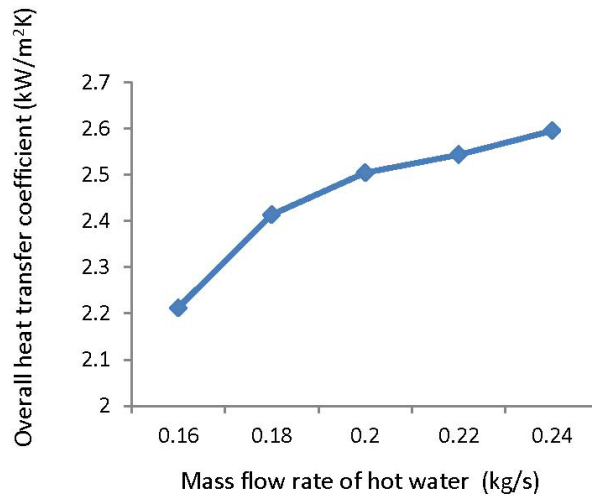
**Figure 5.** Schematic sketch of the evaporator performance assessment experiment.

secondary fluid (*n*-pentane) in the shell side in counter flow configuration. Five experiments were conducted at different hot water mass flow rates.

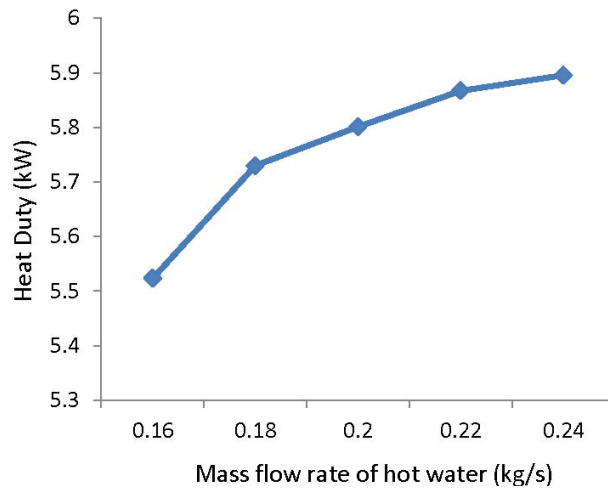
Increase in the flow rate of hot water resulted in increase in the overall heat transfer coefficient as can be seen from curve in **Figure 6**. The curve shows that an increase of hot water flow rate from 0.16 kg/s to 0.24 kg/s increased the overall heat transfer coefficient of the heat exchanger by 17.33%. This is because increase in the mass flow rate of hot water increases the heat energy transferred. Since the specific heat remains almost constant, hot water outlet temperature should decrease to comply with law of conservation of energy and hence as the flow rate of the hot water is increased, the tube side overall heat transfer coefficient also increases. The performance test result indicated that the developed shell and tube heat exchanger performs satisfactorily under standard conditions and the variation of the overall heat transfer coefficient and total heat transferred with the mass flow rate of the hot water is analogous to similar heat exchangers.

Similarly, the variation of heat transferred with mass flow rate of the hot water is shown in **Figure 7**. The curve shows that heat transferred increased by 6.74% with an increase of hot water flow rate from 0.16 kg/s to 0.24 kg/s. This is because increase in the hot water mass flow rate increases overall heat transfer coefficient in a faster rate than the heat energy transferred.

**Figure 8** shows the variation of tube side and shell side pressure drop values with increase in hot water mass

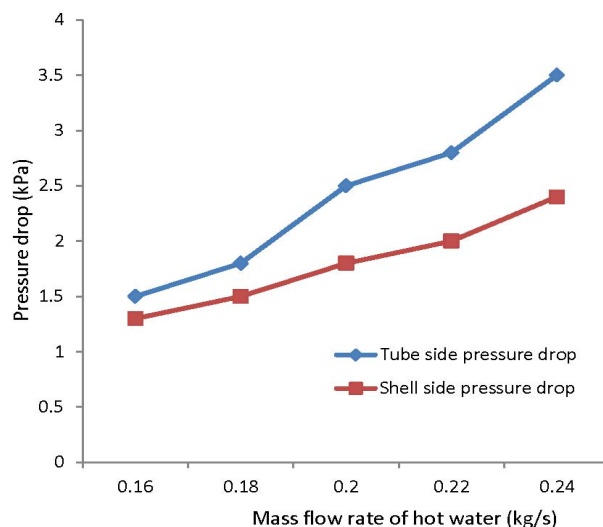


**Figure 6.** Variation of the overall heat transfer coefficient with increase in hot water mass flow rate.



**Figure 7.** Variation of heat duty with increase in hot water mass flow rate.





**Figure 8.** Variation of the pressure drop with increase in hot water mass flow rate.

flow rate. It was observed that the pressure drop increased by more than twice in the tube side and in the shell side pressure drop increased by 84.6% with increase in hot 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 in flow cross section.

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.

#### 4. Conclusions

Shell and tube heat exchanger was designed and fabricated. Experimental test was conducted to study the performance of the heat exchanger. The performance parameters, the overall heat transfer coefficient, heat transferred and tube side pressure drop and shell side pressure drop at different hot water mass flow rates were evaluated. The main points are summarized as follows.

- The results of the performance test indicated that the overall heat transfer coefficient was greater than the assumed overall heat transfer coefficient of heat exchanger. 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 was 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.

#### References

- [1] Mburu, M. (2009) Geothermal Energy Utilization. Exploration for Geothermal Resources.
- [2] (2013) 2013 Geothermal Power: International Market Overview.
- [3] Teguh, P.B., *et al.* (2011) Design of n-Butane Radial Inflow Turbine for 100 kW Binary Cycle Power Plant. *International Journal of Engineering & Technology*, **11**, 55.
- [4] Hadidi, A., Hadidi, M. and Nazari, A. (2013) A New Design Approach for Shell-and-Tube Heat Exchangers Using Imperialist Competitive Algorithm (ICA) from Economic Point of View. *Energy Conversion and Management*, **67**, 66-74. <http://dx.doi.org/10.1016/j.enconman.2012.11.017>
- [5] Sairam, V., Siddarath, B., Saiprasad, C., Taruns, S. and Sujit, G. (2014) Design, Fabrication and Testing of FRP Shell Counter-Flow Heat Exchanger. *International Journal of Engineering Science and Innovative Technology*, **3**, 171-176.

- [6] Shah, R.K. and Sekulic, D.P. (2012) Fundamentals of Heat Exchanger Design. University of Kentucky, Lexington.
- [7] Hettiarachchi, H.D.M., *et al.* (2007) Optimum Design Criteria for an Organic Rankine Cycle Using Low-Temperature Geothermal Heat Sources. *Energy*, **32**, 1698-1706. <http://dx.doi.org/10.1016/j.energy.2007.01.005>
- [8] Teguh, P.B. and Trisno, M.D. (2011) Model of Binary Cycle Power Plant Using Brine as Thermal Energy Sources and Development Potential in Sibayak. *International Journal of Electrical & Computer Sciences*, **11**, 45.
- [9] Kuppan, T. (2000) Heat Exchanger Design Handbook. Marcel Dekker, Inc., New York.
- [10] Incropera, F.P. and Dewitt, D.P. (2006) Introduction to Heat Transfer. New York.
- [11] Thundil, R. (2012) Shell Side Numerical Analysis of a Shell and Tube Heat Exchanger Considering the Effects of Baffle Inclination Angle on Fluid Flow. *International Journal of Thermal Science*, **16**, 1165-1174. <http://dx.doi.org/10.2298/TSCI110330118R>
- [12] Singh, A. and Sehgal, S.S. (2013) Thermohydraulic Analysis of Shell-and-Tube Heat Exchanger with Segmental Baffles. *ISRN Chemical Engineering*, **2013**, Article ID: 548676.
- [13] Kumar, U., Karimi, M.N. and Agrawal, B.K. (2013) Optimization and Selection of Organic Rankine Cycle for Low Grade Heat Recovery by Using Graph Theoretical Approach. *International Journal of Sustainable Development and Green Economics*, **2**, 11-16.
- [14] Pandey, A. (2011) Performance Analysis of a Compact Heat Exchanger. Master's Thesis, National Institute of Technology Rourkela, Rourkela.
- [15] NIST Chemistry WebBook. <http://webbook.nist.gov/chemistry/>
- [16] Patel, S.K. and Mavani, A.M. (2012) Shell and Tube Heat Exchanger Thermal Design with Optimization of Mass Flow Rate and Baffle Spacing. *International Journal of Advanced Engineering Research and Studies*, **2**, 130-135.

## Nomenclatures

$A$ : heat transfer area  
 $n$ : number of tube  
 $T$ : temperature  
 $h$ : enthalpy  
 $C$ : specific heat capacity  
 $\dot{m}$ : mass flow rate  
 $\dot{Q}$ : the rate of heat transfer  
 $d$ : diameter  
 $k$ : thermal conductivity  
 $R$ : thermal resistance

## Greek Letters

$\Delta$ : change in  
 $U$ : overall heat transfer coefficient

## Subscripts

$tb$ : tube  
 $h$ : hot fluid  
 $c$ : cold fluid  
 $i$ : inlet to the system  
 $o$ : outlet from the system  
LMTD: logarithmic mean temperature difference  
 $cf$ : counter flow  
 $p$ : pentane  
 $w$ : water