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Numerical Investigation of Thermal Losses from Air Filled Annulus of a Parabolic Trough Solar Collector

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ABSTRACT

Solar energy has the potential to meet the growing need for global energy consumption. In recent times there is large number of solar energy systems developed, one of such widely used technology is parabolic trough collector (PTC). The PTC technology is one of the successful technologies because it is the most mature and one of the least expensive. PTC receivers with air filled annuli are used mainly for high temperature applications such as food processing industry. This is due to the fact that they are less costly but on the other hand they have high heat loss as compared to vacuum receivers. One of the techniques that can be adopted in order to enhance the thermal performance of the PTC has been discussed in this work. An insulation fiberglass with high heat resistant was inserted into the portion of the receiver annulus that does not receive concentrated sunlight. This study focuses on the calculation of conduction and convection heat losses of the half insulated annulus part only. The performance of the proposed concept was then compared to conventional receiver with air filled annulus. The effect of wind speed and mass flow rate of the working fluid was also considered. The reason for using mass flow rate and wind speed as manipulated variable is because both parameters affect the thermal loss of the system. The results shown that the heat loss from the half insulated is smaller compared to the heat loss from air annulus receiver by 70% depending on the wind speed of the location. Therefore, the proposed receiver is expected to be most suitable replacement for receivers with air annulus.

Keywords: *Solar, Parabolic Trough Collector, Heat Loss, Heat Transfer*

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Introduction

Sustainable renewable energy is derived from natural resources that are continously being replenished, one of such example is the solar energy that is derived directly from the sun [1]. In recent times solar thermal energy has been predicted to be clean energy of tomorrow. Solar energy has been the main agenda of energy development in many developed countries such as Germany and USA and it is a potential source of energy for developing countries such as Malaysia. It is however expected that, through the application of renewable energy technologies, the global renewable energy consumption wouls reach 318 exajoules by the year 2050 [1]. One of the advantages of solar energy is that, it is clean and can be supplied without any environment pollution [2,3]. Fossil energy also could be saved if every house uses more solar energy as their primary heat sources [3]. Parabolic trough collector, solar power tower and parabolic dish collector are three basic principal types of solar energy collector that are available for high temperature application [1]. Parabolic trough collector (PTC) technology is a type of concentrated solar power (CSP) conversion [4]. During the past decades, CSP technology has been important option for harnessing solar energy [5]. PTC are the most mature solar energy and able to generate heat at temperatures up to 400 °C [7,8].

There are several successful commercial PTC for CSP plants that have been tested in a temperature range of 300-400 °C [8]. The use of PTC in the electric generation system has been associated with numerous losses including optical, thermal and geometry losses [9]. The PTC collectors available today uses expensive receivers due to the effort put in place to maintain vacuum in their annuli. Vacuum annuli has the tendency to reduce convection losses. Basically, the receiver of a parabolic trough collector consists of outer transparent enclosure, typically made of glass and inner absorber tube, made of steel (see Figure 1 for details). Al-Ansary and Zeitoun [10] proposed a technique by fitting a heat resistant thermal insulation material into the portion of the receiver annulus that does not receive concentrated sunlight. The result shows that heat loss from the proposed receiver can be smaller than that from a receiver with an air filled annulus as much as 25% when fiberglass is used. In recent study Tijani and Roslan [9] presented a detailed model of heat loss in parabolic trough receivers. Hachicha et al. [11] presented a numerical study based on Large-Eddy Simulations (LES) models for the simulation of the fluid flow and heat transfer around a parabolic trough solar collector and its receiver tube is performed. The aim of this manuscript is to numerically analyse the thermal losses associated with PTC. The effect of changes in environmental conditions (such as wind speed) on heat loss was evaluated. This research was focused on the heat losses that occur at the insulation part which doesn't receive concentrated sunlight. The thermal losses models were simulated using ANSYS fluent solver.

Description of Parabolic Trough Collector (PTC) Module

Parabolic Trough Collectors (PTCs) are linear focus concentrating solar devices suitable for working in high temperature application of 150-400 °C temperature range. A PTC is made up of a parabolic trough shaped mirror that reflects direct solar radiation, concentrating it onto a receiver tube located in onto a receiver tube located in the focal line of the parabola [11]. The energy from sunlight which falls on the surface of the mirror parallel to its plane of symmetry is focused along the focal line. The concentrated radiation heats the fluid that circulates through the receiver absorber tube, thus transforming the solar radiation into thermal energy in the form of the sensible heat of the fluid.

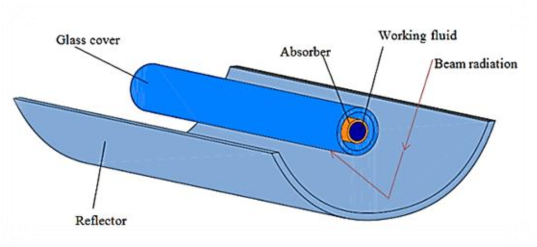


Figure 1: Schematic representation of a parabolic trough collector

Table 1: LS-2 PTC geometry [9]

Item	**Value**
Collector aperture area	39 m^2
Collector aperture width	5 m
Collector aperture length	7.8 m
Glass envelope outer diameter	0.115 m
Glass envelope inner diameter	0.109 m
Absorber tube outer diameter	0.07 m
Absorber tube inner diameter	0.065 m

The heat transfer fluid flows at the receiver of a parabolic trough collector consist of an inner absorber tube made of steel and an outer transparent enclosure made of glass. The objective of using the transparent enclosure is to minimize heat loss by convection to the surrounding due to the interaction between the hot absorber tube surface and the ambient air. The outer glass tube is attached to the steel pipe by means of flexible metal differential expansion joints which compensate for the different thermal expansion of glass and steel when the receiver tube is working at nominal temperature. A schematic representation of a single parabolic trough collector module and its technical specifications are given in Figure 1 and Table 1 respectively.

Losses in Parabolic trough collector

There are three main types of losses associated with parabolic trough collectors [11]:

a) Optical losses
b) Thermal losses from the absorber pipe to the ambient
c) Geometrical losses

For this project, we focused on the heat losses from a half insulated air-filled annulus of the receiver of a parabolic trough collector focusing on thermal losses from the absorber pipe to the inner glass cover.

A summary of a model of the thermal losses from the collector can be developed using a combination of the collector thermal loss from radiation, conduction and convection, as listed in Figure 2.

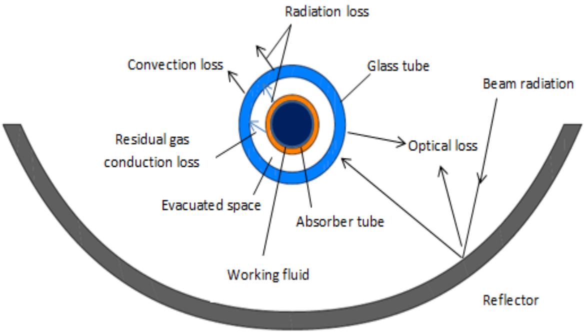


Figure 2: Modes of heat loss from an evacuated tubular absorber

1. Thermal loss from the absorber tube outer wall to the evacuated glass tube (surrounding the absorber) occurs by radiation and residual gas conduction. Due to the high vacuum in the absorber element ($10^{-4}\,mmHg$) convection is normally negligible, however it can be significant if the pressure is allowed to increase.
2. Heat loss from the absorber tube to the ambient also occurs via the vacuum bellows and supports. Heat loss from the glass cover tube occurs by radiation to the sky and by convection.

Description of Half Insulated Air Filled Annulus

The physical model of the investigated problem is illustrated schematically in Figure 3. Thermal insulation is used to fill half of the annulus while the other half is filled with air. Hot water fluid flows inside the tube, transferring heat to the surroundings. The flow is assumed to be steady and laminar flow of about 200 Reynolds number was simulated. From the heat transfer mechanism, the inner absorber tube is expected to have a higher temperature than the outer absorber tube; this is due to the convection effect on the outer surface of the absorber tube. During the same period, conduction and radiation heat transfer will take place within the half insulated annulus. A heat resistant insulating material fiberglass is chosen as the insulator. For efficiency improvement and less energy loss half insulated air filed annulus was proposed. The hypothesis is that only the half insulated air filed annulus is exposed to the direct irradiation where as the top part is insulated with fibre glass so as to resist heat loss by convection.

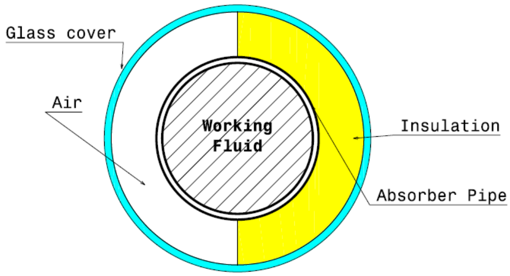


Figure 3: Half insulated-half stationary air filled annulus of PT

Thermal Analysis of Receiver

Convection Heat Transfer between the Heat Transfer Fluid (HTF) and the Receiver Pipe

Based on the Newton's law, the convection heat transfer from the inner surface of the receiver pipe to the HTF is given by $hA(T_s-T_\infty)$ [12].

$$q_{f-pi,conv} = h_f \pi D_{pi} (T_{pi}-T_f) \tag{1}$$

Where,

h_f= HTF convection heat transfer coefficient $(W/m^2 - °C)$

D_{pi}= Diameter of inner pipe (m)
T_{pi}= Pipe inner temperature (°C)
T_f= Fluid temperature (°C)

Conduction Heat Transfer through the Receiver Pipe Wall

The Fourier's Law of conduction through hollow cylinder is used to determined conduction heat transfer through the receiver pipe. The equation given [13] :

$$q_{pi-po,cond} = \frac{2\pi k_{pipe}(T_{pi}-T_{po})}{\ln(\frac{D_{po}}{D_{pi}})} \quad (2)$$

Where,

k = Thermal conductivity of the pipe W/m-K.

In the equation thermal conductivity is constant and depends on the receiver pipe material type. If steel is used, the thermal conductivity is 16.27 W/m-K.

Convection Heat Transfer from Receiver to Glass Envelope

Air in annulus: The convection heat transfer equation between the receiver pipe and glass envelope when there is air in annulus are [14]:

$$q_{po-gi,conv} = \frac{2\pi k_{eff}}{\ln(\frac{D_{gi}}{D_{po}})}(T_{gi} - T_{po}) \quad (3)$$

Where,

k_{eff}= effective thermal conductivity (W/m-K)

Radiation Heat Transfer

The radiation heat transfer equation between the receiver pipe and glass envelope ($q_{po-gi,rad}$) are [6] :

$$q_{po-gi,rad} = \frac{\sigma\pi D_{po}\left(T_{po}^4 - T_{gi}^4\right)}{\left(\frac{1}{\varepsilon_{po}} + \left(\frac{\left(1-\varepsilon_{gi}\right)D_{po}}{\varepsilon_{gi}D_{gi}}\right)\right)} \quad (4)$$

Where,

ε_{po}= Emissivity for pipe outer
ε_{gi}= Emissivity for pipe inner

Fluid Dynamics

The partial differential equations governing the fluid flow and heat transfer in the enclosure include the continuity, the Navier-Stokes and the energy equations. The assumptions that need to be considered are the flow is steady and laminar, long enclosure, flow in the gap is two dimensional and the fluid is assumed to be Newtonian. The continuity equation may be written as shown by Al-Ansary et. al [10]:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{5}$$

The momentum equations in x, y and z directions can be written as:

$$\rho u\frac{\partial u}{\partial x} + \rho v\frac{\partial u}{\partial y} + \rho\frac{\partial w}{\partial z} = -\rho g_x - \frac{\partial P}{\partial x} + \mu[\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}] \tag{6}$$

$$\rho u\frac{\partial v}{\partial x} + \rho v\frac{\partial v}{\partial y} + \rho\frac{\partial w}{\partial z} = -\rho g_y - \frac{\partial P}{\partial y} + \mu[\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}] \tag{7}$$

$$\rho u\frac{\partial w}{\partial x} + \rho v\frac{\partial w}{\partial y} + \rho\frac{\partial w}{\partial z} = -\rho g_z - \frac{\partial P}{\partial z} + \mu[\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}] \tag{8}$$

The energy equations for the air in the gap and the insulation can be written as:

$$\rho C_p u\frac{\partial T}{\partial x} + \rho C_p v\frac{\partial T}{\partial y} + \rho C_p w\frac{\partial T}{\partial z} = k[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}] \tag{9}$$

$$\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 0 \tag{10}$$

For natural convection, the change in density is responsible for inducing the flow. The fluid density is estimated using the ideal gas equation of state, which is provided as an input to the problem. The reference pressure is assumed equal to the standard atmospheric pressure. The properties of air are assumed temperature dependent. The boundary conditions for the current problem are:

- No slip condition along the interface between air and the inner and outer cylindrical surfaces.
- The convection coefficient at the outer surface is h_0 and the ouside ambient temperature is T_0.

- The convection coefficient at the inner surface is h_i and the inside bulk fluid temperature is T_i.

Simulation Model of the PTC

In order to design the geometry model of the PTC, CATIA software was used. The parameters of the PTC geometry are similar to LS-2 PTC (refer to Figure1 and Table 1). After the PTC has been modeled in CATIA, the model is then exported to ANSYS fluent solver for meshing. Meshing is a process of dividing or discretizing the model domain into a finite number of smaller elements. After the meshing process was completed, the domains are defined as fluid or solid according to their respective condition. Figure 4 shows isometric view of half insulated annulus. The material of half insulated annulus is made from fiber.

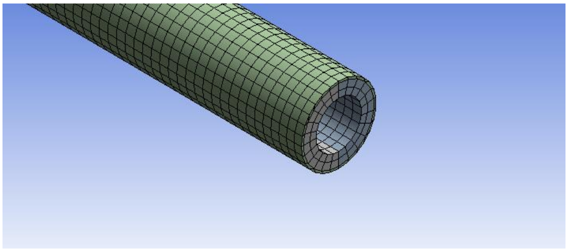


Figure 4: Isometric view of meshed half insulated annulus

Materials which make up the receiver are created or taken from FLUENT library. The thermal properties of the materials are defined as Table 2 below:

Table 2: Materials properties

Material	**Properties**			
	ρ (kg/m^3)	**C_p (J/kg-K)**	**k (W/m-K)**	**μ (kg/m-s)**
Water-liquid	998.2	4182	0.6	0.001003
Air	1.225	1006.43	0.0242	1.7894e-05
Pyrex	2225	835	1.4	
Steel	8030	502.48	16.27	
Fiberglass	48	843	0.037492	

Result and Discussion

In this study the total heat loss due to convection and radiation for air annuli was conducted. It can be observed from Table 3 that the total heat loss for the combination of convection and radiation ranges between 34.90 W/m to 75.14 W/m. The heat loss for half insulated annulus was lower compared to that fully air annulus (see Table 4).

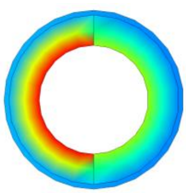
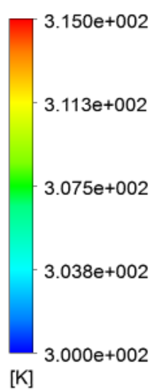
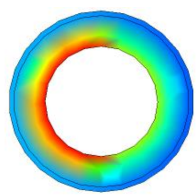
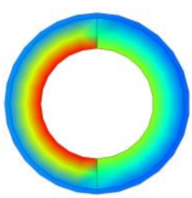
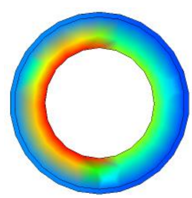
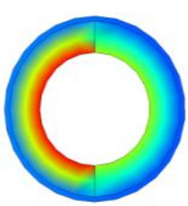
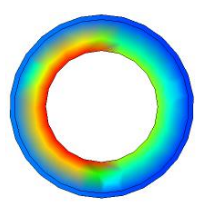
Table 3: Total heat loss for fully air annulus

Heat loss by convection (W/m)	Heat loss by radiation (W/m)	Total Heat Loss (W/m)
18.20	16.70	34.90
26.91	17.43	44.35
33.62	17.77	51.39
39.28	17.97	57.24
44.26	18.11	62.37
48.77	18.21	66.98
52.92	18.29	71.21
56.78	18.35	75.14

Table 4: Total heat loss for a half insulated

Heat loss by conduction (W/m)	Heat loss by radiation (W/m)	Σ Heat Loss (W/m)
3.80	15.78	19.57
4.03	16.70	20.73
4.14	17.13	21.27
4.20	17.40	21.60
4.25	17.58	21.83
4.29	17.72	22.01
4.31	17.83	22.14
4.34	17.92	22.25

The generated data are represented in graphical form to assist in observation, analysis and discussion purpose. The trends and patterns resulting from the simulation are then validated with previously done research study by other researcher to ensure justification of the simulation data. Figure 7 shows the temperature distribution of half insulated annulus and fully insulted annulus. For a half insulated annulus the temperature distribution is highest (315 K) at the bottom part of the absorber tube, where as the lower part has a lower temperature distribution, this shows that much of the heat is trapped at the bottom part of the absorber and this can easily be transfered to the working fluid. The presence of insulation material reduces heat transfer to due to convection and conduction to the surrounding. When compared to the fully air annulus, it can be observed that the temperature distribution is not concentrated at the bottom part of the absorber where it is most needed, instead it is extends to the top part of the absober. This phenomenon is due to the convection heat transfer initiated by the air trapped in the annulus. This concludes that the top part of the absorber is susceptible to more heat losses due to convection to the surrounding. This explanation has been justified by the results obtain in Table 3 and Table 4. The result is also similar to that found in the literature [10].

Wind speed, m/s	**Temperature contours for half insulated annulus**	**Temperature (K)**	**Temperature contours for fully air annulus**
0.5			
1.0			
1.5			

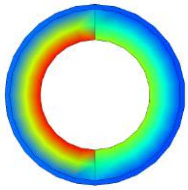
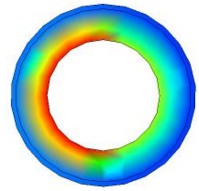
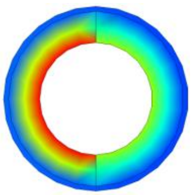
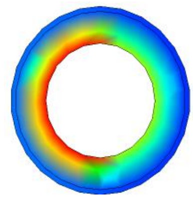
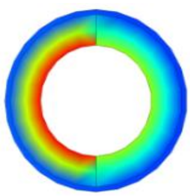
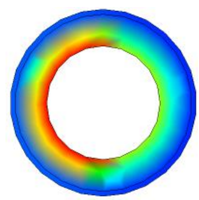
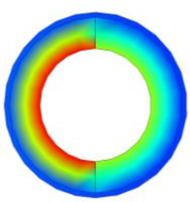
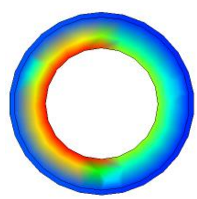
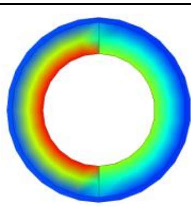
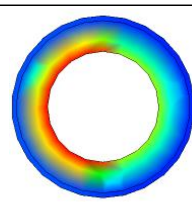
Wind speed, m/s	**Temperature contours for half insulated annulus**	**Temperature (K)**	**Temperature contours for fully air annulus**
2.0			
2.5			
3.0			
3.5			
4.0			

Figure 5: Isothermal zones in half insulated annulus and fully air annulus

Based on Figure 5, it can be observed that the heat loss for half insulated annulus is less compared to the transfer of heat loss for fully air annulus. With increasing of wind speed from 0.5 m/s to 4.0 m/s, the transfer of heat loss also will increase. The heat loss transfer for fully air annulus is kept increasing when the wind speed is increases, this is due to the value of convection which depends on the wind speed.

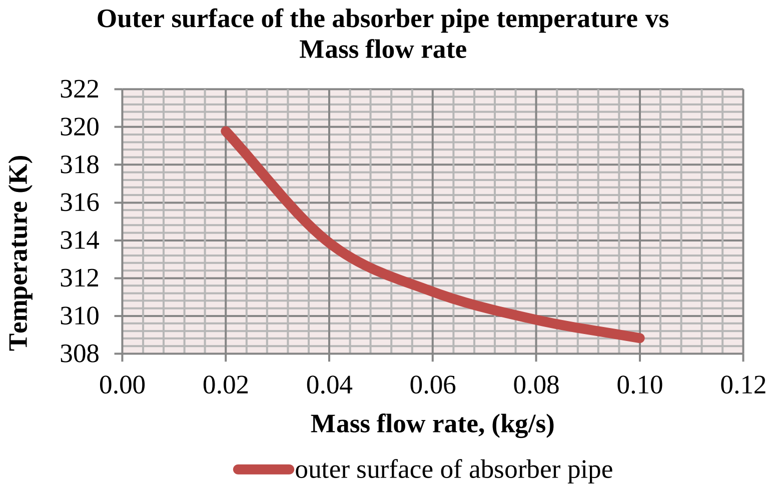


Figure 6: Effect of mass flow rate of HTF on temperature of outer surface of the absorber pipe.

It can be observed from Figure 6 that the outer surface of pipe temperature is inversely proportional to the mass flow rate of air.Iincreasing mass flow rate has resulted in an increase of outer sureface pipe temperature. It means that mass flow rate has a significant influence on the insulated material and glass envelope temperature which eventually affect the thermal losses of the system.

Figure 7 shows that the total heat loss is directly proportional to wind speed velocity for both half insulated annulus and fully air annulus. The drastic heat loss for fully air annulus was observed due to fact that the convective heat transfer coefficient is directly proportional to the value of wind speed. While for half insulated material have only experience small change in total loss because of the thermal conductivity value is constant and not affected by the value of wind speed. The highest total heat loss 170.52 W/m was observed for fully air annulus with mass flow rate of 0.02 kg/s and wind speed of 4 m/s. While the lowest total heat loss 19.57 W/m was observed for half insulated with mass flow rate 0.1 kg/s and wind speed of 0.5 m/s.

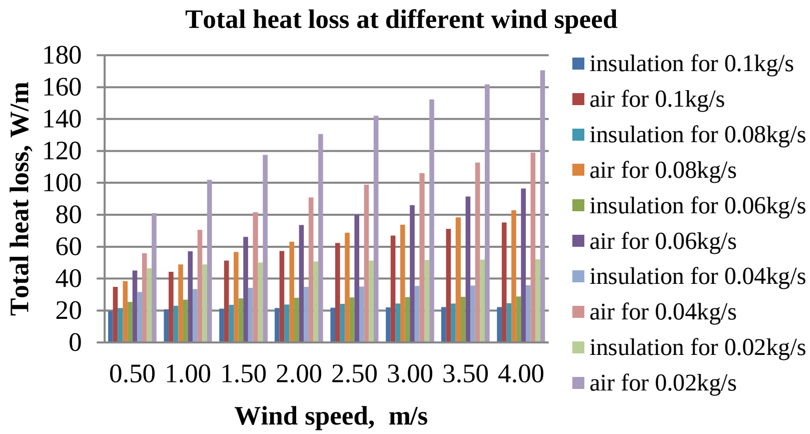


Figure 7: Effect of total heat loss at different mass flow rate and different wind speed

From Figure 8 it can be observed that the percentage of heat loss for half insulated air annulus is directly increase when wind speed also increase. The higher value of mass flow rate and wind speed the higher the percentage decrease compare to air filled annulus. The highest percentage value is 70.38 %. The highest percentage decrease value is at wind speed 4 m/s and mass flow rate 0.1 kg/s. This happen because at strong wind speed the convection heat loss at air annulus is very high. The lowest percentage is when mass flow rate 0.1 kg/s and wind speed 0.5 m/s. The lowest percentage value is 42.66 %.

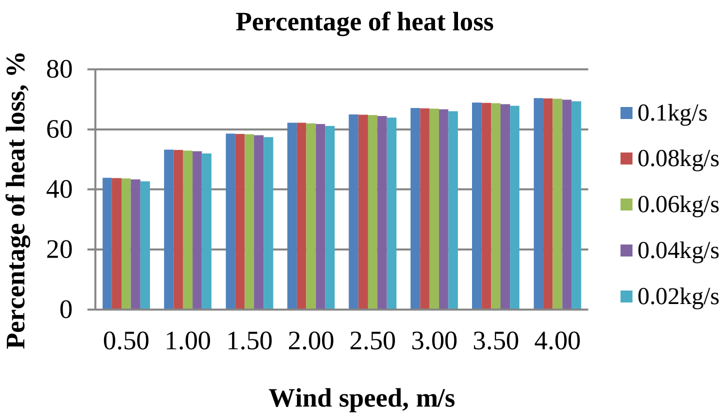


Figure 8: Percentage of heat loss for half insulated compare to fully air annulus

Conclusion

This reasearch manuscript examines the possiblity to reduce thermal losses associated with the system by means of insulation material and to evaluate the effect of wind speed on heat loss. The percentage of heat loss for half insulated annulus can achieve up to 70% depending on the mass flow rate and wind speed. The numerical simulation of the model has been developed using ANSYS FLUENT solver which give the resulting temperature of outward glass envelope as compared to the temperature of the pipe absorber. The thermal loss has been numerically calculated from the resulting glass envelope temperature and pipe absorber. The model was stimulated with different mass flow rate and wind speed to observe the different thermal heat loss behavior.

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