Effect of the shape of connecting pipes on the performance output of a closed-loop hot water solar Thermo-syphon

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ABSTRACT
In order to conserve the environment from pollution, which is caused by the use of the fossil fuels, numerous research works have been carried out in renewable energy area to minimize the dependency on the fossil fuels. There are several energy sources naturally available, and solar energy is considered to be the best amongst them. Therefore it became a motivating area for the researchers in recent years. Thermo-syphon is one of many devices that use solar energy for power generation. Thermo-syphon converts solar energy into internal energy of the working fluid; mainly water. In this work, a computational fluid dynamics (CFD) code has been used to analyse the natural convection phenomenon in a thermo-syphon. The thermo-syphon model consist of steel pipes with an internal diameter of 25mm, along with a condenser having diameter equal to five times the pipe's diameter, has been considered. The study has been carried out under no-loading conditions, for two thermo-syphon models comprising of straight and helical shaped pipes of 10, 20 and 30. A practical solar heat flux of 500W/m^2 has been applied on the pipes. The numerical results depict that the working fluid within the condenser, in case of helical pipes, gains higher temperature as compared to the straight pipes. Furthermore, increase in the number of helical pipes has negligibly small effect on the temperature of the fluid within the condenser, and hence on the performance output of the thermo-syphon.

Keywords
Computation Fluid Dynamics (CFD), Thermo-syphon, Helical pipe, Natural Convection

NOMENCLATURE

q	Heat flux (W/m^2)
U	Overall heat transfer coefficient (W/m^2. °C)
ΔT	the difference in temperature (C˚)
T	temperature of water within the condenser (°C)
T_{avg}	average temperature within the condenser (°C)
t	time of operation (minute)
t_{avg}	average time (minute)
nt	number of turns
np	number of pipes

1. INTRODUCTION
The amount of heat transfer through any material depends on several parameters; such as temperature variations, overall heat transfer coefficient, heat transfer surface area etc., as shown in Eq. (1).

$$q = U\Delta T \tag{1}$$

In recent decades, many researchers have been trying to improve the design of thermo-syphon in order to obtain excessive amount of useful heat energy. KE Amori et al. 2012 conducted a comparative study between a traditional absorber and a new design of solar collectors (known as the accelerated absorber) to analyse the performance of these systems. The performance evaluation was carried out in identical conditions for both the systems, having tilt angle of 33°. The evaluation tests were carried out in the presence of two different types of storage tanks. The results have shown that there is a significant increase in the thermal performance of the thermo-syphon (approximately 60%) for the new system. It has been observed that the temperature within the storage tank for the new design is 13C˚ higher to the conventional type. Subramanian et al. 2012 studied the impact of riser arrangement (zigzag pattern) on the performance of a flat plate collector system, and compared it with the conventional system. Experiments were conducted using copper tubes in header and riser having various geometrical characteristics. The results have shown that the performance efficiency reached 62.9% in the zigzag arrangement. El-Din et al. 2005 experimentally investigated the properties evaluation of the heat transfer in single-phase flow. In their study, a toroidal thermo-syphon type has been used. The parameters of investigation include heated-cooled length ratio, heated length tube diameter ratio, diameter ratio of torus-tube, and angle of inclination. Their results show that the increase in both heated-cooled length ratio and heated length-tube diameter ratio leads to decrease in the heat transfer rate, whereas increase in torus-tube diameter ratio increases the heat transfer rate. Furthermore, it was found out that the range of tilt angles between 30° and 45° produces maximum heat transfer rate. Freegah et al 2013 numerically studied the effects of the length to diameter ratio of the riser, number of connecting pipes, angle of inclination of the thermo-syphon and the heat flux, on the performance of the thermo-syphon. It was found that the heat flux and the length to diameter ratio of the pipes have significant effects on the performance of a thermo-syphon, whereas, the angle of inclination has negligibly small effect. Furthermore, an increase in the number of connecting pipes increases the temperature of the working fluid, as they absorb more solar energy. Gurveer et al 2014 conducted an experimental study to investigate the effects of the inclination angle, wire coil inserts and wire mesh inserts on the thermal performance of a flat-plate solar collector. The results that have been reported indicate that the Nusselt number in novel insert configuration is higher as compared to the conventional system, and hence the thermal performance for the novel insert configuration has been observed to be better than the conventional one.

This study is the continuation of Freegah et al. 2013 and hence the natural convection phenomena, and the distribution of temperature and velocity of the working fluid, has not been discussed in detail in the present study. Computational Fluid Dynamics based tools have been used to carry out an extensive numerical study, on the effects of using helical pipes, on the performance of a closed-loop solar hot water thermo-syphon system. The effects of helical pipes in a thermo-syphon have not been explicitly analysed in the literature, and hence this study is important for the design process of such systems.

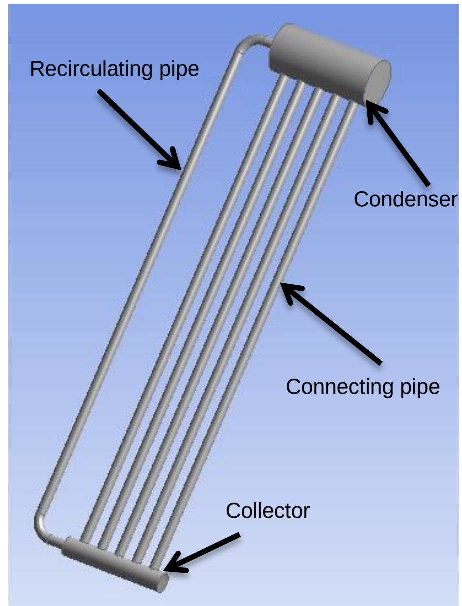
2. NUMERICAL MODELLING

Two thermo-syphon configurations, comprising of helical and straight pipes, have been modelled. The geometry of the two models is shown in Figure 1. An internal diameter of 25mm has been used for both helical and straight connecting pipes, with a thickness of 2mm. Furthermore, the recirculating pipe, which has been used in both the models, has the same diameter and thickness as that of the connecting pipes. It has been assumed that the thermo-syphon is operating under no-load condition. The diameter of the condenser is five times the diameter of the pipes, and the diameter of the collector is twice as that of the pipes. The working fluid considered is water.

Hybrid meshing has been employed, using both hexagonal and tetrahedral elements. Non-uniform mesh distribution has been used, where the mesh elements are concentrated near the wall region, using 10 layers of mesh elements. The mesh contains two million elements, and has been shown previously to describe the flow phenomena with reasonable accuracy. Furthermore, a time step size of 12sec has been used.

Boussinesq approximation has been employed to accurately model buoyant forces being generated. This approximation states that the density differences are sufficiently small to be neglected, except where they appear in terms multiplied by g i.e. the acceleration due to gravity. The essence of the Boussinesq approximation is that the difference in inertia is negligible, but gravity is sufficiently strong to make the specific weight appreciably different. Furthermore, it has been observed by Dehdakhel et. al. 2010 that the Boussinesq approach for the density of the working fluid in a thermo-syphon gives fairly accurate results, and thus has been used in the present study.

Three dimensional Navier-Stokes equations, in addition to the continuity and the energy equations, have been numerically solved in an iterative manner to simulate the transient flow of water in the thermo-syphon for one hour of operational time.



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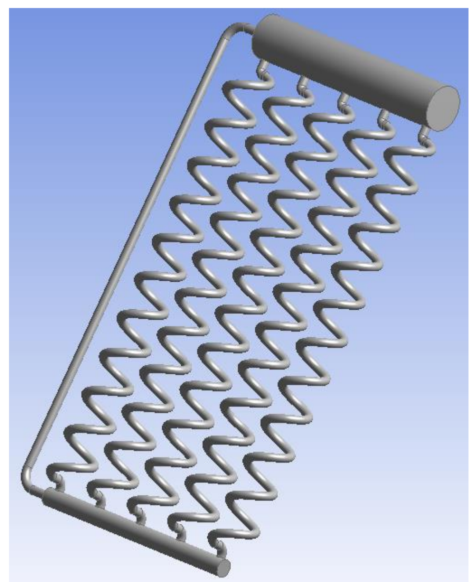


(b)

Figure 1. The geometry of the thermo-syphon (a) straight pipe and (b) helical pipe model.

3. Results and Analysis

Freegah et al 2013 successfully simulated the natural convection phenomena in thermo-syphon; the simulations conducted to investigate the effect of number of the straight pipes, the tilt angle and the length-to-diameter ratio on thermo-syphon's performance. Authors presented numerical results in the form of temperature contour within the thermo-syphon to show the natural convection phenomenon (figure 2).

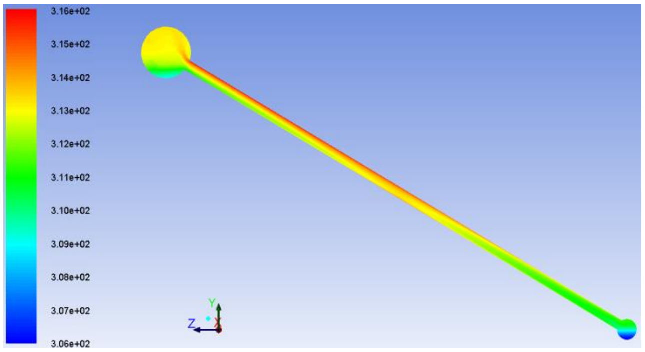


Figure 2 - Temperature variations in the working fluid in the thermo-syphon after 1 hour of operation

In the present study, which is a continuation of Freegah et al 2013, the effect of the number of turns in a helical pipe, on the temperature within the condenser, has been numerical analysed. The boundary conditions are the same for all the numerical simulations.

3.1. Straight pipes

Figure 3 depicts the temperature distribution within the cross-section of the condenser of the thermo-syphon model comprising of straight connecting pipes. It can be clearly seen that the hot working fluid occupies the upper section of the condenser while the cold working fluid settles on the bottom of the condenser.

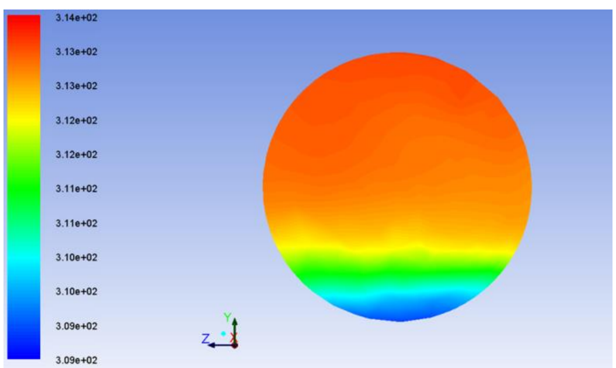


Figure 3. Temperature variations in the condenser after 1 hour of operation

Figure 4 depicts the variation in working fluid's temperature within the cross-section of the condenser for variable number of the connecting pipes. It can be seen that the temperature within the condenser is slightly higher for five connecting pipes as compared to three connecting pipes. It is obvious from the fact that more number of connecting pipes transfers more hot fluid to the condenser, increasing its temperature. After one hour of operation, the difference in the condenser's temperature is 2^{o}C for both thermo-syphon configurations.

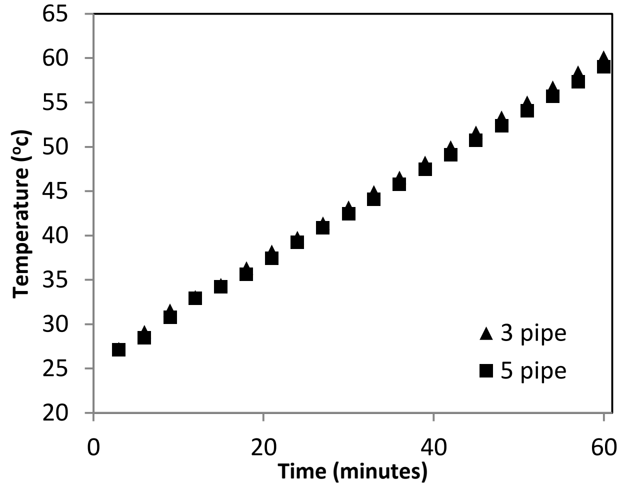


Figure 4. Variations in temperature of the working fluid within the condenser for various numbers of straight pipes

3.2. Helical pipes

Figure 5 shows comparison of the temperature within the condenser for the straight and helical connecting pipes comprising of 10 turns. It can be clearly seen that the hot working fluid occupies the upper section of the condenser while the cold working fluid settles on the bottom of the condenser. It can also be seen that the average temperature of water is higher for helical pipe as compared to the straight pipe. This shows that by using helical pipes, the temperature within the condenser can be increased.

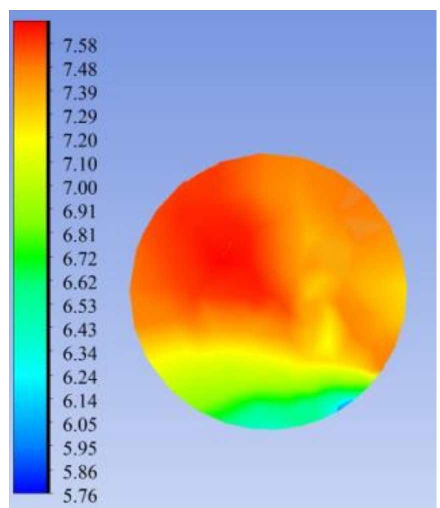
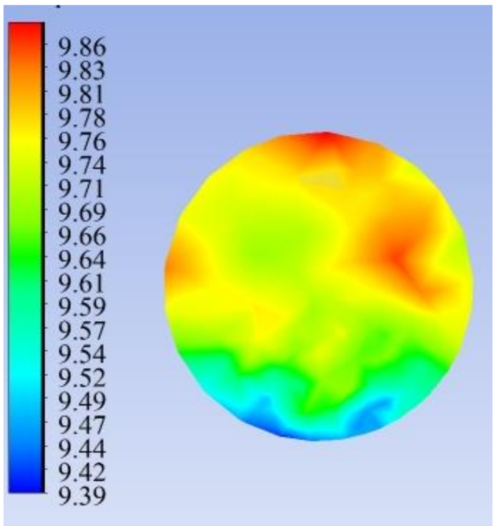


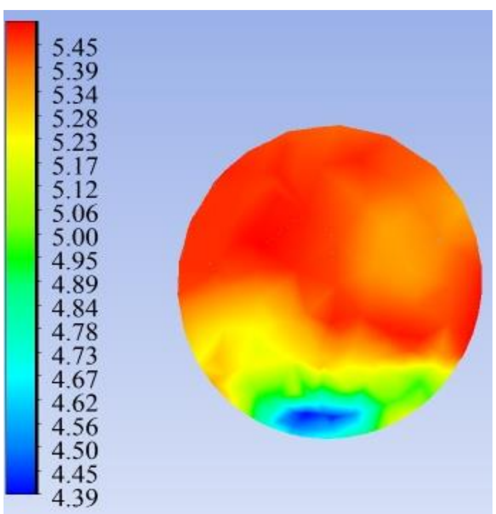
Figure 5. Comparison in condenser's temperature between straight pipe and helical pipe models consisting of 10 turns

Figure 6 depicts the effect of the number of turns of the helical pipes on the condenser's temperature after one hour of operating time. Figure 2(a) shows the difference between 20 and 10 turns, while figure 2(b) shows the difference between 30 and 20 turns. It can be seen that the temperature of the working fluid increases as the number of turns increases. This is true for both the cases i.e. increase from 10 turns to 20 turns, and increase from 20 turns to

30 turns. After one hour of operation, temperature difference within the condenser for straight pipes, 10 turns, 20 turns and 30 turns are 9.86°C, 9.3°C, 5.4°C, 4.39°C respectively.



(a)



(b)

Figure 6. Comparison in condenser's temperature between (a) 20 and 10 turns (b) 30 and 20 turns

Figure 7 depicts the variations in the condenser's temperature for three helical models. It can be seen that the temperature of the condenser increases linearly in all the different cases. Furthermore, the condenser's temperature is considerably higher in case of 30 turns as compared to 20 and 10 turns' cases, after one hour of operation. Meanwhile, the difference in the condenser's temperature between 10 turns and 20 turns is significantly higher than the difference between 20 turns and 30 turns. This is because the increase in the volume of the working fluid from 10 turns to 20 turns model is 100%, whereas it is 50% from 20 turns to 30 turns.

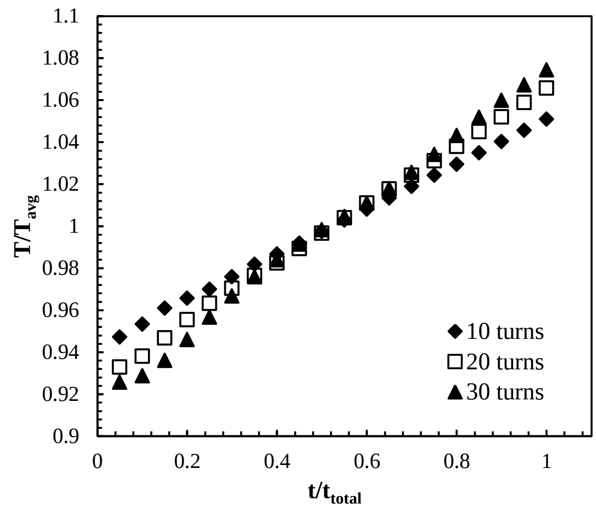


Figure 7. Variations in condenser's temperature for three models consisting of 10, 20, and 30 turns

Figure 8 depicts the variations in the condenser's temperature with 10 turns configuration, however with different number of helical pipes (3 and 5 connecting pipes). After one hour of operating time, the difference in condenser's temperature is 1°C between 3 and 5 connecting pipes. It can be seen that the working fluid's temperature within the condenser is negligibly higher for 5 connecting pipes as compared to 3 connecting pipes.

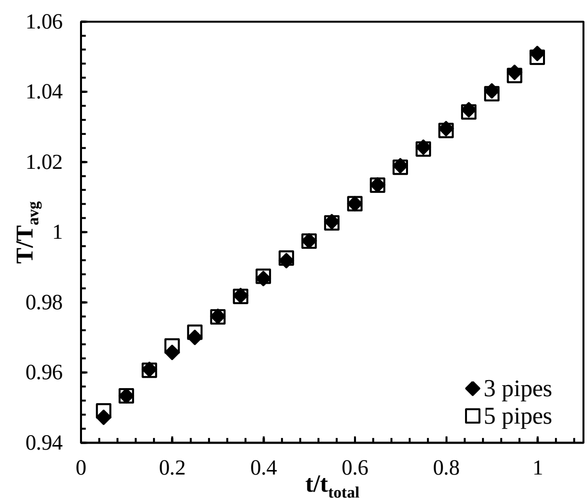


Figure 8. Variations in condenser's temperature for two helical models consisting of 3 and 5 connecting pipes

Temperature variations in the working fluid within the condenser for straight and helical pipes, at different times of operation, have been summarised in table 1. It can be clearly see that increase in time of operation and number of turns lead to increased temperature within the condenser. For example, for three connecting pipes, after one hour of operation, the temperature of the working fluid increases by 12.66%, 24.78%, and 30.10% for 10, 20 and30 turns respectively, compared to straight pipes model.

Table 1. Temperature of water in straight and helical pipe models at different times of operation

Time (minute)		15	30	45	60
Temperature in 3 straight pipes model (C˚)		33.40	39.80	46.10	52.40
Temperature in 5 straight pipes model (C˚)		34.81	42.54	50.19	57.79
Temperature in 3 helical pipes model (C˚)	10 turns	34.40	43.10	51.60	60.10
	20 turns	36.80	47.50	58.60	69.80
	30 turns	37.10	50.50	62.10	75.10
Temperature in 5 helical pipes model (C˚)	10 turns	34.20	42.20	50.71	58.99
	20 turns	36.06	46.53	57.67	68.54
	30 turns	35.29	50.16	60.51	74.13
	20 turns	3.40	8.50	12.90	15.67
	30 turns	1.35	15.18	17.10	22.03

From the numerical results, using multiple regression analysis, equation (2) has been formulated in order to calculate working fluid's temperature within the condenser for various thermo-syphon configurations. In order to check validity of Eq. (2) T/T_{avg} values have been calculated using Eq. (2), and compared against the results presented in table 1. It has been observed that Eq. (2) gives an average percentage error of 9.3% compared to the results presented in table 1, and hence this equation can be used to predict the temperature within the condenser with 90.7% accuracy.

$$\frac{\mathrm{T}}{\mathrm{T_{ave}}} = \begin{cases} \dfrac{(10^{0.028})\left(\frac{\mathrm{t}}{\mathrm{tavg}}\right)^{0.454}}{(\mathrm{nt})^{0.0124}(\mathrm{np})^{0.00137}} & \text{for } 0 < \left(\frac{\mathrm{t}}{\mathrm{tavg}}\right) \leq 0.8 \\ (10^{0.028})\left(\frac{\mathrm{t}}{\mathrm{tavg}}\right)^{0.454}(\mathrm{nt})^{0.0124}(\mathrm{np})^{0.00137} & \text{for } 0.8 < \left(\frac{\mathrm{t}}{\mathrm{tavg}}\right) \leq 1.6 \end{cases} \tag{2}$$

4. Conclusions

In the present work, CFD simulations have been conducted for thermal performance evaluation for two types of thermo-syphon, namely helical connecting pipes and conventional thermo-syphon. From the numerical results, it can be concluded that a considerable enhancement in the performance output of the thermo-syphon is obtained for the helical pipe configurations, in comparison with the conventional model. Furthermore, increasing the number of turns in the helical connecting pipes increases the condenser's temperature. Moreover, increasing the number of helical connecting pipes does not enhance the performance of the thermo-syphon significantly. It is expected that this study will help in the design process of thermo-syphons with optimal thermal performance.

5. References

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