THEORETICAL MODEL OF NATURAL CIRCULATION FLOW AND HEAT TRANSFER WITHIN ONE-ENDED INCLINED PIPE

by

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The most commonplace natural flow inside one-ended inclined pipes today is water heating systems. In this study, a model was created for the estimation of the pipe outlet temperature of the fluid with the energy balance for the inside of the tank, flow rate calculation of natural circulation, and other thermal calculations in a one-ended inclined pipe. In addition, this model has been compared with the Li model in the literature, and it is easier and more successful. 100 W, 200 W, 400 W, 600 W, and 800 W thermal power was applied to a one-ended inclined pipe, and the temperature values were recorded in five minute periods in the experiments that lasted a total of six hours. As the average of the experiments, the estimation results for the current model and the Li model are: the average percent relative error rates are 6.02 and 15.1 and the coefficient of determination, R^2, are 0.9865 and 0.9683, respectively.

Key words: *natural flow, temperature prediction, one-ended inclined pipe flow*

Introduction

A natural circulation system operates according to natural laws like gravity and buoyancy. The difference in density caused by heating and cooling in the system allows the flow to start automatically. This system does not require the use of any machine, also the maintenance and operating costs of the system are low. The most commonplace natural flow inside one-ended inclined pipes today is water heating systems.

Thermosyphon or natural circulation solar water heating systems are the simplest and most widely used solar energy collection and utilization devices. They are widely used in Australia and Israel and are gaining popularity in Japan, the USA, and elsewhere [1]. It consists of a collector, storage tank, and connecting pipes [2].

There are many various studies in the literature on natural flow in the pipe [3-8]. For instance, Li *et al*. [3] describe a numerical and theoretical study on a horizontal single-ended evacuated tube. The results showed that the secondary flow had a significant influence on the natural circulation flow rate and the temperature distribution within the tube. Pleshanov *et al*.

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[4] describe the research and design of a fluidized bed steam boiler natural circulation circuit. There are two variants of research results of the designed circulation circuit. The first circulation circuit variant was designed as a complex circuit. In the second variant, the flow is divided into independent sections. Results demonstrate that the specific quantity of metal of complex and simple circuit variants is approximately the same with equal reliability. Bejjam and Kumar [5] developed and simulated a 3-D geometry of a natural circulation loop (NCL) by using the software, ANSYS FLUENT 14.5 The results showed that the present 3-D numerical model could be useful to estimate the performance of a NCL. Riahi and Taherian [6] have tested a natural circulation closed thermosyphon flat plate solar water heater. As a result, it was found that such systems can provide ample energy to satisfy the demand for hot water, contrary to the misperception among locals. Bocanegra *et al*. [7] investigated the different cooler-heater configurations (vertical and horizontal) in the NCL. Misale *et al*. [8] presented an experimental study regarding the natural circulation in connected vertical rectangular loops. The results obtained with two NCL characterized by different internal diameters could be considered as the first attempt to try optimizing several connected NCL. Apart from these studies, there are also studies in the literature, for instance, using the 2-D velocity field PIV in a cylindrical vessel with a natural thermal convection system [9].

Although the theoretical and experimental studies [10-15] of the heat transfer and the natural convection in a single-ended tube can date the back to early 1950's, they mainly focused on the vertical and inclined tubes. The buoyancy force in the vertical tube and a component of the force in the inclined tube are parallel with the tube axis. It drives the free convection to flow along the tube axis. However, the buoyancy force in the horizontal tube is perpendicular to the tube axis. The free convection is driven to flow around the tube circumference. Thus, the theoretical models of the vertical or inclined tube are inappropriate for the horizontal tube [3].

In this study, it is aimed to estimate the pipe outlet temperature of the fluid, calculate the mass-flow rate of natural circulation, and other thermal calculations in a one-ended inclined pipe. With the model created within the scope of the study, the pipe outlet temperature of the fluid was easier and more successful than the Li *et al*. model [3]. In addition, it was questioned whether stopping points were formed in the inclined pipe in the experiments carried out at different thermal power, and thus, information about the continuity of the natural circulation throughout the experiment was obtained. Considering that it is very important to create practical models for such complex problems, we believe that this study will contribute to the literature.

Material and methods

The experimental set-up is shown in fig. 1. In the experimental set-up, the locations of the thermocouples placed in the tank, tank inlet, and inclined pipe are shown in fig. 2. In this experimental set-up, the heat transfer oil (Mobilterm-605) in the inclined galvanized pipe is heated by rod resistances and the power given by the boxed variac can be adjusted. The heat transfer oil is heated in

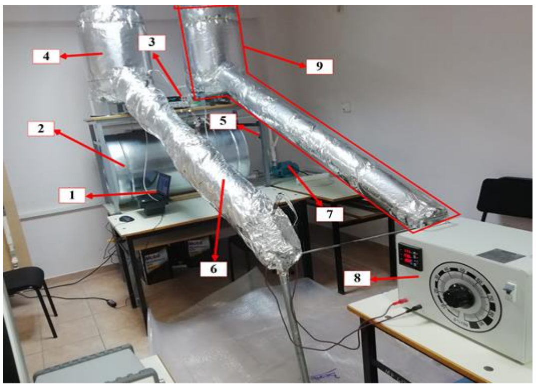

Figure 1. The experimental setup; *1 – PC, 2 – cooling water tank, 3 – datalogger, 4 – oil tank, 5 – flowmeter, 6 – inclined pipe, 7 – centrifugal pump, 8 – boxed variac, and 9 – unused spare part*

the galvanized inclined pipe and can naturally be directed to the low-temperature tank. This tank is made of a 2 mm galvanized sheet and insulated with 10 cm thick glass wool. The galvanized inclined pipe is insulated with 20 cm thick glass wool. The heat in the heat transfer oil can be transferred to the water utilizing the copper coil placed inside the tank. By using a centrifugal pump, the cooling water is circulated through the coil. However, the cooling system was not operated in this study.

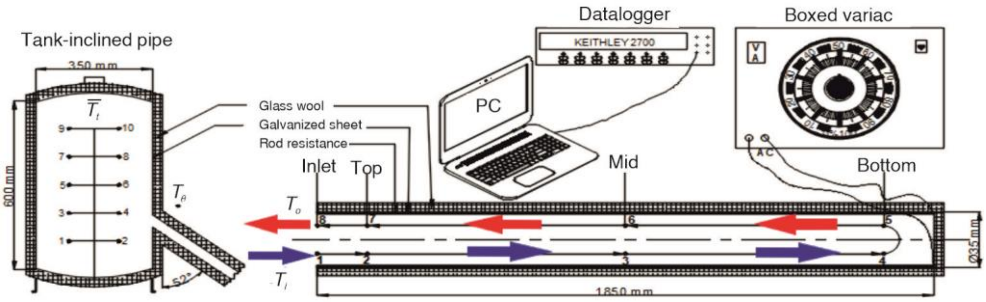

Figure 2. The schematic diagram of the experimental set-up

The measurement set-up consists of a 5 kVA monophase boxed variac, a Keithley 2700 data logger with an uncertainty of ±0.15%, and a computer. There is a digital sensitive multimeter display on which the current and voltage values can be read on the boxed variac. In the experiments, the uncertainty of the thermal power applied to the one-ended inclined pipe was determined using the Kline and McClintock methods [16]. accordingly, the uncertainty remains below 3%.

The schematic view of the experimental set-up is given in fig. 2, it is thought that a streamline passes at the points numbered from 1 to 8 within the one-ended inclined pipe. Experimental temperature values at these points are known. The temperature values at points 1 and 8, which are close to the tank entrance, are respectively, taken as the pipe inlet and outlet (or tank outlet and inlet) temperature of the heat transfer oil. In addition, the average temperature of the 10 thermoelements in the tank was calculated as the average temperature of the tank. All temperature measurements were made with a Keithley 2700 multimeter/data acquisition system and fiberglass insulated *K*-type thermocouple wire was used.

Questioning the existence of a natural circulation

In Lighthill's study [11], a modified parameter given in eq. (1) is proposed that predicts the existence of a stagnant region for inclined pipes:

$$\tau = \mathrm{Ra}r\cos(\theta)/L \tag{1}$$

where Ra is the Rayleigh number and depends on tube radius, wall, and tube axis temperature and L – the pipe length. According to Lighthill [11], if a stagnant region occurs τ is the critical number.

When the known values for the current study are substituted in eq. (1):

$$\tau = \mathrm{Ra}r\cos(\theta)/L \Rightarrow 350 = \mathrm{Ra}0.0175\cos(52°)/1.85 \tag{2}$$

Here, the critical Rayleigh number is found as 60097.96. If the calculated Rayleigh number in our current study is below this value, a stagnant region will occur, if it is equal to or above this value, no stagnant region will occur, and the existence of natural circulation will be proven.

Benhia and Morrison [15] stated that while determining the stagnant region in a natural flow, they calculated the Rayleigh number according to the temperature difference between the pipe heating jacket and the tank, unlike Lighthill. In the present study, Rayleigh numbers were calculated with eq. (3) according to the lower and upper fluid temperature difference of the pipe section to question the existence of natural circulation in the inclined pipe and whether any stagnant region will occur:

$$\mathrm{Ra} = \frac{\mathrm{g}\beta\Delta T D^3}{\nu\alpha} \tag{3}$$

Rayleigh numbers at the tank entrance, top, middle, and bottom of the inclined pipe were calculated for each experiment.

Development of a theoretical model

A model has been created to calculate the flow rate of the natural circulation and other thermal calculations in the one-ended inclined pipe-tank system. The assumptions made while creating the model are:

- In the pipe section, the internal and external flow cross-sectional areas are equal.
- A streamline passes at the points numbered from 1 to 8 within the one-ended inclined pipe (fig. 2).
- The mass-flow, $\dot{m}$, along the pipe axis is uniform.
- The thermal losses of the pipe are neglected.

The main purpose of this study is to estimate the temperature of the fluid at the inclined pipe outlet (tank inlet).

Mass-flow and energy calculations

The flow rate is assumed to be uniform in one-ended inclined pipe, and eq. (4) can be written:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = \dot{m}_5 = \dot{m}_6 = \dot{m}_7 = \dot{m}_8 = \dot{m} \tag{4}$$

The properties of Mobiltherm605 heat transfer oil used in the experiments were available and limited to a few temperature values in the product catalog [17]. The properties of the oil for the operating temperature range were obtained from the relevant company. Equation (5), which gives the specific heat depending on the temperature, was obtained by using these data. Similarly, the temperature-dependent density equation is obtained in eq. (6):

$$c_p = 3.6567T + 817.46 \tag{5}$$

$$\rho = -0.4687T + 992.17 \tag{6}$$

In the calculations, the mass-flow rate was found in two different ways and its average was used.

As a first way to find the mass-flow rate, eq. (7) for energy balance along the streamline can be written. Here, the necessary simplifications are made, and the mass-flow rate is found by eq. (8):

$$\dot{q}_{12} + \dot{q}_{23} + \dot{q}_{34} + \dot{q}_{45} + \dot{q}_{56} + \dot{q}_{67} + \dot{q}_{78} = \dot{q} \tag{7}$$

$$\begin{gathered}\left(\dot{m}_2 c_{p2} T_2-\dot{m}_1 c_{p1} T_1\right)+\left(\dot{m}_3 c_{p3} T_3-\dot{m}_2 c_{p2} T_2\right)+\left(\dot{m}_4 c_{p4} T_4-\dot{m}_3 c_{p3} T_3\right)+ \\ +\left(\dot{m}_5 c_{p5} T_5-\dot{m}_4 c_{p4} T_4\right)+\left(\dot{m}_6 c_{p6} T_6-\dot{m}_5 c_{p5} T_5\right)+\left(\dot{m}_7 c_{p7} T_7-\dot{m}_6 c_{p6} T_6\right)+\left(\dot{m}_8 c_{p8} T_8-\dot{m}_7 c_{p7} T_7\right)=\dot{q} \\ \dot{m}_8 c_{p8} T_8-\dot{m}_1 c_{p1} T_1=\dot{q} \Rightarrow \dot{m} c_{p8} T_8-\dot{m} c_{p1} T_1=\dot{q} \Rightarrow \dot{m}\left(c_{p8} T_8-c_{p1} T_1\right)=\dot{q}\end{gathered} \tag{8}$$

Here, it is seen that it is sufficient to consider the inlet-outlet points of the inclined pipe near the tank entrance to find the mass-flow rate.

As a second way to find the mass-flow rate, the energy difference between the upper part and the lower part of the inclined pipe can be equalized to the heat supplied to the pipe as stated in eq. (9). Here, the necessary simplifications are made, and the mass-flow rate is found by eq. (10):

$$\dot{q}_{85}+\dot{q}_{41}=\dot{q} \tag{9}$$

$$\begin{gathered}(\dot{m}_8 c_{p8} T_8-\dot{m}_5 c_{p5} T_5)+(\dot{m}_4 c_{p4} T_4-\dot{m}_1 c_{p1} T_1)=\dot{q} \Rightarrow \dot{m}(c_{p8} T_8-c_{p5} T_5+c_{p4} T_4-c_{p1} T_1)=\dot{q} \\ \dot{m}=\dot{q} /\left[c_{p8} T_8-c_{p5} T_5+c_{p4} T_4-c_{p1} T_1\right]\end{gathered} \tag{10}$$

The average of the mass-flow rates found in two different ways was used in the calculations.

The heat value $\dot{q}$ is also equal to the sum of the change in the energy of the oil in the tank and the heat loss in the tank system. This equation is given in eq. (11). When necessary corrections are made with eq. (12) and eq. (13), eq. (14) is obtained, and the heat transfer coefficient of the tank is found:

$$\dot{q}=\Delta\dot{E}_t+\dot{q}_{t-\text{lost}} \tag{11}$$

$$\Delta\dot{E}_t=\frac{m_t c_{p,t}\Delta\bar{T}_t}{\Delta t} \tag{12}$$

$$\dot{q}_{t-\text{lost}}=\dot{q}-\Delta\dot{E}_t=\dot{q}-\left[\frac{m_t c_{p,t}\Delta\bar{T}_t}{\Delta t}\right]=u_{t-\text{lost}} A_t(\bar{T}_t-T_\infty) \tag{13}$$

$$u_{t-\text{lost}}=\frac{\dot{q}_{t-\text{lost}}}{A_t(\bar{T}_t-T_\infty)}=\frac{\dot{q}-\left[\dfrac{m_t c_{p,t}\Delta\bar{T}_t}{\Delta t}\right]}{A_t(\bar{T}_t-T_\infty)} \tag{14}$$

Estimating the pipe outlet temperature of the fluid with the energy balance for the inside of the tank

The energy balance for the interior of the tank can be written as in eq. (15):

$$\rho_t c_{p,t}\forall_t\frac{\Delta\bar{T}_t}{\Delta t}=\dot{Q}_i-\dot{Q}_o-\dot{Q}_{\text{lost}}=\dot{m}_i c_{p,i} T_i-\dot{m}_o c_{p,o} T_o-\bar{u}_{t-\text{lost}} A_t\left(\bar{T}_t-T_\infty\right) \tag{15}$$

Here, the density and specific heat values can be written in terms of the temperature of heat transfer fluid, eqs. (2) and (3).

The volume of the tank is $\forall_t=0.05\ \text{m}^3$, the heat transfer coefficient is $\bar{u}_{t-lost}=2.77$ W/m^{2}K (this value was obtained experimentally and is the average of all cases), and the surface area is $A_t=0.852\ \text{m}^2$. The inlet and outlet flow rates are equal, $\dot{m}_i=\dot{m}_o=\dot{m}$.

When eq. (15) is arranged, the following eq. (16) is obtained:

$$3.6567T_i^2+817.46T_i=\frac{\dot{m}\left(3.6567T_o+817.46\right)T_o+\bar{u}_{t-\text{lost}}A_t(\bar{T}_t-T_\infty)+\left(-0.4687\bar{T}_t+992.17\right)\left(3.6567\bar{T}_t+817,46\right)\forall_t\Delta\bar{T}_t/\Delta t}{\dot{m}} \tag{16}$$

Using this equation, the temperatures T_i of the heat transfer oil coming out of the pipe and entering the tank were calculated for each experiment and period.

In the literature, in the study of Li *et al*. [3], a theoretical model was developed that calculates the secondary flow to predict the natural circulation flow and axial temperature distribution in a horizontal and one-ended tube. The assumptions made while developing this model are:

- In the tube cross-section, the inflow and outflow cross-sectional areas are half and equal.
- The mass-flow is uniform along the tube axis.
- The temperature of the fluid entering the lower section of the tube is equal to the temperature of the tank.
- The heat loss of the tube is neglected.

In the current model, except for item c, other assumptions are accepted as they are. The temperature of the flow at the entrance to the pipe (outlet from the tank) is taken at point 1 in fig. 2, close to the entrance of the pipe to the tank. In this study, with a simpler model than the Li model. By using the energy balance for the inside of the tank, the pipe outlet temperature of the fluid and the mass-flow rate of the fluid were estimated using the energy equations in the pipe.

In the Li model, by using the flow equations in the horizontal tube, the driving pressure of the circulation flow is equalized to the friction pressure, and the mass-flow rate of the fluid and the pipe outlet temperature are found as follows:

$$\dot{m}_o=\frac{\left(\rho_i-\rho_o\right)\left(L-x\right)d_e^2\rho_t\pi D^3\text{g}}{256\alpha\left(\mu_o+\mu_i\right)L^2} \tag{17}$$

$$T_o=\frac{256\alpha\left(\mu_o+\mu_i\right)L^2q_s}{c_p\left(\rho_i-\rho_o\right)d_e{}^2\rho_t\pi D^2\text{g}}+T_t \tag{18}$$

Since the Li model was developed for the horizontal pipe, the assumptions in this model were emphasized to compare with the existing model. In the current model, the value of the flow in the lower part of the pipe section close to the tank inlet is taken as the inlet temperature of the pipe, and in the Li model, this value is taken instead of the tank temperature, which gives a more successful result. Therefore, the angle of inclination does not matter when comparing these two models.

Results and discussions

Rayleigh numbers at the tank entrance, top, middle, and bottom of the inclined pipe for each experiment are given in fig. 3.

The following inferences can be made from these graphs:

- For each test, the Rayleigh number in the tank entrance, upper and middle part of the inclined pipe was below the critical value at the beginning of the test, but as soon as the temperature started to increase, it rose above this value and natural convection currents started. There was no stagnant point during the experiment.

- For each test, the Rayleigh number at the bottom of the inclined pipe was lower than the other parts, and a stagnant point was formed by falling below the critical value from time to time throughout the experiment. The reason for this is that the fluid at the bottom of the pipe is at the farthest distance from the tank inlet, and it is more difficult for the fluid to overcome friction and gravity than at the top.
- In the experiments carried out at 100, 200, and 400 W thermal power, the Rayleigh number at the tank entrance of the inclined pipe was the lowest value after the bottom of the pipe. This situation was maintained for the first three hours and two hours, respectively, for the experiments performed at 600 and 800 W thermal power. In the continuation of the experiment, the Rayleigh number at the top of the inclined pipe decreased and the Rayleigh number at the tank entrance increased and exceeded this value. This has been observed especially in experiments performed at high thermal power (fig. 3d and 3e). The Rayleigh number tends to fluctuate throughout the experiment compared to tests at lower thermal power (fig. 3a-3c). The reason for this is that fluid molecules move randomly and unsteadily, as sudden temperature changes occur at high thermal powers.
- The Rayleigh number in the middle of the inclined pipe is highest in almost all experiments. Because the flow circulation is faced with the obstruction of the fluid mass in the tank at the entrance of the tank and the difficulty of returning to the tank at the bottom of the pipe. In the middle part, the flow circulation will find its way when it overcomes these obstacles.
- While the Rayleigh number followed a smooth profile during the experiments at low thermal power such as 100, 200, and 400 W, it followed a turbulent profile in the experiments performed at high thermal power such as 600 and 800 W. This situation is thought to be caused by the sudden temperature changes in the flow in the high thermal power inclined pipe and the irregular motion of the molecules.

The flow values calculated according to the data in the experiments are given in fig. 4. Accordingly, the average flow rates in the experiments with 100 W, 200 W, 400 W, 600 W, and 800 W thermal power, respectively; were 10.81 kg/h, 16.85 kg/h, 31.21 kg/h, 35.54 kg/h, and 34.65 kg/h. In experiments at low thermal power, the flow rate is lower and changes less throughout the experiment. In high thermal power experiments; the flow rate is higher and varies widely throughout the experiment. This situation is thought to be caused by the sudden temperature changes in the flow in the high thermal power inclined pipe and the irregular motion of the molecules.

The average percent relative error rates and the coefficient of determination, R^2, of the estimation results obtained by taking the temperature values [°C] for the current model and the Li model are given in tab. 1 for the experiments performed at different thermal powers. These results are in tab. 1 was obtained as:

- First, the pipe outlet temperatures (point 8 in fig. 2) were estimated for each step (5 minutes) of the experiments for the current model and the Li model.
- Then, by using these temperature values, the percent relative error rates and the coefficients of determination are calculated according to the experimental data.
- Finally, the average of these values was added to the table.

The energy is given to the tank and the lost energy ratios are given in fig. 5. Accordingly, the percentage of lost energy in experiments with 100 W, 200 W, 400 W, 600 W, and 800 W thermal power, respectively; 13, 22.5, 18.75, 25, and 27.5. It is seen that as the thermal power in the experiment increases, the energy loss rate increases. The average heat transfer coefficient of the tank was experimentally found to be 2.77 W/m^{2}K.

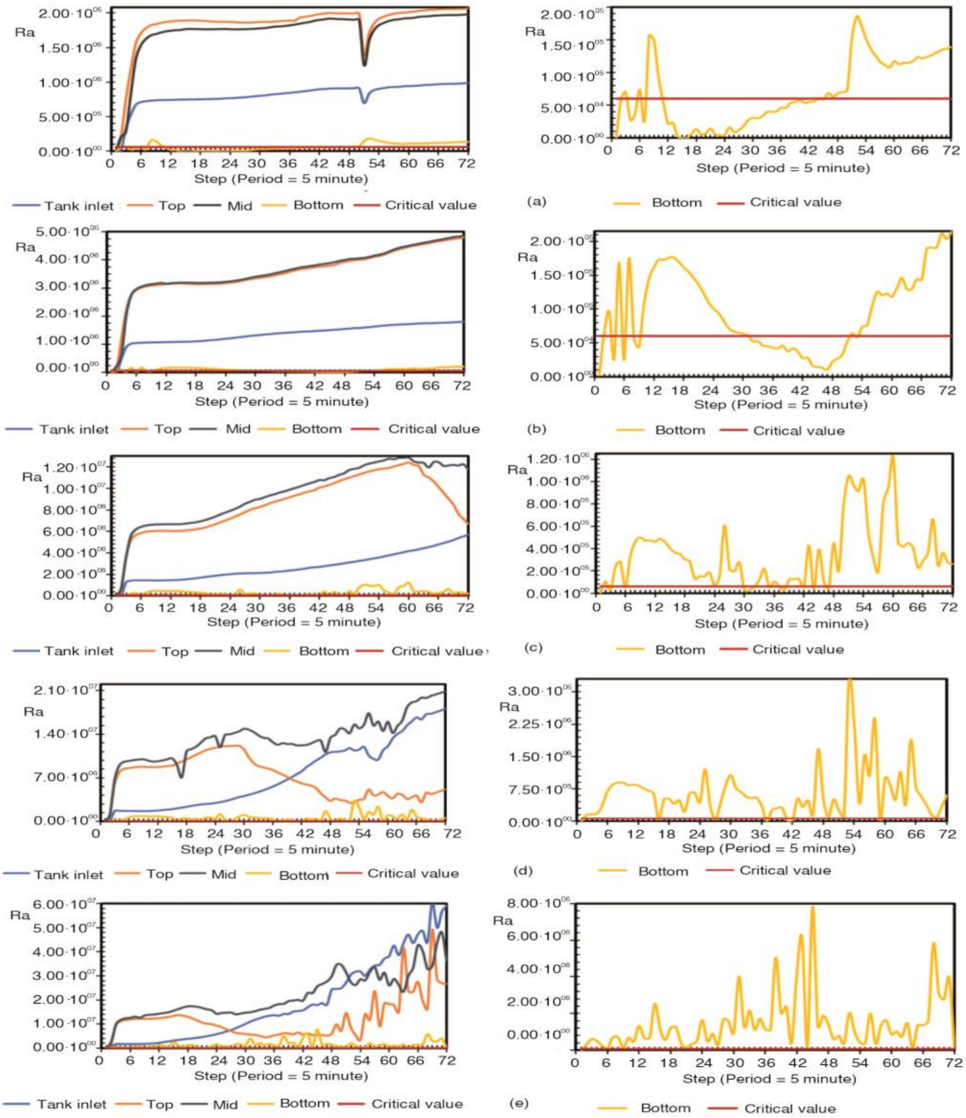

Figure 3. Rayleigh numbers for each experiment; (a) 100 W, (b) 200 W, (c) 400 W, (d) 600 W, and (e) 800 W (*for color image see journal web site*)

Figure 4. Flow values calculated according to the experimental data (*for color image see journal web site*)

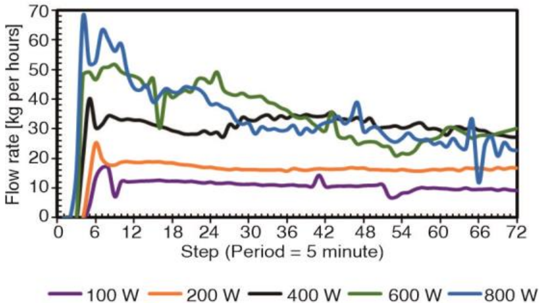

Table 1. Temperature prediction results

Experimental	Model	Error [%]	R^2	Experimental	Model	Error [%]	R^2
100 W	Li	8.76	0.9526	600 W	Li	17.99	0.9859
	Current	2.63	0.9961		Current	13.67	0.9947
200 W	Li	13.60	0.9268	800 W	Li	17.44	0.9917
	Current	4.41	0.9431		Current	4.35	0.9995
400 W	Li	17.71	0.9847	All experimental average	Li	15.1	0.9683
	Current	5.04	0.9991		Current	6.02	0,9865

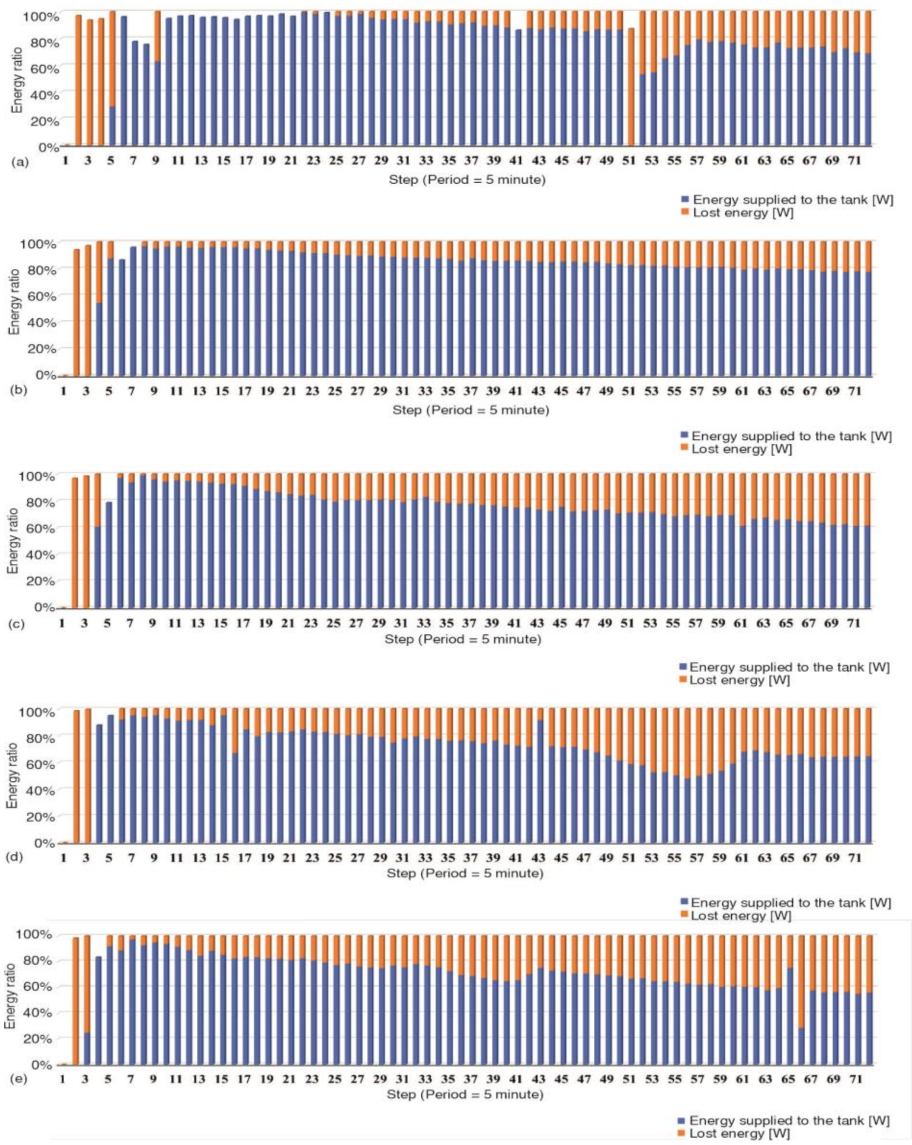

Figure 5. The energy supplied to the tank and lost energy; (a) 100 W, (b) 200 W, (c) 400 W, (d) 600 W, and (e) 800 W

The temperature of the pipe outlet (point 8 in fig. 2) obtained with the current model and the Li model, the percent relative error rates, and the graphs indicating the values against the measured temperature values are shown in fig. 6 for the experiments performed at 100 W, 200 W, 400 W, 600 W, and 800 W thermal power, respectively.

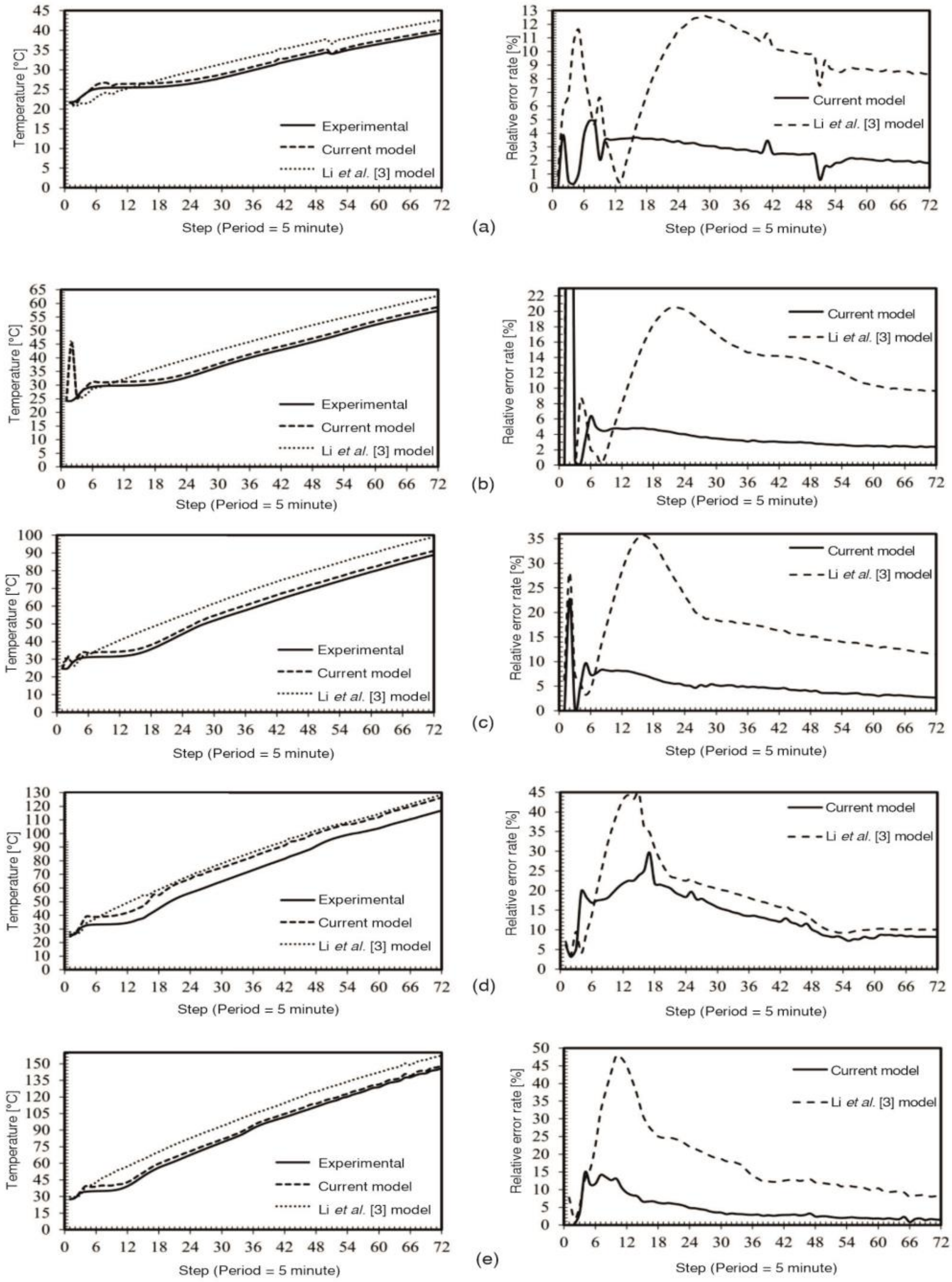

Figure 6. The percent relative error rates and the temperature values; (a) 100 W, (b) 200 W, (c) 400 W, (d) 600 W, and (e) 800 W

The following inferences can be made from these graphs:

– The current model gave more successful results than the Li model.
– The Li model shows the highest error rate of the model in the first hour of the experiments. The reason for this is that the average temperature of the tank is taken as the pipe inlet temperature in the Li model and the heat cannot be fully distributed throughout the tank at the beginning of the experiment.
– It is seen that the percent relative error rate of the Li model decreases significantly after 1-1.5 hours of the experiment. In the current model, this ratio varies less throughout the experiment. The reason for this difference between the two models is that while stratification is taken into account in the current model, the average temperature of the tank is accepted as the pipe inlet temperature and stratification is not taken into account in the Li model.
– The coefficient of determination, R^2, gave better results in the current model for each experiment.
– It is thought to be caused by reasons such as error rates, assumptions made in the model, and undesired heat losses.

Conclusions

In this study, a theoretical model has been developed to predict the temperature of the pipe outlet as well as the mass-flow rate of natural circulation of fluid in a one-ended inclined pipe. While creating the model, the energy balance in the tank was used. This model was compared with the Li model in the literature and more practical and successful results were obtained.

The important findings obtained in this study can be listed as follows.

- The mass-flow rate does not change much during the experiments at low thermal power (100 W and 200 W) and the natural circulation is stable. This situation is reversed in high thermal power experiments (400 W, 600 W, and 800 W).
- As the thermal power increases, the energy loss rate increases.
- The outlet temperature of the heat transfer oil from the closed inclined pipe reached approximately 45 °C, 65 °C, 100 °C, 130 °C, and 150 °C, respectively, for the experiments carried out at 100, 200, 400, 600, and 800 W thermal power at the end of the six-hour experiments.
- The current model is easier and more successful than the Li model.
- The approach that a streamline passes at the points numbered from 1 to 8 within the one-ended inclined pipe in fig. 2 yielded successful results.
- The existence of the natural circulation was questioned and the place and time of the occurrence of the stagnant regions were determined.
- It has been understood that the development of prediction methods is vital in such complex problems.
- Since the most commonplace of natural flow inside one-ended inclined pipes today is water heating systems, it is necessary to focus on different studies on this subject. For instance, values such as the temperature of the fluid along the pipe or the average temperature of the fluid in the tank can be estimated. Moreover, with the development of prediction methods with similar studies, measurement costs can be minimized.

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Nomenclature

A_t – surface area of the tank, [m^2]
c_p – specific heat, [Jkg^{-1}K^{-1}]
D – inside diameter of the pipe, [m]
d_e – hydraulic diameter, [–]

$\Delta\dot{E}_t$ – internal energy gained, [W]
g – gravitational acceleration, [ms^{-2}]
Ra – Rayleigh number, [–]
r – inner radius of the inclined pipe, [m]
T_∞ – ambient temperature, [K]
Δt – period, [5 minute = 300 second]
$\Delta\bar{T}_t$ – average temperature difference, [K]
$u_{t\text{-lost}}$ – thermal conductivity coefficient, [Wm^{-2}K^{-1}]
L – length of the pipe, [m]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
m_t – total mass of heat transfer oil, [kg]
$\dot{q}$ – useful energy, [W]
q_s – solar radiation, [Wm^{-2}]
$\dot{q}_{t\text{-lost}}$ – lost energy of the tank, [W]
Q – thermal energy, [W]
$\bar{T}_t$ – average tank temperature, [K]
$\forall_t$ – volume of the tank, [m^3]

Greek symbols

α – verification coefficient, [–]
β – thermal expansion coefficient, [K^{-1}]
θ – angle of inclined pipe, [º]
ρ – density, [kgm^{-3}]
ν – kinematic viscosity, [m^{2}s^{-1}]
μ – dynamic viscosity, [kgm^{-1}s^{-1}]

Subscripts and superscripts

i – inlet
o – outlet
t – tank

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